Surface Production Operations

PUMPS AND COMPRESSORS

Maurice Stewart

VOLUME FOUR

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Volume IV

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Gulf Professional Publishing
An imprint of Elsevier

Gulf Professional Publishing is an imprint of Elsevier
50 Hampshire Street, 5th Floor, Cambridge, MA 02139, United States
The Boulevard, Langford Lane, Kidlington, Oxford, OX5 1GB, United Kingdom

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Library of Congress Cataloging-in-Publication Data
A catalog record for this book is available from the Library of Congress

British Library Cataloguing-in-Publication Data
A catalogue record for this book is available from the British Library

ISBN: 978-0-12-809895-0

Publisher: Joe Hayton
Senior Acquisition Editor: Katie Hammon
Editorial Project Manager: Joanna Collett
Production Project Manager: Sruthi Satheesh
Cover Designer: Alan Studholme

Typeset by SPi Global, India

Dedication

Dedicated to my wife, Eli; my son, Chad;
and in memory of my parents,
Maurice and Bessie Stewart.

Acknowledgments

This book is essentially the summary of the knowledge accumulated by the author through 45 plus years of practice as a facility engineer in the upstream oil and gas industry. I would like to take this opportunity to express my appreciation and gratitude to my many friends, colleagues, and mentors for providing invaluable learning opportunities. A real debt is owed to the 50,000-plus professional men and women of the organizations that I have taught and worked with through my 45-plus years in the oil and gas industry and made a reality the ideas in this volume. I have been privileged to work with the best-of-the-best companies of the world, and this book is dedicated to them for their vision and perseverance.

Although I cannot mention everyone who has helped me along the way, I would like to say thank you to my colleagues and friends.

A special thanks to **Ken E. Arnold PE**, the senior coauthor of the first two volumes of *Surface Production Operations* series. Much of the information contained in volumes 3, 4 and 5 was originally contained in *Volume 1 and 2*. Ken is also responsible for encouraging me to teach two-day courses for the Society of Petroleum Engineers on Surface Production Operations, which, in turn, lead to the first two volumes of the series. Without Ken's expert contributions, the initial *Volume 1 and 2* and subsequent *Volume's* would not have been possible.

I want to thank my international business partners and very close friends—**Jamin Djuang**, Managing Director of PT Loka Datamas Indah (LDI Training); **Chang Choon Kiang "CK,"** Managing Director of Professional Training Southeast Asia (PTSEA); and **Clement "Clem" Nwogbo**, Managing Director of Resource Plus. Together, for over 30 years, we have provided training to over 50,000 professionals in every oil and gas producing region of the world.

I also want to thank my friends and colleagues Mike Zimmerman, Jeff Post, Robert Neil, and David Rodrigues of Chevron Angola (CABGOC); Al Ducote, Greg Abdelnor, and Sam Sowunmi of Chevron Nigeria Limited; Hidayat Maruta and Bambang Indrawan with Chevron Pacific Indonesia (CALTEX); Khun Aujchara and Bundit Pattanasak of PTTEP; Jorge Velez, Brian Devenish, and Suharyadi Suharjan of MAXUS; Sugeng Suhantono of Medco Indonesia; Teo Weband Mochammad Zainal-Abidin of Total Indonesie.

Last, but certainly not least, I want to thank my good friend and colleague, **Heri Wibowo**, who not only prepared all illustrations, tables, photographs, and figures but also took my poorly typed raw manuscript and transformed it into a format suitable to submit to the publisher. Heri also took loosely drawn sketches or concepts, often sketched on the back on a napkin and transformed them into a very detailed drawing illustrating the exact points I was trying to get across in the text. Heri was responsible for editing and pulling everything all together at the end. Without Heri's efforts this volume would not exist.

I am also indebted to a number of pump, compressor, and driver manufacturers; oil and gas companies; and professional associations for granting permission to use technical data, information, or artwork. I thanked them throughout the text when using their material but those deserving special thanks are as follows:

- Chevron USA
- Hydraulic Institute
- American Petroleum Institute
- American Society of Mechanical Engineers
- Ingersoll-Dresser Pumps
- Sulzer Pumps
- Pump Handbook, Karassik, Caterpillar and Durametallic
- Worthington Pumps
- Flowserve Corporation
- Duramettalic Corporation
- Rexnord
- Steelflex Couplings
- Lovejoy Couplings
- Peerless Pump Company
- Goulds Pumps
- Sundstrand Fluid Handling Company
- Marcel Dekker, Inc
- Metering Pump Handbook by McCade, Lanckton, and Dwyer
- GASO Pumps
- Continental EMSCO
- FWI
- Gardner Denver
- Moyno Industrial Pumps
- IMO Industries
- Wilden Pumps and Engineering Company
- Watson-Marlow/Bredel Pumps
- GPSA Engineering Data Book
- Compressors: Selection and Sizing by Royce Brown
- Demag Delaval
- Elliott Company
- Cooper Cameron Corporation
- Ariel Corporation
- Cooper Industries Energy Group
- Southwest Research Institute (SWRI)
- Siemens Energy and Automation
- Electrical Machinery-Dresser Rand
- Caterpiller
- Power Transmission Design Handbook
- National Electrical Manufacturers Association (NEMA)
- Internal Combustion Engines and Air Pollution by Edward F. Obert
- Woodward Governors
- Westinghouse Electric Corporation

- Cooper-Rolls
- General Electric Company
- ABB
- Stewart Stevenson Services

My sincere apologies to anyone I may have overlooked.

Maurice I Stewart Jr. PE, LLC
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Disclaimer

Overview of pumps, compressors, and drivers

1

1.1 Overview

After wells are successfully drilled and completed, the fluid produced must be transported to a facility where it is separated into oil, water, and gas; conditioned to meet process input requirements; treated to remove impurities such as H_2S, CO_2, H_2O, and solids; and refined or stored for eventual sales. Fig. 1.1 is a simplified block diagram that illustrates the basic "wellhead to sales" concept. The diagram begins with the wellhead choke which is used to control the rate of flow from each well. The fluid from the well travels through a flowline to the production facility where the fluid is separated, conditioned, treated, processed, measured, refined, or stored.

Volume 1: Design of Oil-Handling Systems and Facilities and *Volume 2: Design of Gas-Handling Systems and Facilities* of the *Surface Production Operations* series present the basic concepts and techniques necessary to select, specify, size, operate, and troubleshoot oil, water and gas handling, conditioning, and processing facilities. *Volume 3: Facility Piping and Pipeline Systems*; *Volume 4: Pumps, Compressors, and Drivers*; *Volume 5: Process Equipment-Pressure Vessels, Heat Exchangers and Storage Tanks*; and *Volume 6: Instrumentation, Process Control, and Safety Systems* of the *Surface Production Operations* series build upon the information that is presented in *Volumes 1 and 2*.

1.1.1 Pumps

Pumps serve many purposes in production facilities. The largest pumps are usually shipping or sales pumps which increase the pressure of oil or condensate so that it can flow into a sales pipeline or be loaded into tankers, barges, railroad cars, or trucks. Other large pumps are used with water injection systems for disposing of produced water or for waterflooding. Smaller pumps are used to pump liquids from low to higher pressure vessels, to pump liquids from tanks at a low elevation to tanks at a higher elevation, or to transfer liquids for further processing.

A facility's utility system often has many pumps, which may be used for firewater wash down and utility water, heat medium, fuel oil or diesel, and hydraulic systems.

The project/facility engineer must be able to select the proper pump for each application, determine horsepower requirements, design the piping system associated with the pump, and specify materials and details of construction for bearings, seals, and so on. On standard applications the project/facility engineer may allow the vendor to specify materials and construction details for the specified service conditions. Even then, the project/facility engineer should be familiar with different alternatives so that he or she can better evaluate proposals and alternate proposals of vendors.

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00001-6

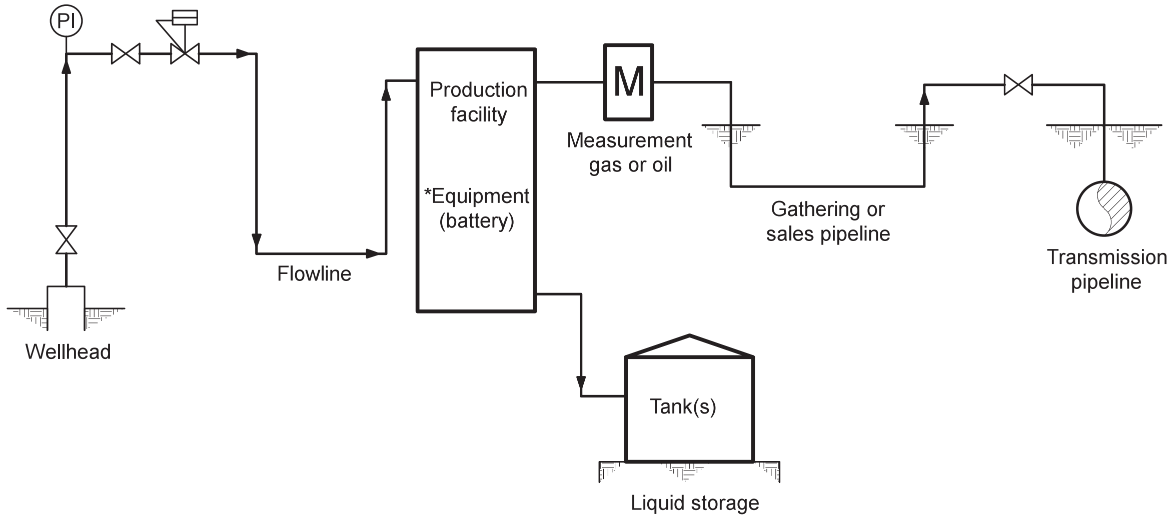


Fig. 1.1 Block diagram of "wellhead to sales" concept. (*Interconnection piping between production equipment-separators: treaters, heaters, and tankage.)

1.1.2 Compressors

Compressors are used whenever it is necessary to flow gas from a low-pressure system to a high-pressure system. Flash gas from low-pressure vessels used for multistage stabilization of liquids, oil treating, water treating, and so on, often exists at too low a pressure to flow into the gas sales pipeline. Sometimes this gas is used as fuel and the remainder is flared or vented. In many instances, it is economically attractive to compress this gas to a high-pressure system so it can be sold.

Compression may also be required due to environmental reasons. Flash gas, which might otherwise be flared, may be compressed for sales, or gas produced with oil (associated gas) may be compressed for reinjection to avoid flaring or to help maintain reservoir pressure.

In some marginal gas fields and in many larger gas fields that experience a decline in flowing pressure with time, it may be economical to allow the wells to flow at surface pressures below that required for gas sales. In such cases, a booster compressor (one in which the ratio of discharge to suction pressure is low) may be installed. Booster compressors are also used on long pipelines to restore pressure lost to friction.

The use of large compressors is probably more prevalent in oil field facilities than in gas field facilities. Oil wells often require low flowing surface pressures, and the gas that flashes off the oil in the separator must be compressed. Often, natural gas is injected into the tubing of the well to lighten the column of liquid and reduce hydrostatic backpressure. This "gas lift" gas is produced with the well fluids at low pressure. Compressors are used so the gas lift gas can be recirculated and injected back into the well at high pressure. Gas lift compressors are characterized by both high overall ratios of discharge to suction pressure ("pressure ratios") and relatively high throughputs.

As with pumps, the project/facility engineer must select the type of compressor to use for each application, determine power requirements, design the piping system associated with the compressor, and specify materials and details of construction for bearings, cylinders, and so on. On standard applications, the project/facility engineer may allow the vendor to specify materials and construction details for the specified service conditions. However, even in these instances the project/facility engineer should be familiar with different alternatives so that he can better evaluate vendors' proposals and alternative proposals.

1.1.3 Drivers

Drivers must be capable of developing the required horsepower and torque to drive mechanical equipment such as pumps, compressors, and fans used throughout the production facility. The most common drivers used are electric motors, internal combustion engines, combustion gas turbines, and steam turbines.

Three-phase alternating-current (AC) induction and synchronous motors are used primarily with reciprocating and centrifugal pumps. Motor types, performance characteristics, enclosures, and maintenance considerations should be reviewed when selecting an electric motor.

Two- and four-stroke internal combustion engines are typically used with reciprocating pumps and compressors. Several factors such as support systems, emissions control, arrangements, natural aspiration, supercharging, and turbocharging should be reviewed when selecting an internal combustion engine.

Combustion gas turbines (CGTs) are almost always used with centrifugal compressors. Types of turbines as well as their performance characteristics, emissions, and auxiliary systems should be reviewed when selecting a CGT.

1.2 Scope and application

Volume 4 of the *Surface Production Operations* series is directed both to entry-level personnel and experienced personnel who are looking for a reference on pumps, compressors, and drivers.

It covers approximately 90% of the most commonly used pumps, compressors, and drivers, focusing on four areas: fundamentals, specifying and purchasing, troubleshooting, and modifying. It gives little attention to large, complex or to specialty pumps, compressors, and drivers, although such machines are important in their individual applications. *This text should not be used as a substitute for sound engineering judgment*.

This text contains guidelines that can be used as is or modified for local or geographical preferences, priorities, or experiences.

The intent of this text is to provide practical, useful information based on standard industry practice and the author's personal experience.

1.3 Organization

Volume 4 is divided into three parts:

- Part 1: Pumps (Chapters 2–6)
- Part 2: Compressors (Chapters 7–11)
- Part 3: Drivers (Chapter 12)

Each of these parts is interrelated and contains:

- Information on principles of operation
- Examples of typical equipment
- Criteria for equipment selection
- Hydraulic and thermodynamic system design
- Installation, start-up, and troubleshooting

A detailed summary of each of the three parts follows:

Part 1: Pumps

- *Chapter 2* provides an overview of all categories of pumps and is a guide which directs the reader to more detailed information on specific pump categories in later chapters. Engineering principles, system hydraulic design, and basic selection criteria are included in this chapter because it is the first step in work involving all pump categories. This is a critical step in pump selection and often troubleshooting as well.

- *Chapter 3* describes how centrifugal pumps work, lists their limitations, and explains how to select the right centrifugal pump for a given application. Process piping considerations, changing a centrifugal pump's performance, regulating flow rate, construction details, start-up, operations and maintenance considerations are included in this chapter.
- *Chapter 4* discusses engineering principles, pump types, performance considerations, application and selection criteria, and describes commonly used reciprocating pumps. Pump construction details, materials of construction, installation considerations, start-up, operations and maintenance considerations are also included.
- *Chapter 5* discusses engineering principles, pump types, mechanical features, selection guidelines, service considerations, configurations, and installation considerations for rotary pumps.
- *Chapter 6* briefly describes several pump types, including artificial lift pumps (electrical submersible, sucker rod, and hydraulic turbine pumps), jet pumps, air diaphragm pumps, regenerative pumps, and slurry pumps.

Part 2: Compressors

- *Chapter 7* is an overview of dynamic (centrifugal and axial) and positive displacement compressors and an overview that directs readers to more detailed information on specific types of compressors found in subsequent chapters of the book. This chapter also provides background information on the principles of compression, including a discussion of thermodynamics. It is not essential that the reader read this entire chapter, but they may wish to use it as reference material when selecting a compressor. To confirm one's initial selection of a new compressor, one may find that unique site conditions or economic factors pose serious problems for the original choice of machine. Thus it may be necessary to evaluate two or more categories, or combination of categories, of machines for a specific application.
- *Chapter 8* discusses engineering principles, types of machines and configurations, materials of construction, and performance characteristics of dynamic compressors. It contains sufficient information, when used in conjunction with industry codes, standards, and recommended practices, to understand how to specify and apply centrifugal compressors including auxiliaries and support systems. The discussion is primarily aimed at heavy-duty multistage units, but the information can be applied to smaller and less severe duty compressors as well.
- *Chapter 9* discusses engineering principles, types of reciprocating compressors, configurations, and performance characteristics. It contains sufficient information for understanding how to specify and apply reciprocating compressors including auxiliaries and support systems.
- *Chapter 10* discusses application of compression theory and practical solutions. Many example problems are included to help the reader develop Performance Maps so they can better understand how to select new and evaluate existing compressor applications. Topics include, but are not limited to, effects of compressor performance due to a reduction in capacity, effect on compressor discharge pressure, effect on compressor load, effect of adding clearance, determining the range of a compressor, effects of speed, designing a compressor to meet gas sales line pressure, and determining safety set points. Compressor suction and discharge line sizing and sizing suction and discharge pulsation bottles are also included.
- *Chapter 11* discusses engineering principles, compressor types (screw, liquid ring, sliding vane, straight load), mechanical features, and system considerations for rotary compressors.

Part 3: Drivers

- *Chapter 12* reviews the fundamentals and characteristics of commonly used drivers. It provides an overview of the various categories of drivers and discusses the choice of these drivers. This chapter will enable the reader to select the appropriate category of driver for an intended service.

References

Stewart, M., Arnold, K., 2007. Surface Production Operations: Volume 1. Design of Oil-Handling Systems and Facilities, 3rd ed. Elsevier, Amsterdam.

Stewart, M., 2014. Surface Production Operations: Volume 2. Design of Gas Handling Systems and Facilities, 3rd ed. Gulf Professional Publishing, Waltham, MA.

Stewart, M., 2015. Surface Production Operations: Volume 3. Facility Piping and Pipeline Systems. Gulf Professional Publishing, Waltham, MA.

Pump fundamentals

2

Nomenclature

a	piston rod cross-sectional area, mm^2 (in^2)
A	plunger or piston cross-sectional area, mm^2 (in^2)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
d	displacement per pumping chamber, m^3 (gal)
D	pump displacement, m^3/h (gpm)
d'	pump piston or plunger diameter, mm (in.)
d_w	diameter of wrist pin, mm (in.)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/s
g	acceleration due to gravity, 9.81 m/s^2 (32.2 ft/s^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_f	pipe friction loss, m (ft)
HHP	hydraulic horsepower, kW (hp)
H_p	total head required for pump, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
K	a factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in.)
m	number of pistons, plungers, or diaphragms
n	stroke rate or crank revolutions per min, rps (rpm)
NPSH	net positive suction head, m (ft)
$NPSH_A$	net positive suction head available, m (ft)
$NPSH_R$	net positive suction head required, m (ft)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
Q	flow rate, m^3/h (ft^3/s)
q	flow rate, m^3/h (gpm)
Q'	flow rate, m^3/h (BPD)
S	stress, kPa (psi)
s	stroke length, mm (in.)
S'	valve slip, percent
(SG)	specific gravity of liquid relative to water

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00002-8

V	velocity in suction line, m/s (ft/s)
Vol	volume of surge tank, m^3 (ft^3)
$(\mathbf{Vol})_g$	required gas volume, m^3 (ft^3)
Z	elevation above or below pump centerline datum, m (ft)
ΔP	static pressure, kPa (psi)
ρ	density of fluid, kg/m^3 (lb/ft^3)

2.1 Engineering principles

2.1.1 Background

Pumps serve many purposes in production facilities. The largest pumps are usually shipping or pipeline pumps which increase the pressure of oil or condensate so that it can flow into a pipeline or be loaded into tankers, barges, railroad cars, or trucks. Large pumps are also used with water injection systems for disposing produced water or for waterflooding. Smaller pumps are used to pump liquids from low to higher pressure vessels, to pump liquids from tanks at a low elevation to tanks at a higher elevation, or to transfer liquids for further processing. A production facility's utility system often has many pumps, which may be used for firewater wash down and utility water, heat medium, fuel oil or diesel, and hydraulic systems.

The engineer must be able to select the proper pump for each application, determine horsepower requirements, design the piping system associated with the pump, and specify materials and details of construction for bearings, seals, and so on. On standard applications the engineer may allow the vendor to specify materials and construction details for the specified service conditions. Even then, the engineer should be familiar with different alternatives so that he or she can better evaluate proposals and alternative proposals of vendors.

There are a number of factors that should be considered when selecting a pump for a specific application. The primary goal is the same, that is, providing a pump that maximizes company profits (low cost) while providing safe, reliable (trouble free) equipment that satisfies operating requirements and meets local environmental constraints. Selecting the right pump is difficult due to the number of different kinds of pumps available.

Meeting the primary goal involves optimizing the following primary factors:

- Minimizing the initial pump cost
- Meeting safety and environmental concerns
- Reducing installation and commissioning expenses
- Reducing maintenance expenses
- Maximizing reliability
- Starting up on time
- Maintaining maximum production

Sound engineering judgment is important when deciding which of the listed factors are the most important.

2.1.2 Types of fluids

Pumps are used to move fluids. Fluids include liquids, dissolved gases, and solids. Gases include dissolved air and hydrocarbon vapors. Solids include sand, clay, and scale. Most common types of fluids pumped in upstream operations are water, crude oil, condensate, lube oils, glycols, and amines.

Each fluid has different physical properties. Physical properties that must be taken into consideration when sizing and selecting a pump are as follows:

- Suction temperature
- Specific gravity
- Viscosity
- Vapor pressure
- Solids content (if predictable)
- Lubricity (slipperiness)

2.1.3 Pump classification

A pump can be defined as "as mechanical device that adds energy to a liquid for the purpose of increasing its flow rate and static pressure." Pumps are divided into two major categories: positive displacement and kinetic energy. These two categories are further divided into numerous subdivisions.

2.1.3.1 Positive displacement pumps

Positive displacement pumps add energy to a fluid by applying force to the liquid with a mechanical device such as a piston or plunger. A positive displacement pump decreases the volume containing the liquid until the resulting liquid pressure equals the pressure in the discharge system. That is, the liquid is compressed mechanically, causing a direct rise in potential energy. Most positive displacement pumps are reciprocating pumps in which linear motion of a piston or plunger in a cylinder causes the displacement. In rotary pumps, another common positive displacement pump, a circular motion causes the displacement. There are several manufacturers of positive displacement pumps which are often found in high-pressure services. As shown in Fig. 2.1, positive displacement pumps are classified as either:

- *Reciprocating*—use pistons, plungers or diaphragms to displace fluid.
- *Rotary*—operate through the mating action of gears, lobes, or screw-type shafts.

2.1.3.2 Kinetic energy (dynamic) pumps

Kinetic energy (dynamic) (energy associated with motion) is added to a liquid to increase its velocity and, indirectly, its pressure. Kinetic energy pumps operate by drawing liquid into the center of eye of a rapidly rotating impeller. Radial vanes on the impeller then throw the liquid outward toward the impeller rim. As liquid is thrown outward, more liquid is drawn into the resulting low-pressure area in the eye through a suction port in the pump casing. As liquid leaves the impeller, it comes into contact with the pump casing or volute. The casing is shaped to direct the liquid

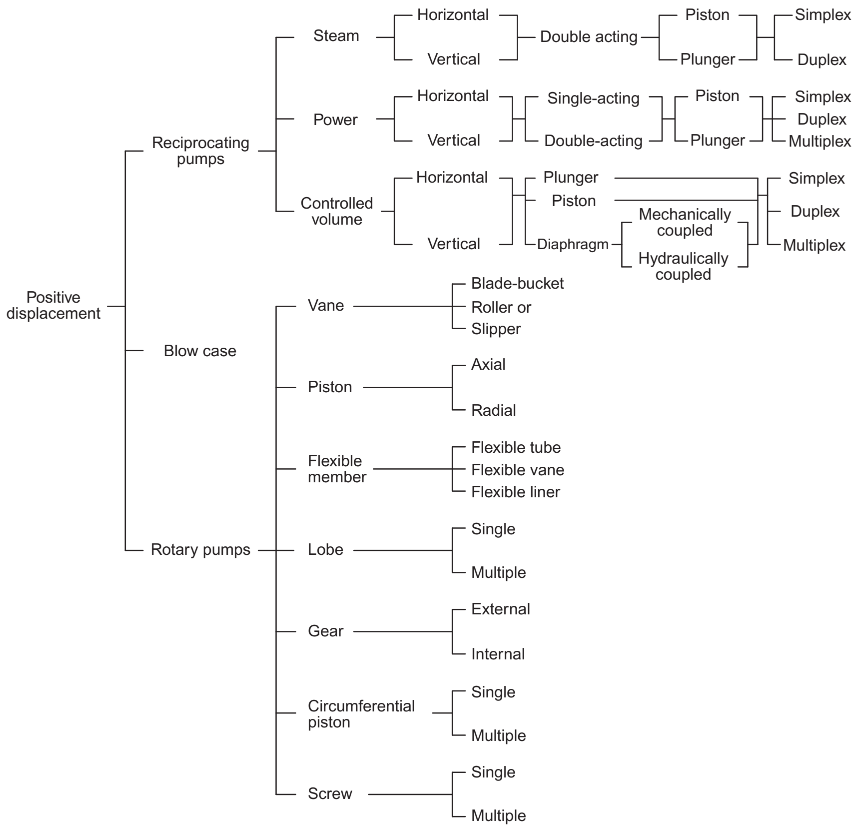


Fig. 2.1 Positive displacement pump classifications.
Courtesy of Hydraulic Institute.

toward a discharge port. The casing slows the liquid and converts some of its velocity into pressure. As shown in Fig. 2.2, kinetic energy pumps are classified as either:

- *Centrifugal*: Includes radial, mixed, and axial flow designs. They account for over 80% of pumps used in production operations because they exhibit uniform flow, free of low frequency pulsations and are not subject to mechanical problems.
- *Regenerative turbine*: Often times the regenerative turbine pumps are considered to be positive displacement type pumps. They are available in single stage and multiple stages, for low flow/high-pressure applications.
- *Special effects pumps*: Reversible centrifugal pumps and rotating casing (pilot) pumps fall in this category.

Fig. 2.3 illustrates pumps that are commonly encountered in production operations. Pumps are commonly rated by

- Flow rate
- Differential pressure/head

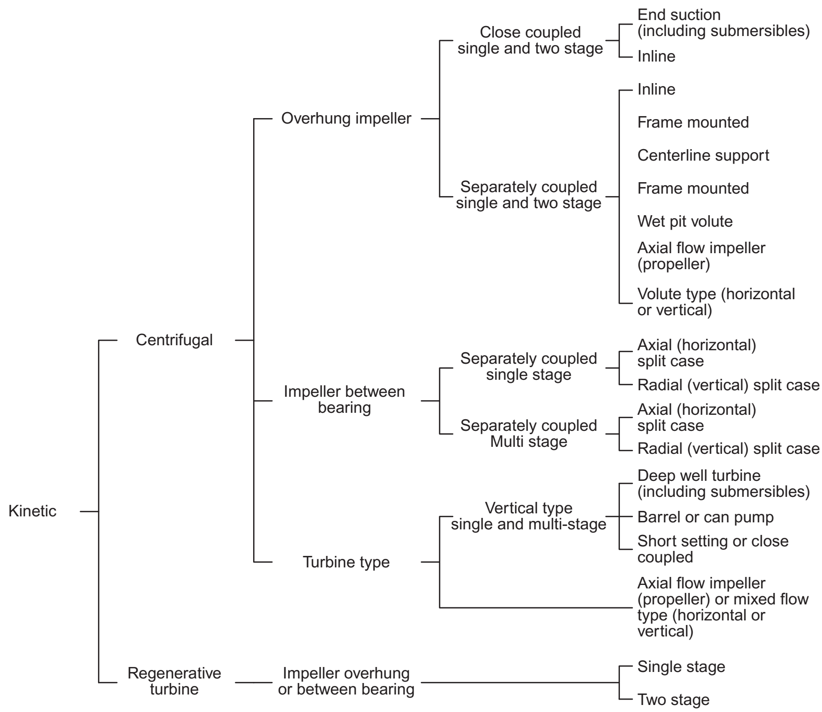


Fig. 2.2 Kinetic energy pump classifications.

- Horsepower
- Suction pressure

2.1.4 Positive displacement pumps

Positive displacement pumps, although not as common as centrifugal pumps, are also widely used in production operations. The three major positive displacement subdivisions are reciprocating, rotary, and controlled volume (metering). The reciprocating pumps use pistons, plungers, and diaphragms to displace the fluid while rotary pumps use vanes, gears, screws, lobes, and so on.

2.1.5 Centrifugal pumps

Centrifugal pumps are by far the most widely used pumps and have been estimated to make up 80% of all pumps used in production operations. They are widely accepted because they combine a relatively low cost with high reliability, compact size, non-pulsating flow, and easy maintenance. They are also widely available, cover a broad flow/pressure application ranges, and can operate over a wide flow range.

Centrifugal pumps are usually purchased to meet one of two levels of duty.

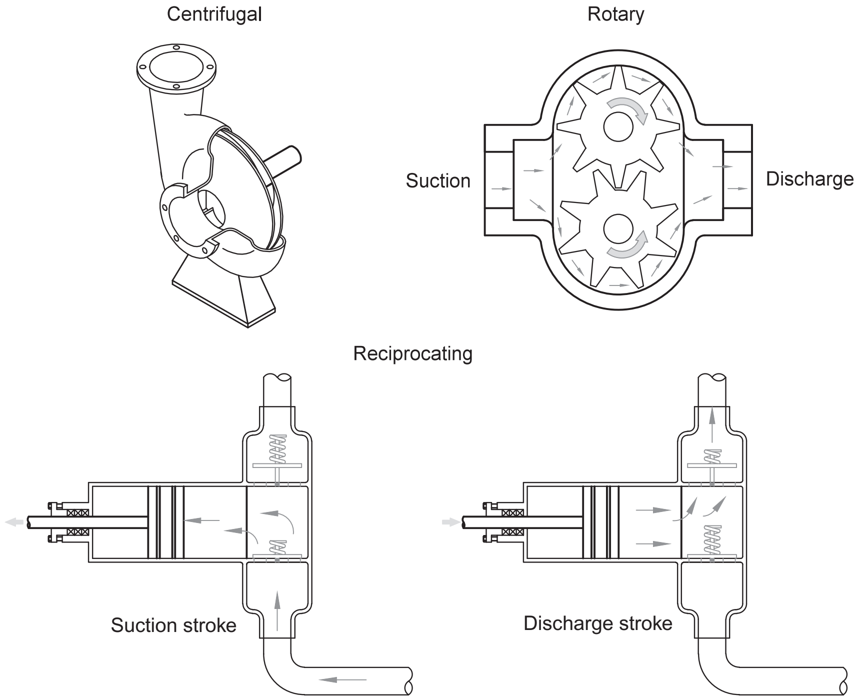


Fig. 2.3 Pumps commonly encountered in production operations.

Medium Duty pumps are used in general, noncritical, nonhazardous service. These pumps are usually built to American National Standards Institute (ANSI) Standard B73.1 "Horizontal End-Suction" or ANSI Standard B73.2 "Vertical In-Line" specifications.

Heavy-duty pumps are used in critical, hazardous, or "heavy-duty" service including chemicals, refining, and other producing services. These pumps are usually built to American Petroleum Institute (API) Standard API 610 "Centrifugal Pumps for General Refinery Services" specification. Centrifugal pumps used as Fire Pumps are most often heavy-duty pumps.

2.1.6 Miscellaneous pumps

Several other types of pumps have been designed to accommodate specific needs. These pumps include

- Air diaphragm pumps
- Regenerative turbine pumps
- Jet pumps
- Slurry pumps

Sucker rod and electric submersible pumps (ESP) pumps are both very common and important in production operations. These pumps are discussed in Chapter 6. Progressive cavity pumps are covered in the Rotary Pump discussion in Chapter 5.

2.1.7 Pumping system design

The process of designing any pumping service involves three major activities:

- Process design
- Mechanical design
- Vendor specifications

2.1.7.1 Process design

- *Obtain flow rate*. Define any flow variation that should be included in design, such as start-up conditions, future expansion, and maximum flow. Select a value for rated flow rate. Convert the required rated flow rate at pumping conditions into US gallons per minute.
- *Determine the liquid properties critical to pump design*. Properties of importance include specific gravity, temperature, viscosity, pour point, and so on. Values are required at pumping conditions and, in some cases, at ambient conditions.
- *Calculate available suction conditions such as rated suction pressure, maximum suction pressure, and* $NPSH_A$.
- *Determine the effect of the selected control system on pump performance requirements (i.e., constant bypass, backpressure control valve, or variable speed).*
- *Calculate the minimum discharge pressure requirement of the pump*.
- *Calculate the total dynamic head (TDH) at the specific gravity corresponding to rated pumping temperature*.

2.1.7.2 Mechanical design

- Determine the design pressure and temperature required for the pump and its associated piping.
- Select pump type and driver type.
- Select materials of construction.
- Determine sparing (backup) requirements and the need for parallel operation.
- Determine other installation requirements, such as control system details, auto-start of standby pump, and so on.
- Select shaft seal type and determine the requirements for an external flushing or sealing system.
- Estimate utility requirements.
- Document the design: calculations, studies, design specification text, utility requirement estimate summary, and so on.

The following factors listed have a significant influence on the mechanical design and investment cost.

- Number of pumps installed (in parallel)
- Type of casing material
- Net positive suction head available ($NPSH_A$)
- Total dynamic head (TDH) requirement

- Flow rate per pump in gallons per minute, gpm (m^3/h)
- Design pressure and temperature
- Type of pump selected
- Corrosiveness/toxicity of the fluids
- Solids content of the fluids
- Power requirements
- Type of driver selected

2.1.7.3 Vendor specifications

Factors that have the greatest influence on the selection of the most cost-effective pump type include

- Capacity, gpm (m^3/h)
- Total dynamic head (TDH)
- Maintenance
- Viscosity
- Capacity control

Within the general type selections, a particular construction style is most influenced by

- Discharge pressure
- $NPSH_A$
- Fluid temperature
- Space and weight limitations

2.2 Hydraulic principles

Hydraulics deals with the mechanical properties of water and other liquids and the application of these properties as they apply to engineering. Hydraulics is divided into the following two areas:

- Hydrostatics (fluids at rest)
- Hydrodynamics (fluids in motion)

2.2.1 Hydrostatics

2.2.1.1 General considerations

A liquid has a definite volume when compared to a gas, which tends to expand indefinitely. When unconfined, a liquid seeks the lowest possible level. Due to its fluidity, a liquid will conform to the shape of its container.

2.2.1.2 Pressure

The pressure existing at any point in a liquid body at rest is caused by the atmospheric pressure exerted on the surface, plus the weight of the liquid above that point. This pressure is equal in all directions.

2.2.1.3 Temperature

For most liquids, an increase in temperature decreases viscosity, decreases specific gravity, and increases volume. Temperature affects:

- Type of pump construction
- Material selection
- Corrosive properties of a fluid
- Pumps flange pressure ratings

2.2.1.4 Head

The term "head" is used to represent the vertical height of a static column of liquid; it corresponds to the energy contained in the liquid per unit mass. Head can also be considered as the amount of work necessary to move a liquid from its original position to the required delivery position. In this case, the head includes the extra work necessary to overcome the resistance (friction) to flow. In general, a liquid at any point may have the following three types of head:

- *Static pressure head* represents the energy contained in the liquid due to its pressure.
- *Potential head* represents the energy contained in the liquid due to its position measured by the vertical height above some plane of reference.
- *Velocity head* represents the kinetic energy contained in the liquid due to its velocity.

The static discharge head of a pump is the height at which a pump can raise a fluid. If one disconnects the discharge pipe on a pump and extends it vertically the discharge head is the height at which the pump raises the fluid. As shown in Fig. 2.4, the more pressure the pump delivers the higher the head will be. Let us assume the head in Fig. 2.4 is 60 ft (18 m). As shown in Fig. 2.5, as the level in the suction tank (static

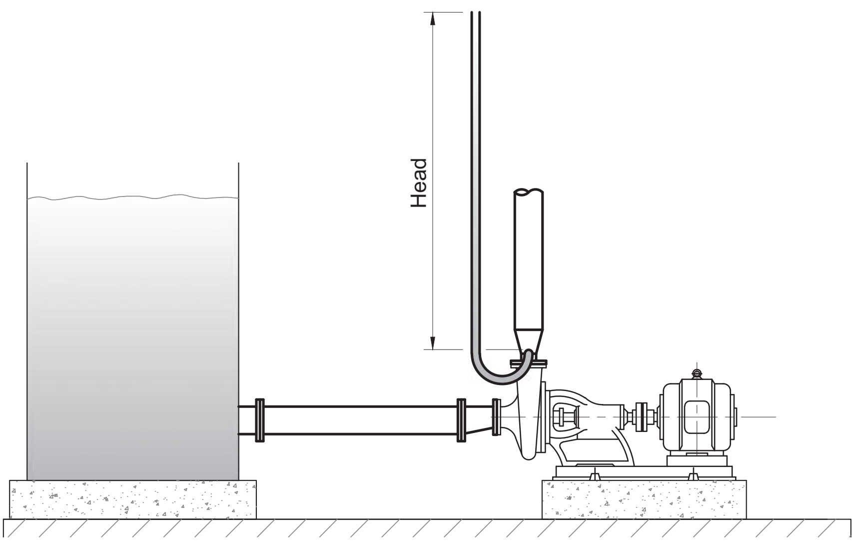


Fig. 2.4 The meaning of head.

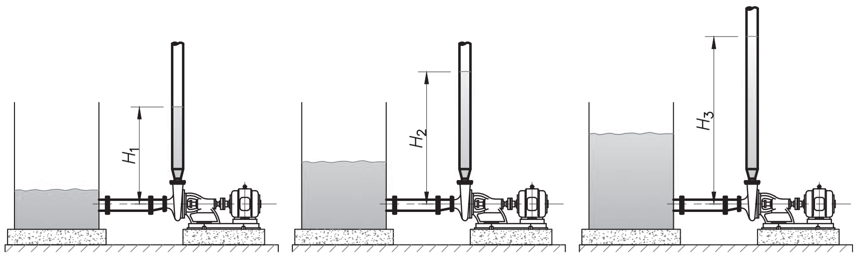


Fig. 2.5 How discharge head increases as the level in the suction tank increases.

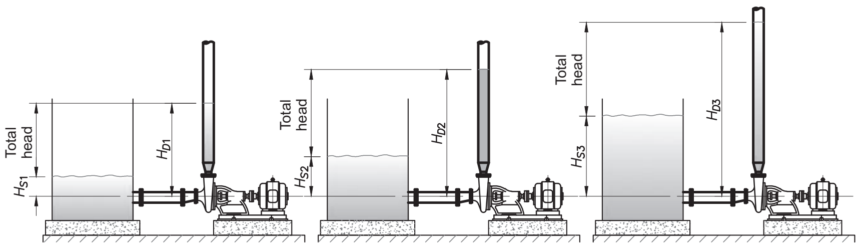


Fig. 2.6 The effects of increasing the suction head on discharge head and total head.

suction head) increases ($H_3 > H_2 > H_1$) the static discharge head increases and vice versa.

In order for a pump manufacturer to show the head capability for a specific pump, the suction head, H_S, is subtracted from the discharge head, H_D, to produce the total head, H_T, available. Fig. 2.6 shows the effects of increasing the suction head on discharge head and total head.

As shown in Fig. 2.7, if a pump is lifting a fluid from below the pump (suction lift) centerline the pump will still produce the same total head (H_T), but the discharge head (H_D) will be reduced.

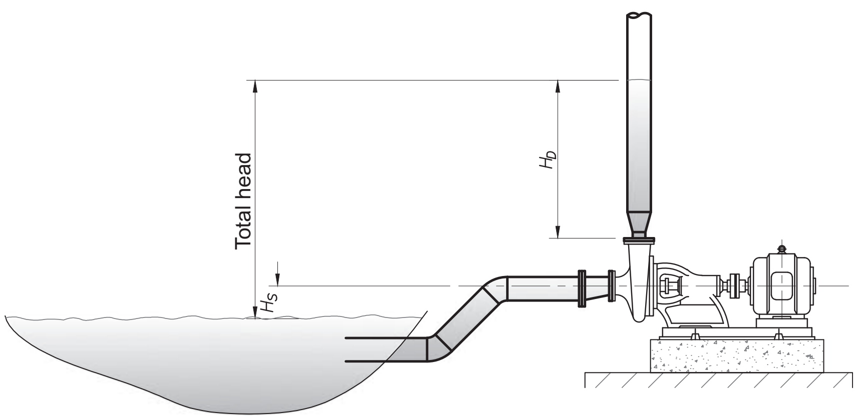


Fig. 2.7 Effect of low level on the pump suction (suction lift).

2.2.1.4.1 Centrifugal pumps

Operating pressure is expressed in feet (meters) of the fluid that is being pumped. Suction and discharge pressures are expressed as suction head and discharge head. Pressures are expressed in feet (meters) of head because it is more important to know how much a pump can raise the fluid it is pumping, rather than the amount of pressure the pump is adding to the fluid.

2.2.1.4.2 Positive displacement pumps

Operating pressures are almost always expressed in terms of pressure (psi)(bar).

The relationship of head to pressure is expressed as follows:

Field units

$$\text{Head (ft)} = \frac{\text{Pressure (psi)} \times 2.31}{\text{Specific Gravity of the Fluid}} \tag{2.1a}$$

SI units

$$\text{Head (m)} = \frac{\text{Pressure (kPa)} \times (0.102)}{\text{Specific Gravity of the Fluid}} \tag{2.1b}$$

Conversely,

Field units

$$\text{Pressure (psi)} = (SG)(h) \tag{2.2a}$$

SI units

$$\text{Pressure (kPa)} = (SG)(h) \tag{2.2b}$$

where

SG = density of the fluid
h = height of the fluid column above a reference point, ft (m)
Pressure = pressure, psi (kPa)

The head of fluid is not related to the area occupied by the fluid. The types of head are illustrated in Fig. 2.8. Water is the reference fluid. Density of fluids other than water can be determined from the following formula.

$$SG = \frac{\rho_f}{\rho_w} \tag{2.3}$$

where

SG = specific gravity of fluid
ρ_f = density of fluid, Ibs/ft^3, (kg/m^3)
ρ_w = density of water, Ibs/ft^3, (kg/m^3)
= 62.34 Ibs/ft^3 at 60°F
= 1000 kg/m^3 at 4°C

Since there are 144 in^2/ft^2, the following relationships exist.

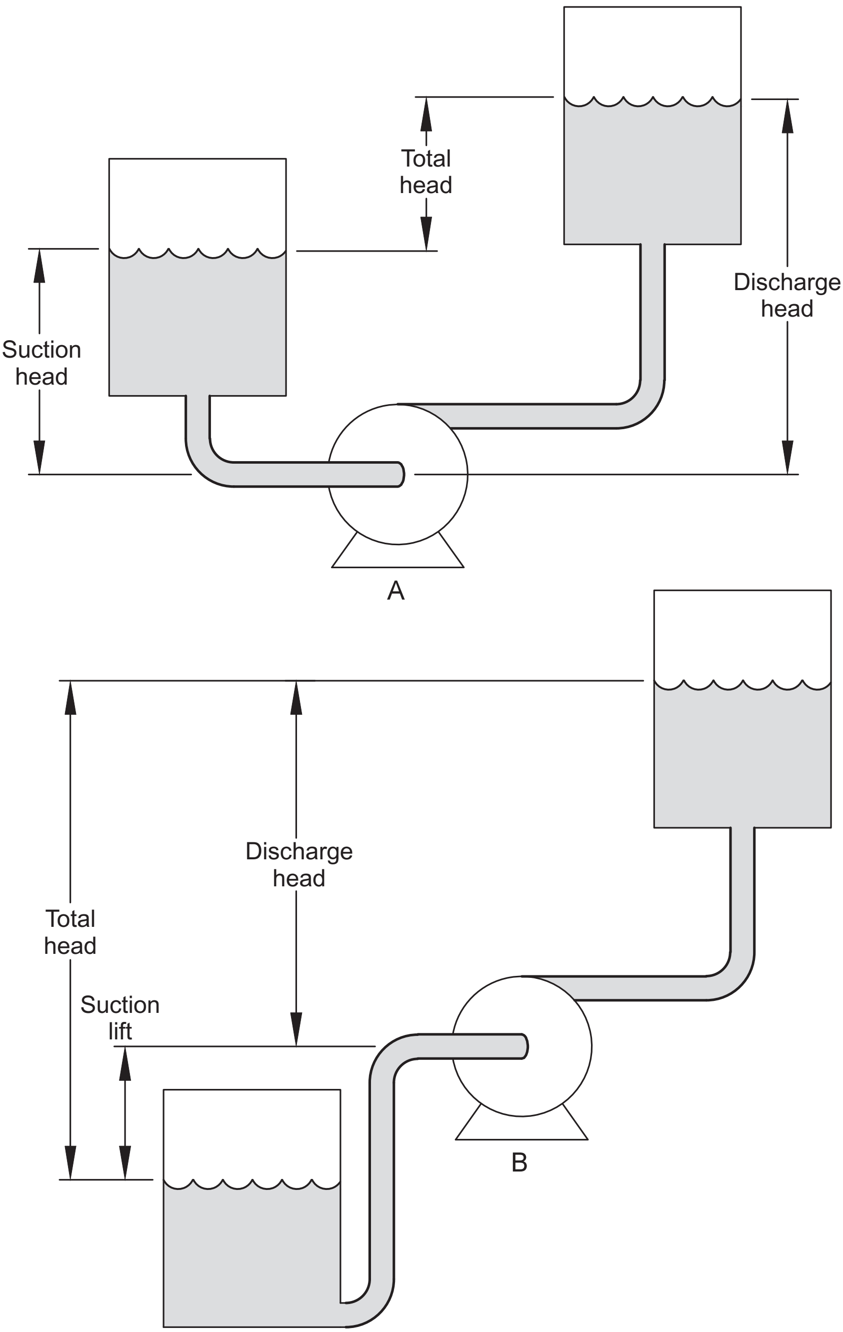


Fig. 2.8 Types of head. (A) Suction head. (B) Suction lift.

$$\text{Pressure (psig)} = \left(\frac{62.34\,\text{lbs}}{ff^3}\right) h(SG) \left(\frac{\text{ft}^2}{144\,\text{in}^2}\right) \tag{2.4}$$

$$= \frac{(SG)}{2.31} h \tag{2.5}$$

or

$$= 0.433h(SG) \tag{2.6}$$

Fig. 2.9 illustrates the relationship between head and pressure. As an example, 10 ft (3 m) of clean water (SG = 1.0) in a tank exerts a static pressure of 4.33 psi (29.41) kPa), while 100 ft (30 m) exerts 43.3 psi (294.11 kPa). In a kinetic energy type pump application, we would state 10 ft (3 m), while for the positive displacement pump, it would be correct to state 4.33 psi (29.41 kPa).

It is important to realize that although the heads of different fluids are the same, their pressures are different because of the difference in specific gravities. As an example, assume three 100-ft (30-m) tall tanks completely filled with gasoline, water,

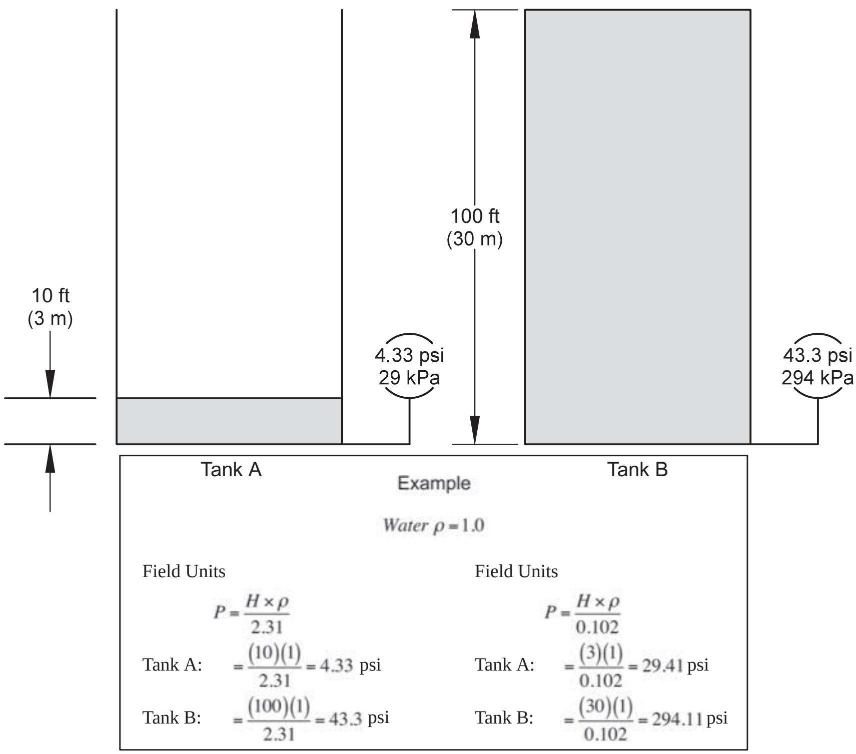


Fig. 2.9 Head vs pressure.

and molasses. The pressure measured at the bottom of each tank is different due to the differences of specific gravities of gasoline (0.75), water (1.0), and molasses (1.45).

2.2.1.5 Static suction head vs static suction lift

Static suction head is the vertical distance between a fluid level and a datum line when the supply is located above the datum (Fig. 2.10).

Static suction lift is the vertical distance between a fluid level and the datum line when the datum line is located above the supply (Fig. 2.11).

Datum line is the centerline of the pump inlet connection or the horizontal centerline of the first-stage impeller on a vertical pump.

Theoretical lift is the pressure to lift a fluid and comes from atmosphere pressure. At sea level, atmospheric pressure is 14.7 psia (100 kPa) (refer to Fig. 2.12). Theoretically, atmospheric pressure can lift a column of water 34 ft (10.2 m) if there are no air leakages (full vacuum in the suction line) into the system (refer to Fig. 2.13). This vertical distance (34 ft) (10.2 m) that a pump lifts water below the level of the pump is called theoretical lift.

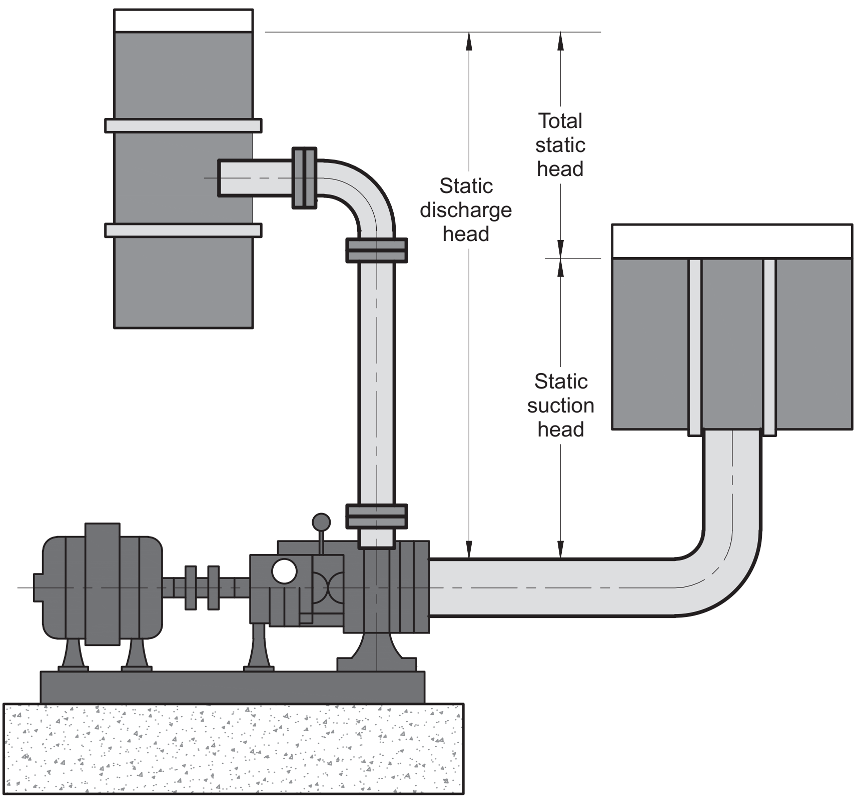


Fig. 2.10 Examples of static suction head.

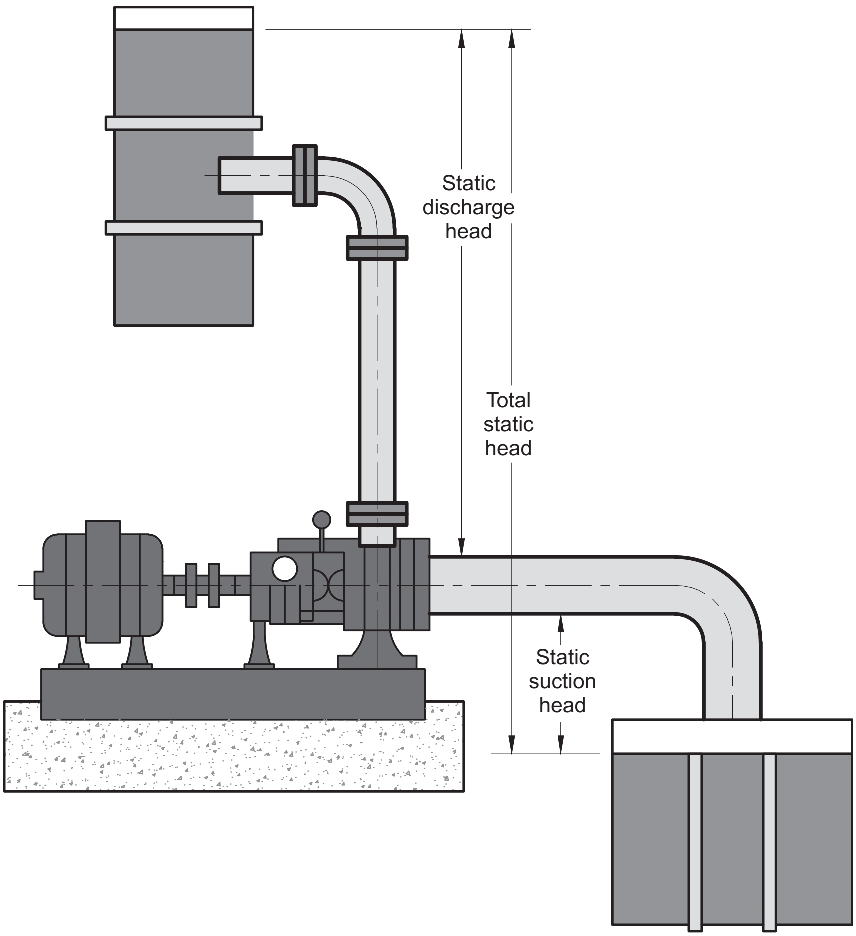


Fig. 2.11 Example of static suction lift.

Under perfect conditions.
Field units

$$\text{Pressure} = 0.433\,h\,(SG)$$

$$\text{Pressure} = 0.433h(SG) \tag{2.7}$$

$$14.7 = 0.433h(1.0)$$

Rearranging and solving for h

$$h = 34\,\text{ft}$$

SI units

$$h = 100\,\text{kPa}\,(0.102) = 10.2\,\text{m} \tag{2.1b}$$

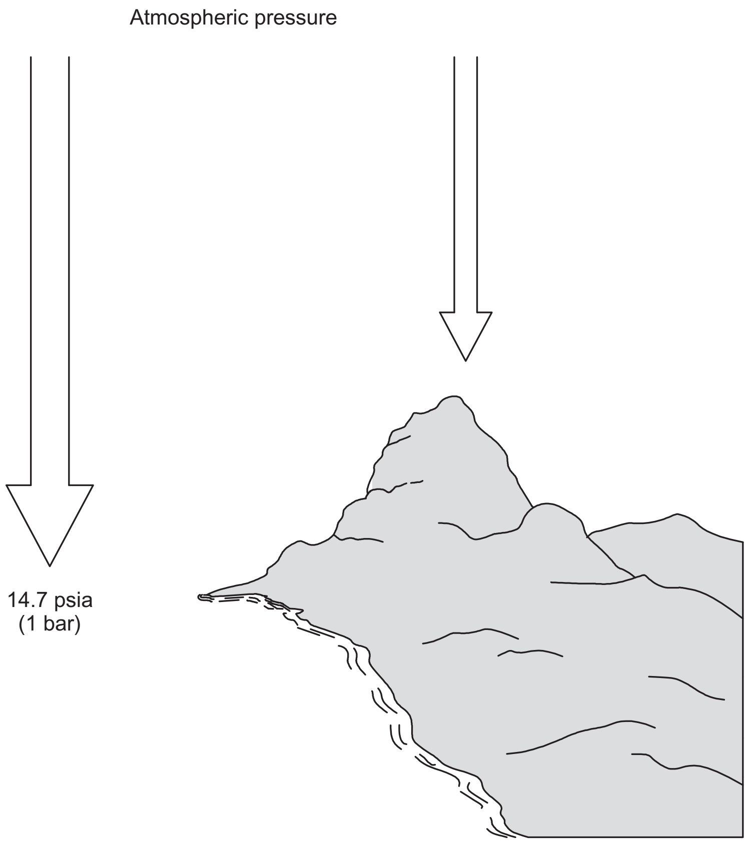


Fig. 2.12 Effects of elevation on atmospheric pressure.

Actual suction lift: Since a full vacuum is never achieved and since some lift is lost to friction in the suction line, the maximum actual suction lift for a positive displacement pump is about 22 ft (6.71 m) and 15 ft (4.5 m) for a centrifugal pump, when pumping water from an open air tank (Fig. 2.14). Positive displacement pumps can operate with lower suction pressures or higher suction lifts because they can create stronger vacuums. Suction lift will be greater if the pressure in a closed tank is greater than atmospheric pressure.

2.2.1.6 Submergence

Submergence is often confused with either suction static head or suction static lift. Submergence is defined differently for vertical and horizontal pumps. Fig. 2.15 illustrates the relationship between static head, static lift, and submergence.

Vertical pumps: Submergence relates the fluid level to the setting of the pump.

Horizontal pumps: Submergence relates to the amount of fluid level necessary in the source vessel or tank to prevent the formation of a vortex and the resulting drain of vapors into the pump.

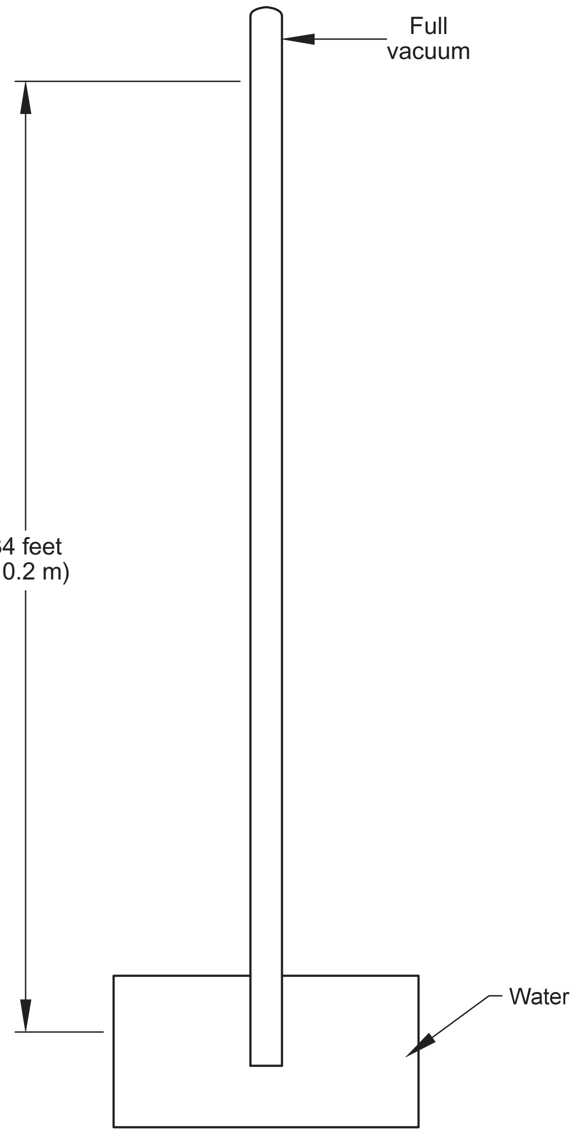


Fig. 2.13 Theoretical lift.

2.2.1.7 Vapor pressure

Vapor pressure is important when calculating the net positive suction head (NPSH). It is measured in psia (bara) and is a direct function of fluid temperature. At 60°F (15.6°C) the vapor pressure of water is 0.3 psia (0.7 ft). At 212°F (100°C) the vapor pressure of water is 14.7 psia (101 kPa).

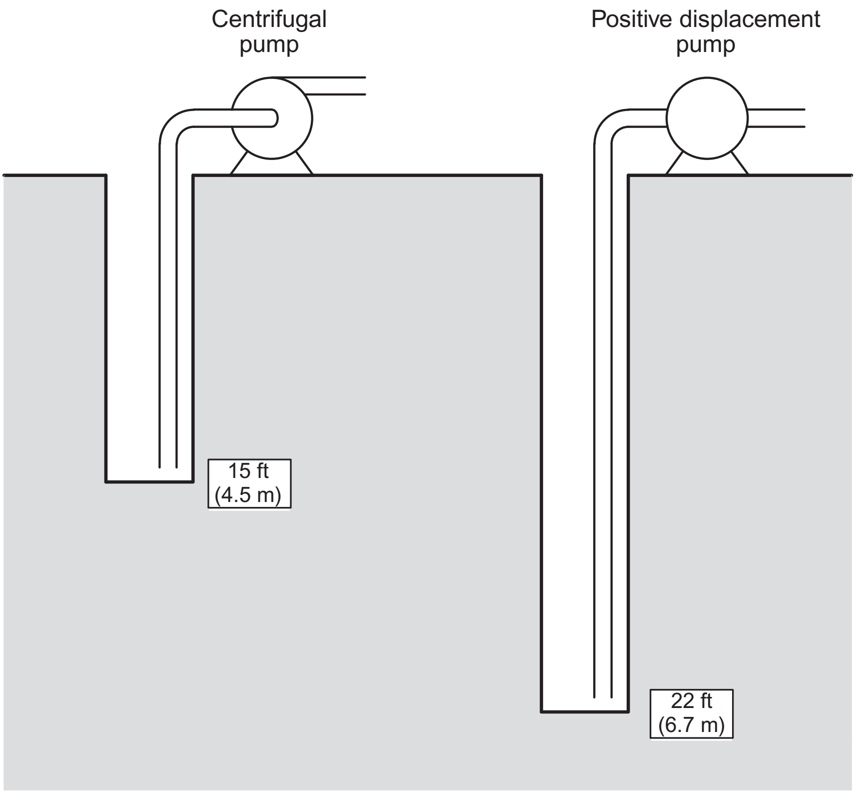


Fig. 2.14 Actual suction lift for a centrifugal and positive displacement pump.

2.2.1.8 Suspended solids

The amount and type of suspended solids entrained in the fluid can affect the characteristics and behavior of that fluid. Increased concentrations of solids increase:

- Specific gravity
- Viscosity
- Abrasiveness

The type and concentration of suspended solids can affect

- Style of pump selected
- Materials of construction

Suspended solids affect the selection of impeller design in centrifugal pumps, which in turn affects:

- Wear rate
- Efficiency
- Power consumption

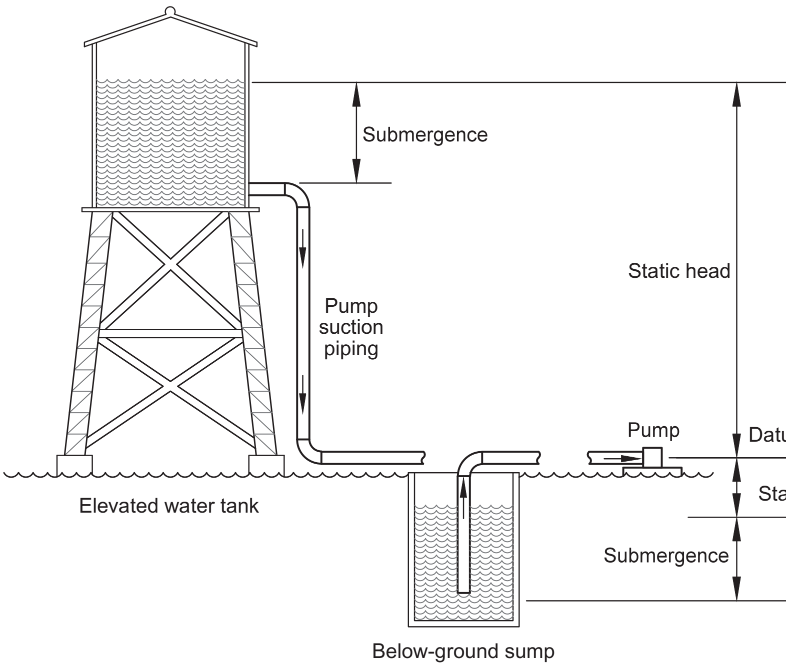


Fig. 2.15 Static head, static lift, and submergence in hydrostatics.

2.2.1.9 Dissolved gases

Small amounts of dissolved gases have little effect on flow rate or other pumping requirements. If large amounts of gas enter the liquid through piping leaks or as a result of a vortex in vessels, the specific gravity of the fluid will decrease and thus will also reduce the amount of NPSH available at the pump suction.

2.2.1.10 Viscosity

Viscosity offers resistance to flow due to friction within the fluid. Viscosity levels have a significant impact on

- Pump type selection
- Efficiency
- Head capability
- Warm-up

High viscosity affects centrifugal pumps by

- Decreasing the pump efficiency
- Decreasing the head performance
- Increasing the power requirements

Viscosity of all fluids varies with temperature. Reciprocating pumps can operate with viscosities up to 660 cSt (3000 Saybolt Second Universal (SSU)) and rotary pumps up

Table 2.1 Typical pump applications as a function of viscosity

Viscosity SSU (centistokes)	Example fluids @ 100°F (38°C)	Guidelines
<50 (7.4)	Water Beer Gasoline Alcohol 0.85 crude oil	Minimum for rotaries, centrifugals efficiency
< 150 (32.1)	0.90 crude oil Tri-ethylene glycol	Centrifugal always preferred over rotaries where a choice is available
150 (32.1)	0.95 crude oil	Centrifugal head-capacity curve begins to deteriorate
150–1000 (32.1–220)	SAE crankcase oils	Centrifugal still preferred over rotaries where choice is available despite performance falloffs
1000–3000 (220–660)	First cut molasses	Centrifugal seldom used, reciprocating pumps generally used
> 3000 (660)	Blackstrap molasses Asphalts	Centrifugal should never be used

to 6,000,000 SSU (1,320,000 cSt (6,000,000 SSU). A Saybolt viscometer measures the number of seconds it takes for a liquid to flow through a specified orifice and it is expressed as Saybolt Seconds Universal (SSU). Table 2.1 provides guidelines on typical practices.

2.2.1.11 Corrosive nature

The corrosive nature of the fluid being pumped has a bearing on pump type selection, materials of construction, and corrosion allowance. Special mechanical seals and flushing arrangements may be required.

2.2.2 Hydrodynamics

2.2.2.1 Overview

Hydrodynamics is the study of fluids in motion.

2.2.2.2 Continuity equation

The continuity equation (conservation of mass) states:

$$(\rho \times A \times V)_1 = (\rho \times A \times V)_2 \tag{2.8}$$

where

V = Velocity, ft/s (m/s)

Subscripts 1 and 2 refer to the continuity of flow that exists between two different points in the system. For incompressible fluids this simplifies to

$$(A \times V)_1 = (A \times V)_2 \tag{2.9}$$

2.2.2.3 Bernoulli's equation

Bernoulli's Law states that as a fluid flows from one point to another in a piping system the total of potential, static and velocity head at the upstream point (subscript 1) equals the total of the three heads at the downstream point (subscript 2) plus the friction drop between points 1 and 2. Bernoulli's equation states the following:

$$\frac{V_1^2}{2g} + \frac{P_1}{\rho} + Z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\rho} + Z_2 + h_f \tag{2.10}$$

where

$\frac{V^2}{2g}$ = velocity head, ft/s (m/s)
P/ρ = pressure head, ft (m)
Z = elevation head, ft (m)
h_f = head losses, ft (m)

An examination of each of the terms provides a better understanding of the general equation for modeling a pumping system, which is presented later in this chapter.

2.2.2.4 Velocity head

Potential energy that has been converted to kinetic energy is expressed by the following term:

$$\frac{V^2}{2g} \tag{2.11}$$

The velocity head is the first term in the momentum equation
where

V = Average velocity of the liquid in the pipe, ft/s (m/s)
= 0.4085 (gpm/d^2)
g = Acceleration due to gravity, ft/s^2 (Table 2.2)
= 32.17 ft/s^2 (9.81 m/s^2 at sea level at 45° north latitude
= Values vary with latitude and altitude

Velocity head is often expressed as
Field units

$$= 0.00259 \frac{(\text{gpm})^2}{d^4} \tag{2.12}$$

Table 2.2 Relation of acceleration of gravity (g) to latitude

Latitude degree	Acceleration due to gravity[a] (ft/s^2)
0	32.088
10	32.093
20	32.108
30	32.130
40	32.158
45	32.174
50	32.187
60	32.215
70	32.238
80	32.253
90	32.258

[a]At sea level; for altitudes above sea level, subtract 0.003 ft/s^2 for each 1000 ft.

or

$$= 0.0155\ V^2 \tag{2.13}$$

where

gpm = gallons per minute flow rate
d = inside pipe diameter, in (cm)

Velocity heads increase the amount of work required of a pump. It is usually not included in actual system calculations when piping velocities are kept within the prescribed limits recommended in *Volume 3* "*Plant Piping and Pipeline Systems*." Velocity heads are included in the total dynamic head on centrifugal pump curves.

2.2.2.5 Static head

The static head corresponding to any pressure depends on the weight of the liquid according to the following equation:
Field units

$$\text{Head (ft)} = \frac{\text{Pressure (psi)} \times (2.31)}{SG} \tag{2.14a}$$

SI units

$$\text{Head (m)} = \frac{\text{Pressure (kPa)}(0.102)}{SG} \tag{2.14b}$$

An examination of Eqs. (2.14a), (2.14b) shows that head is a constant for a pump at a specific operating condition; however, the differential pressure developed is a direct function of the liquid being pumped. For example, if the head is 100 ft and the specific gravity is equal to 1.0, then differential pressure increase is equal to $\frac{[(100)(1.0)]}{(2.31)} = 43.3$ psi but if $SG = 0.8$, then the differential pressure increase

$=\frac{[(100)\times(0.80)]}{2.31}=34.6$psi. This fact is often overlooked by inexperienced pump designers and is good to keep in mind.

The velocity head energy component is often used in system head calculations as a basis for establishing entrance losses, losses in valves and fittings, losses in other sudden enlargements, and exit losses by applying the appropriate resistance coefficient (K) to the $V^2/2\,g$ term. In system head calculations for high head pumps, the change in velocity head is a small percentage of the total head and is not significant. In low head pumps, however, it can be a substantial percentage and must be considered.

2.2.2.6 Head losses (h_f)

Head loss is potential energy that is converted to kinetic energy. Head losses are due to the frictional resistance of the piping system (pipe, valves, fittings, entrance, and exit losses). Unlike velocity head, friction head cannot be ignored in system calculations. Values vary as the square of the flow rate. Head losses can be a significant portion of the total head. The importance of head losses depends upon the magnitude of the total head. For example, an error of 6 ft (1.8 m) in estimating 2600 ft (780 m) of head is insignificant while an error of 6 ft (2 m) in estimating 30 ft (9 m) of head for a crude oil transfer pump could be significant.

2.2.2.7 Control losses

Control losses occur on the discharge side of a centrifugal pump that has been equipped with a flow control (backpressure) valve for the purpose of controlling flow rate. Again, the potential energy is converted to kinetic energy. Next to static head, control losses are frequently the single most important factor in calculating the pump's total dynamic head. For pump applications, control losses are treated separately from head losses, even though they are included in the "h_f" term in the momentum equation.

2.2.2.8 Acceleration head (H_A)

Acceleration head is used to describe the losses associated with the pulsating flow associated with reciprocating pumps. It is included in the "h_f" term of the momentum equation.

2.2.2.9 Suction head (H_S)

The suction head is the sum of the suction vessel operating gauge pressure (converted to feet (m)), the vertical distance between the suction vessel fluid level and the pump reference points, less head losses in the suction piping (discounting velocity head). Expressed as

Field units

$$H_s=\frac{P_1\times 2.31}{SG}+H_1-\frac{P_{f1}\times 2.31}{SG} \quad (2.15a)$$

SI units

$$H_s = \frac{P_1 \times (0.102)}{SG} + H_1 - \frac{P_{f1} \times (0.102)}{SG} \tag{2.15b}$$

Reducing

Field units

$$= \frac{\left(P_1 - P_{f1}\right)(2.31)}{SG} + H_1 \tag{2.16a}$$

SI units

$$= \frac{\left(P_1 - P_{f1}\right)(0.102)}{SG} + H_1 \tag{2.16b}$$

where

H_s = suction head of fluid being pumped, ft (m)
P_1 = Suction vessel operating pressure, psig (kPa)
= Lowest normal operating pressure on the surface of the pumped fluid
H_1 = Height of fluid in suction vessel above pump reference point, ft (m)
= The vertical distance, in feet (m), between the surface of the pumped liquid and the center of a horizontal pump suction flange
= Value is negative (−) when the pump is above the liquid surface.
P_{f1} = Pressure drop, line losses, due to friction in the suction piping, psi (kPa)

2.2.2.10 Discharge head (H_D)

The discharge head is the sum of the discharge vessel operating gauge pressure (converted to feet or meters), the liquid level in the discharge vessel above the pump reference point, pressure drop due to friction in the discharge piping, and control losses (discounting velocity head). Expressed as

Field units

$$H_D = \frac{P_2 \times 2.31}{SG} + H_2 + \frac{P_{f2} \times 2.31}{SG} + \frac{P_c \times 2.31}{SG} \tag{2.17a}$$

SI units

$$H_D = \frac{P_2 \times (0.102)}{SG} + H_2 + \frac{P_{f2} \times (0.102)}{SG} + \frac{P_c \times (0.102)}{SG} \tag{2.17b}$$

Reducing

Field units

$$= \frac{\left(P_2 + P_{f2} + P_c\right)(2.31)}{SG} + H_2 \tag{2.18a}$$

SI units

$$= \frac{\left(P_2 + P_{f2} + P_c\right)(0.102)}{SG} + H_2 \tag{2.18b}$$

where

H_D = discharged head of fluid being pumped, ft (m)
P_2 = discharge vessel operating pressure, psig (kPa)
H_2 = operating or normal height of fluid in the discharge vessel above the pump reference point, ft (m)
P_{f2} = pressure drop, line losses, due to friction in the discharge piping, psi (kPa)
P_c = discharge flow control valve losses, ft (m)

2.2.2.11 Total dynamic head (TDH)

The total dynamic head (TDH) is the difference between the pumping system discharge and suction heads. It is equal to the difference in pressure gauge readings (converted to feet or meters) across and existing operating pump (discounting velocity heads).

A pump's total dynamic head (TDH) is the difference between the suction and discharge heads (refer to Fig. 2.16)

$$\text{TDH} = H_D - H_S \tag{2.19}$$

Substituting
Field units

$$= \frac{(P_2 - P_1 + P_{f1} + P_{f2} + P_c)(2.31)}{SG} + H_2 - H_1 \tag{2.20a}$$

SI units

$$= \frac{(P_2 - P_1 + P_{f1} + P_{f2} + P_c)(0.102)}{SG} + H_2 - H_1 \tag{2.20b}$$

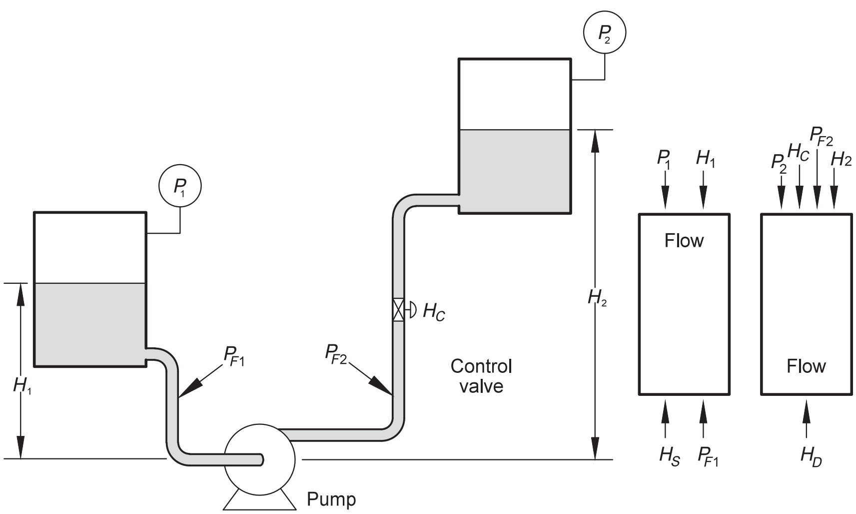


Fig. 2.16 Total head calculations.

where

TDH = total dynamic head, ft (m)

A pump's required discharge pressure rating is the combination of the maximum expected suction pressure and the maximum pressure developed by the pump. Centrifugal pumps require suction head and total dynamic head values be converted to units of pressure whereas reciprocating pumps do not require conversions.

2.2.2.12 Net positive suction head (NPSH)

The NPSH is defined as the total suction head (H_S), in feet (m) of liquid (absolute at the pump centerline or impeller eye) less the vapor pressure (in feet (m)), of the liquid being pumped.

2.2.2.13 Net positive suction head required ($NPSH_R$)

The $NPSH_R$ is the amount of NPSH required to move and accelerate the fluid from the pump suction into the pump itself (refer to Fig. 2.17). It is determined either by test or calculation by the pump manufacturer for the specific pump under consideration. The $NPSH_R$ is a function of

- Fluid geometry
- Smoothness of the surface areas

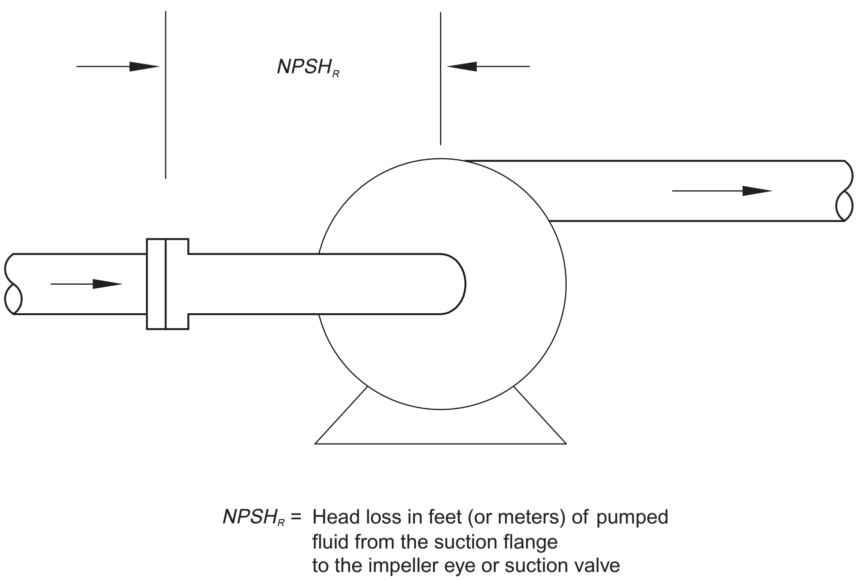


Fig. 2.17 Net positive suction head required ($NPSH_R$).

For centrifugal pumps, other factors included are as follows:

- Type of impeller
- Design of impeller eye
- Rotational speeds

2.2.2.13.1 $NPSH_R$ reductions for centrifugal pumps

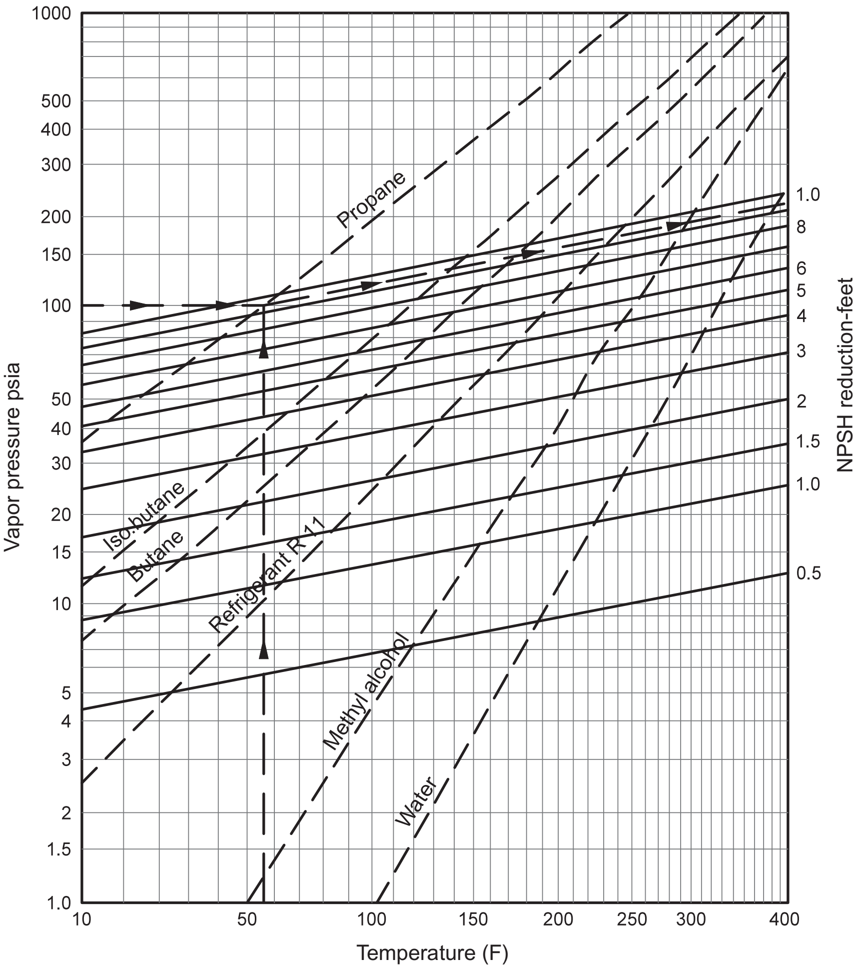
The $NPSH_R$ requirement for centrifugal pumps is determined on the basis of handling cool water. Field experience and laboratory testing have confirmed that centrifugal pumps handling gas-free hydrocarbon fluids and water at elevated temperatures will operate satisfactorily with harmless cavitation and less $NPSH_A$ than would be required for cold water.

Fig. 2.18 shows the reductions in NPSH that should be considered when handling hot water and certain gas-free pure hydrocarbon liquids. The use of Fig. 2.18 is subject to certain limitations some of which are summarized as follows:

(1) The NPSH reductions are based on laboratory test data at steady-state suction conditions and on gas-free pure hydrocarbon liquids shown; its application to other liquids must be considered experimental and is not recommended.
(2) No NPSH reduction should exceed 50% of the NPSH required for cold water or 10 feet (3 m) whichever is smaller.
(3) In the absence of test data demonstrating NPSH reductions >10 ft (3 m) the graph has been limited to that extent and extrapolation beyond that point is not recommended.
(4) Vapor pressure for the liquid should be determined by the bubble point method—do not use the Reid vapor pressure.
(5) Do not use the graph for liquids having entrained air or other noncondensable gases which may be released as the absolute pressure is lowered at the entrance to the impeller, in which case additional NPSH may be required for satisfactory operation.
(6) In the use of the graph for high-temperature liquids, particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure which might require additional NPSH to provide a margin of safety, far exceeding the reduction otherwise permitted for steady-state operation.

The procedure in using Fig. 2.18 is best illustrated by going through an example. Assume a pump requires a $NPSH_R$ of 16 ft (4.88 m), based on cold water at the design capacity, is required to handle pure propane at 55°F (12.8°C) which has a vapor pressure of approximately 100 psia (689.5 kPa). Entering Fig. 2.18 at a temperature of 55°F (12.8°C) and intersecting the propane curve shows a reduction of 9.5 ft (2.9 m) (which is greater than one half the cold water $NPSH_R$). The corrected value of the $NPSH_R$ is one half the cold water $NPSH_R$ or 8 ft (2.4 m).

Assume the same pump has another application to handle propane at 14°F (−10°C) where its vapor pressure is 50 psia (344.7 kPa). In this case the graph in Fig. 2.16 shows a reduction of 6 ft (1.8 m) which is less than one-half of the cold water NPSH. The corrected value of NPSH is, therefore, 16 ft (4.9 m) less 6 ft (1.8 m) or 10 ft (3 m). For a more detailed discussion on the use of this chart and its limitations, one should follow the requirements of the *Hydraulic Institute* Standards.



Note: Note chart has been constructed from test data obtained using the liquids shown. For applicability to other liquids refer to the text.

Fig. 2.18 NPSH reduction for pumps handling hydrocarbon liquids and high-temperature water.
Courtesy of Ingersoll-Dresser Pumps.

2.2.2.14 Net positive suction head available ($NPSH_A$)

The $NPSH_A$ is a critical factor in pump performance. It is a result of the suction system design. In practice, $NPSH_A$ is the differential pressure between (1) the actual pressure at the lowest pressure point in the pump, and (2) the pressure at which the liquid begins to vaporize (flash). $NPSH_A$ is the "available" pressure above the liquid's vapor pressure that prevents vaporization (or cavitation). As the liquid accelerates into the spinning impeller eye, its pressure drops. If pressure falls below the vapor pressure cavitation will result. For centrifugal pumps, the $NPSH_A$ is always expressed in feet of the liquid being pumped.

A pump's $NPSH_A$ is usually set by the height of the lowest liquid level in a column, vessel, or tank. With boiling liquids, static head (H_1) is the only source of $NPSH_A$. (Elevating the vessel or column, or establishing a low liquid level (LSL) which satisfies pump NPSH requirements is preferred over the use of pumps in a pit or vertical in a suction can.)

The $NPSH_A$ must be equal to or greater than $NPSH_R$ in order not to cavitate or flash across the pump. Cavitation results in decreased efficiency, capacity, and head. Flashing results as the pump cavity is filled with vapors, and as a result, the pump becomes vapor locked. The end result of this operation is usually pump seizure.

$NPSH_A$ is not a function of the pump itself but of the piping system for the pump. It can be calculated from the following (refer to Fig. 2.19)

Field units

$$NPSH_A = \frac{\left(P_1 + P_a - P_V - P_{f1}\right)(2.31)}{SG} + H_1 \qquad (2.21a)$$

SI units

$$NPSH_A = \frac{\left(P_1 + P_a + P_V + P_{f1}\right)(0.102)}{SG} \qquad (2.21b)$$

where

P_a = atmospheric pressure, psia (kPa absolute)
P_v = liquid vapor pressure at pumping temperature, psia (kPa absolute)

A primary factor in calculating the $NPSH_A$ for a pump is the vapor pressure of the liquid handled. One commonly used method, Reid vapor pressure, requires a certain amount of liquid to be evaporated in the measuring apparatus before the vapor pressure is indicated. Such vapor pressures are too low for determining when gas evolution will start (the point that will affect pump performance). This error is variable, being small for fractioned liquids and greater for crudes. The true vapor pressure (TVP) at the pumping temperature should be used for $NPSH_A$ calculations rather than vapor pressure by the Reid method.

When determining the TVP, one should not overlook the possibility of dissolved gases in the liquid. A frequent cause of NPSH trouble is dissolved or entrained gas in the liquid pumped. When tested by the bubble point method, water which has been aerated has a higher "vapor pressure" than water which has not been aerated. The same is true for hydrocarbons or other liquids. When the pressure of a liquid containing dissolved gases is reduced, the gas dissolved in the liquid may evolve and cause an effect similar to cavitation.

Vapor pressure is a function of temperature alone for any given composition of liquid. For some fluids, a small increase in temperature causes a relatively large increase in vapor pressure. When selecting a pump for such a fluid (e.g., water), the $NPSH_A$ should be calculated at the highest probable fluid temperature.

Fig. 2.20 is a vapor pressure curve for water. For other fluids, refer to the GPSA Engineering Data Book, Hydraulic Institute Engineering data Book or similar. The

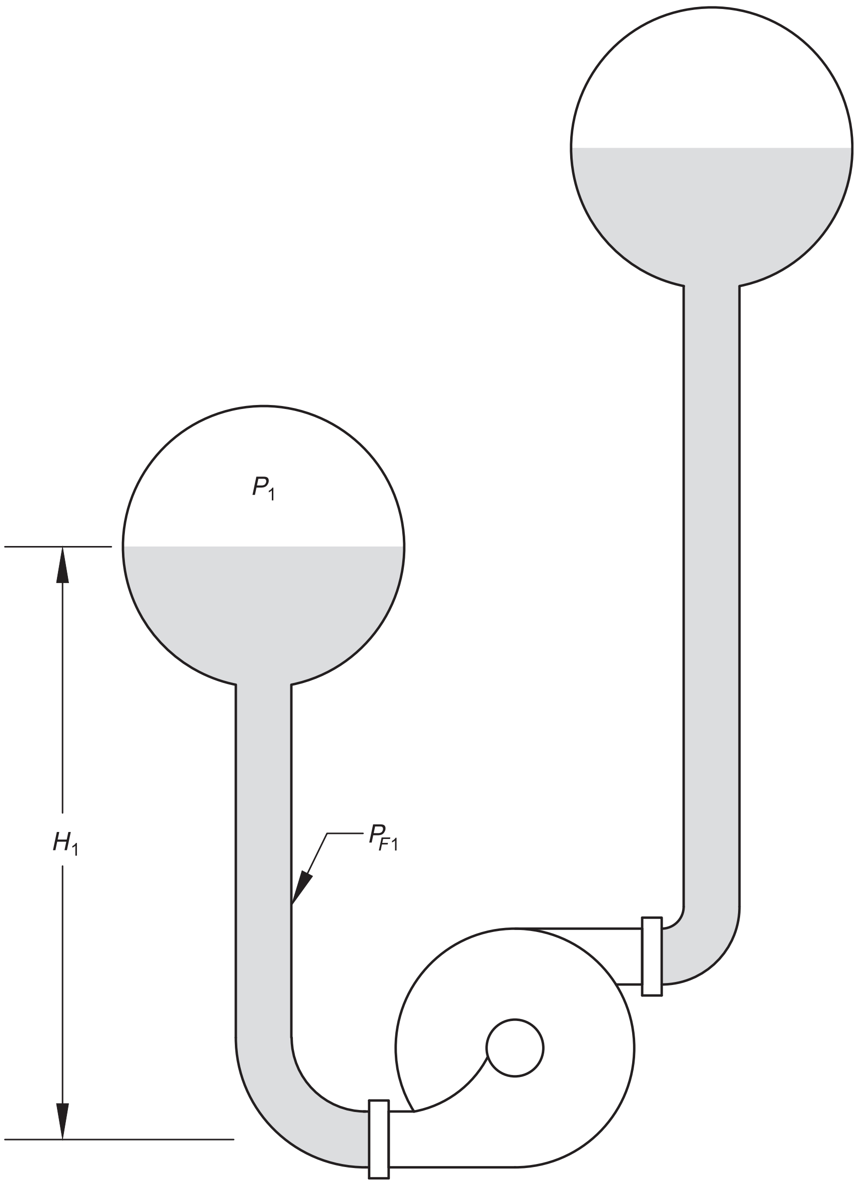


Fig. 2.19 Net positive suction head available $NPSH_A$.

$NPSH_A$ decreases linearly with increases in fluid temperatures and pipe friction losses. Since pipe friction losses vary as the square of the flow, $NPSH_A$ also varies as the square of the flow. Thus, $NPSH_A$ will be the lowest at the maximum flow requirement. Accordingly, it is important to recognize the need for calculating $NPSH_A$ (and $NPSH_R$) at maximum flow conditions as well as maximum fluid temperature, not just at design.

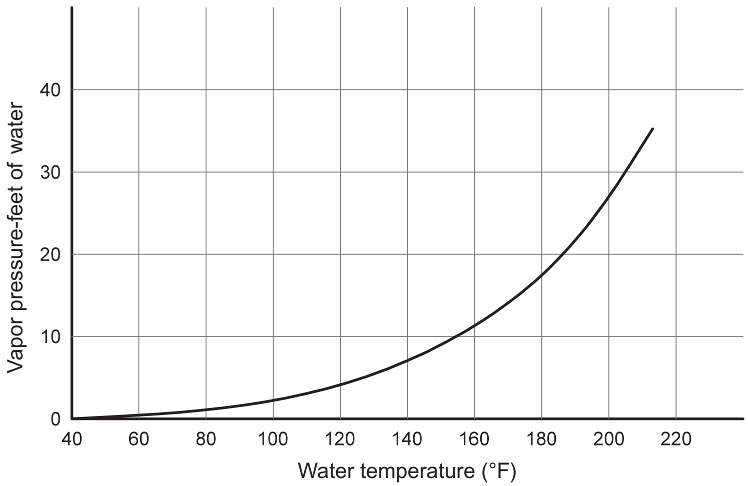


Fig. 2.20 Vapor pressure of water.

Unless subcooled, a pure component hydrocarbon liquid is typically in equilibrium with the vapors in a pressure vessel. Thus, increases in the vessel operating pressure are almost fully offset by a corresponding increase in the vapor pressure. When this happens

Field units

$$NPSH_A = H_1 - P_{f1}\left(\frac{2.31}{SG}\right) \tag{2.22a}$$

SI units

$$NPSH_A = H_1 - P_{f1}\left(\frac{0.102}{SG}\right) \tag{2.22b}$$

2.2.2.14.1 Acceleration head considerations for reciprocating pumps

The $NPSH_A$ for a reciprocating pump is calculated in the same manner as for a centrifugal pump, except in the $NPSH_R$ for a reciprocating pump some additional allowances must be made for the reciprocating action of the pump; this additional requirement is termed *acceleration head*. This head is required to accelerate the liquid column on each suction stroke so that there will be no separation of this column in the pump or suction line.

If this minimum condition is not met the pump will experience a fluid knock caused when the liquid column, which has a vapor space between it and the plunger (piston), overtakes the receding plunger (piston). This knock occurs approximately two-thirds of the way through the suction stroke. If sufficient acceleration is provided for the liquid to completely follow the motion of the receding face of the plunger (piston), this knock will disappear.

If there is insufficient head to meet minimum acceleration requirements of NPSH, the pump will experience cavitation, resulting in loss of volumetric efficiency.

In addition, serious damage can occur to the plungers (pistons), valves, and packing due to the forces released in collapsing the gas or vapor bubbles.

The head required to accelerate the fluid column is a function of the length of the suction line, the average velocity in this line, the rotating speed, the type of pump, and the relative elasticity of the fluid and the pipe, calculated as follows:

$$H_A = (LVnC)/Kg \tag{2.23}$$

where

H_A = acceleration head, feet (m)
L = length of suction line, feet (m)
V = velocity in suction line, ft/s (m/s)
= [(gpm)(0.321)]/[area of the suction pipe, in^2]
n = pump speed, rpm
C = 0.200 for duplex, single-acting
= 0.115 for duplex, double-acting
= 0.066 for triplex, single or double-acting
= 0.040 for quintuplex, single or double-acting
= 0.028 for septuplex, single or double-acting
= 0.022 for nonuplex, single or double-acting

Note: The constant "C" will vary from the earlier values for unusual ratios of connecting rod length to crank radius.

K = a factor representing the reciprocal of the fraction of the theoretical acceleration head which must be provided to avoid a noticeable disturbance in the suction line (K = 2.5 for hot oil; 2.0 most hydrocarbons; 1.5 for amine, glycol, water; 1.4 for deaerated water; and 1.0 for liquids with small amounts of entrained gases)
g = gravitation constant
= 32.174, ft/s^2 (9.81 m/s^2 at sea level at 45° north latitude)

The suction piping usually consists of pipes of various sizes. The acceleration head should be calculated for each section separately. The total acceleration head is obtained by adding the acceleration heads for each section.

2.2.2.15 NPSH margin

The NPSH margin is the difference between $NPSH_A$ and $NPSH_R$. The Hydraulic Institute recommends 3 to 5 ft (1 to 1.5 m) for centrifugal pumps and 7 to 10 ft (2 to 3 m) for reciprocating pumps. Perhaps a better approach is to specify a pump with a $NPSH_A$ that is less than what was calculated. We can use the following relationship

$$\text{Specified } NPSH_A = \frac{\text{Calculated } NPSH_A}{\text{Safety Factor}} \tag{2.24}$$

Typical safety factors

1.00 = existing system
1.10 = new services that have stable and well-controlled suction conditions

1.25 = new or old services that tend to have rapid, frequent, or severe fluctuations in suction conditions, such as a boiler feed water pump

Availability of pumps and pump models depends upon the calculated value of $NPSH_A$. For example

$NPSH_A < 7$ ft (2.25 m): Only a small number of pumps could be selected.
$NPSH_A > 25$ ft (7.5 m): Virtually any pump can be considered.

2.2.2.16 Inadequate suction conditions

When a new system offers insufficient NPSH margin for optimum pump selection, either

- $NPSH_A$ must be increased,
- $NPSH_R$ must be decreased, or
- Both

To increase the $NPSH_A$ the following can be done:

- Increase the static head (H_S)
  - Raise/add liquid level from reservoir, expansion tank, and so on
  - Lower the elevation of the selected pump
  - Reduce suction piping friction losses
  - Add a booster pump
  - Increase pressure in the system
- Add a pulsation damper
- Cool the liquid

A properly installed pulsation damper with a short, full-size connection to the pump or suction pipe can absorb the cyclical flow variation and reduce the pressure fluctuation in the suction pipe to that corresponding to a length of 5 to 15 (1.5 to 4.5 m) pipe diameters, if kept properly charged.

There is a similar pressure fluctuation on the discharge side of every power pump, but it cannot be analyzed as readily because of the greater influence of liquid and piping elasticity and the smaller diameter and much greater length of the discharge line in most applications. However, a pulsation damper can be just as effective in absorbing the flow variation on the discharge side of the pump as on the suction side and should be used if pressure fluctuation and piping vibration is a problem.

There are several ways to correct NPSH problems, depending on whether the system is an open or closed system. In an open system, one can raise the liquid level to increase suction pressure or lower the pump. Sometimes one may need to increase the suction line size in order to lower the friction losses. A closed system may require installation of a booster pump, raising system pressure, or maybe a different pump with a lower $NPSH_R$ requirement.

To reduce $NPSH_R$ the following can be done:

- Use slower speeds
- Use double-suction impeller

- Use large impeller eye
- Change to a low $NPSH_R$ pump
- Oversize pump
- Use different design impellers or inducers
- Use several smaller pumps with lower $NPSH_R$'s in parallel
- Use an inducer.

Inducers have severely restricted operating flow range abilities relative to noninducer pumps. Inducers should never be used in erosive services.

Some of the corrections suggested before are impractical and could instead simply require a change of pumps.

When an existing pumping system exhibits insufficient NPSH margin, it is too late to use the aforementioned solutions without going through an expensive change. The majority of these problems can be traced to suction line flow restrictions (orifice plates, plugged strainers, partially closed valves, etc.) and inadequate source tank fluid levels (including vortexing).

Example 2.1. Calculation of acceleration head (field units)
Given:

(a) A 2″ x 5″ triplex pump running at 360 rpm displaces 80 gpm of water.
(b) Suction line consists of 4 ft of 4-in. pipe and 20 ft of 6-in. pipe.

Determine:

Determine the acceleration head

Solution:

(1) Determine the average velocity in the 4-in. pipe

$$V_4 = \frac{(0.321)(80)}{(12.73)} = 2.02\,\text{fps}$$

(2) Determine the average velocity in the 6-in. pipe

$$V_6 = \frac{(0.321)(80)}{(28.89)} = 0.89\,\text{fps}$$

(3) Determine the acceleration head in the 4-in. pipe

$$H_{A4} = \frac{(4)(2.02)(360)(0.066)}{(1.4)(32.2)} = 4.26\text{ ft}$$

(4) Determine the acceleration head in the 6-in. pipe

$$H_{A6} = \frac{(20)(0.89)(360)(0.066)}{(1.4)(32.2)} = 9.38\,\text{ft}$$

(5) Total acceleration head

$$\begin{aligned} H_A &= H_{A4} + H_{A6} \\ &= 4.26 + 9.38 \\ &= 13.64\,\text{ft} \end{aligned}$$

2.2.2.17 Power requirements

The hydraulic horsepower (HHP) for a centrifugal pump is a theoretical value calculated from the rated capacity and differential head, assuming a 100% efficient pump. It can be calculated as:
Field units

$$HHP = \frac{(TDH)(\rho)Q}{550} \tag{2.25a}$$

SI units

$$HHP = \frac{(TDH)(\rho)Q}{367{,}000} \tag{2.25b}$$

where

HHP = hydraulic horsepower, hp.; 1 hp =550 ft lb/s) (kW)
TDH = pump head, ft (m)
ρ = density of liquid, lb/ft^3 (kg/m^3)
Q = flow rate, ft^3/s (m^3/h)

By making the appropriate unit conversions, Eqs. (2.25a), (2.25b) may be expressed as:
For kinetic energy pumps
Field units

$$HHP = \frac{(SG)(q)(TDH)}{3960} \tag{2.26a}$$

SI units

$$HHP = \frac{(SG)(q)(TDH)}{367.6} \tag{2.26b}$$

For positive displacement pumps
Field units

$$= \frac{(\text{gpm})(\Delta P)}{1714e} \tag{2.27a}$$

SI units

$$HHP = \frac{(m^3/h)(\Delta P)}{3600} \tag{2.27b}$$

Field units

$$HHP = \frac{(Q)(\Delta P)}{58{,}766} \tag{2.28}$$

where

HHP = hydraulic horsepower
ΔP = maximum working pressure of PD pumps, psig (kPa)

Q = flow rate, bpd (m^3/h)
q = flow rate, gpm (m^3/h)
SG = specific gravity relative to water

The input power to the shaft of the pump is called the brake horsepower and it is given by

$$BHP = HHP/e \tag{2.29}$$

where

BHP = brake horsepower
e = pump efficiency factor, obtained from the pump manufacturer or approximated from Fig. 2.21

For electric motor-driven pumps, the energy consumption can be estimated by

$$KWH/\text{day} = \frac{17.9(BHP)}{\text{Motor Efficiency}} \tag{2.30}$$

Note that changes in specific gravity will affect the power calculations but does not affect the head capability of a pump. The pump efficiency chart (Fig. 2.21) should be

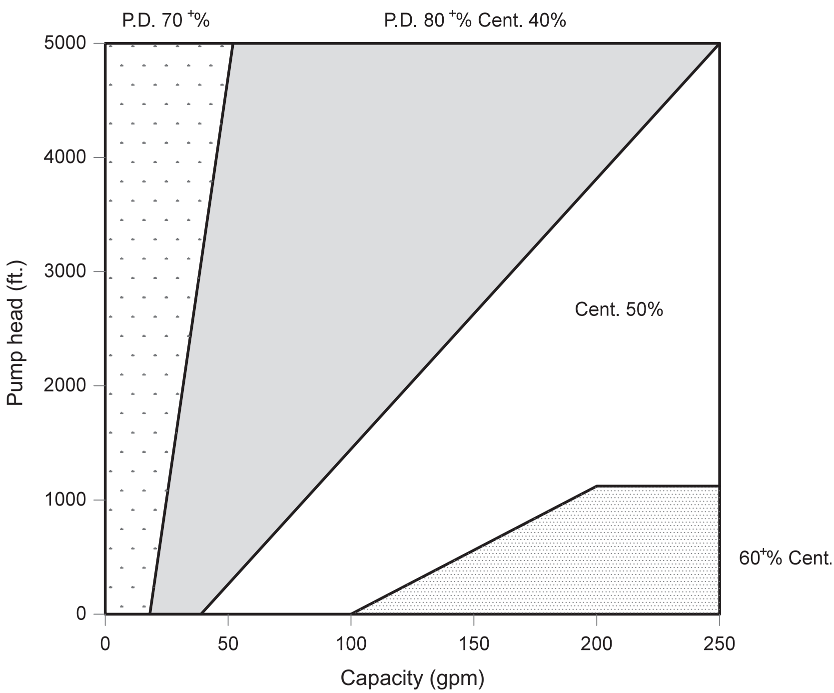


Fig. 2.21 Pump efficiency chart (for screening estimates only).

used for estimating purposes only. It is useful to determine whether a particular pumping application is best met by using a positive displacement or centrifugal pump. Positive displacement pumps offer high efficiencies at high TDHs and low flow rates. Centrifugal pumps offer better efficiencies at high flow rates and low TDH. For moderate TDHs and flows, either type could be used, depending on factors other than efficiency.

Example 2.2. Calculation of pump head and horsepower (field units)

Given:

(a) A pump takes suction from a vessel operating at 15 psig discharges to a tower operating at 500 psig.
(b) Fig. 2.22 and the following data:

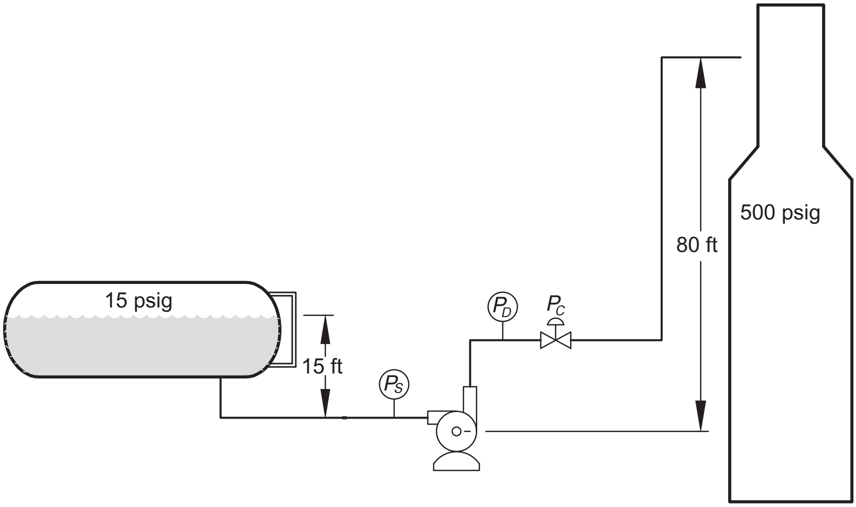


Fig. 2.22 Schematic for Example 2.2.

Flow	150 gpm
Specific gravity	0.8
Line losses	2 psi suction
Control losses	4 psi discharge
Static head	10 psi
	15 ft suction
	80 ft discharge
Atmospheric pressure	14.7 psia
Kinetic pump efficiency	0.52
PD pump efficiency	0.82
Motor efficiency	0.92

Determine:

Determine the following for both kinetic and positive displacement pumps:

(1) Pump suction head
(2) Pump discharge head
(3) Total dynamic head
(4) Pump horsepower required
(5) Electrical energy consumption

Solution:

1. Determine pump suction head

$$\begin{aligned} H_s &= \frac{(P_1 - P_{f1})(2.31)}{SG} + H_1 \\ &= \frac{(15-2)(2.31)}{0.8} + 15 \\ &= 52\,\text{ft}\,(\text{Kinetic}) \end{aligned}$$

or

$$= 52\left(\frac{0.8}{2.31}\right)$$

$$= 18\,\text{psig}\,(\text{Positive Displacement})$$

2. Determine pump discharge head

$$\begin{aligned} H_D &= \frac{(P_2 + P_{f2} + P_c)(2.31)}{SG} + H_2 \\ &= \frac{(500+4+10)(2.31)}{0.8} + 80 \\ &= 1564\,\text{ft}\,(\text{Kinetic}) \end{aligned}$$

or

$$= 1564\left(\frac{0.8}{2.31}\right)$$

$$= 542\,\text{psia}\,(\text{Positive Displacement})$$

3. Determine the total dynamic head

$$\begin{aligned} TDH &= H_D - H_s \\ &= 1564 - 52 \\ &= 1512\,\text{ft}\,(\text{Kinetic}) \end{aligned}$$

or

$$= 1512\left(\frac{0.8}{2.31}\right)$$

$$= 524\,\text{psig}\,(\text{Positive Displacement})$$

4. Determine the pump horsepower required

$$
\begin{aligned}
BHP &= \frac{(\text{gpm})(TDH)(SG)}{3960e} \\
&= \frac{(150)(1512)(0.8)}{3960(0.52)} \\
&= 88(\text{Kinetic})
\end{aligned}
$$

or

$$
\begin{aligned}
BHP &= \frac{(\text{gpm})(\Delta P)}{1714e} \\
&= \frac{(150)(524)}{1714(0.82)} \\
&= 56(\text{Positive Displacement})
\end{aligned}
$$

5. Determine the electrical energy consumption

$$
\begin{aligned}
KWHR/\text{Day} &= \frac{(17.9)(BHP)}{\text{Motor Efficiency}} \\
&= \frac{(17.8)(88)}{0.92} \\
&= 1.712(\text{Kinetic})
\end{aligned}
$$

or

$$
\begin{aligned}
KWHR/\text{Day} &= \frac{(17.9)(56)}{0.92} \\
&= 1090(\text{Positve Displacement})
\end{aligned}
$$

2.3 Selection criteria

2.3.1 General considerations

The first decision one must make when selecting a pump is to determine the type of pump to use. The "pump type" can be either centrifugal, reciprocating, rotary, metering, seal-less, or miscellaneous. This section discusses the advantages, features, and limitations of each pump type. Once a pump type is selected, one should refer to the respective chapter in this volume for a detailed discussion.

2.3.2 Selection fundamentals

Generally, centrifugal pumps should usually be considered first. The reason for this is due to their low cost, simplicity, reliability, and nonpulsating flow characteristics. However, under the following conditions, reciprocating or rotary pumps may be more applicable:

- *Very high head and low capacity*. A positive displacement (reciprocating or rotary) pump is designed for these conditions. However, centrifugal pumps of special design (high speed) have been built to pump as little as 10 to 20 gpm (against pressures of 1000 psi or more).
- *Low speed drivers*. A centrifugal pump for normal oil service operates at nominal speeds of 1800 or 3600 rpm. A pump driven by an internal combustion engine operating at 600 rpm will require a speed increaser. A gearbox can easily cost more than the pump. In this case, a positive displacement pump may prove more economical.
- *High efficiency requirements over a range of pressure conditions*. This situation frequently occurs in pipeline transportation. A positive displacement pump at any capacity remains reasonably efficient at pressures between 25% and 100% of its rating. A centrifugal pump will not remain efficient in these circumstances.

In pipeline service, centrifugal pumps usually are connected in series while reciprocating pumps are connected in parallel. These arrangements are most suitable to the pump's characteristics.

- *High viscosity*. The efficiency of centrifugal pumps starts to fall significantly when the viscosity of the oil exceeds 100 cSt or 500 SSU. While centrifugal pumps are used successfully at higher viscosities, especially where the pump is large, rotary or reciprocating pumps will probably be more economical.
- *Emulsification*. Impellers impose great agitation on pumped stock, making centrifugal pumps a poor choice where water/oil emulsions must be avoided. Both rotary and reciprocating pumps are better equipped to prevent emulsification.

When selecting a pump for a given service, conflicting factors may make the choice unclear. For example, if a pumped fluid is both highly viscous and contains abrasives, a rotary pump may be the first choice for the viscous fluid. However, because of the abrasives, ultimately it would be a poor choice.

2.3.2.1 Pump selection charts

Pump selection charts, Fig. 2.23, provide additional information that are helpful in choosing between centrifugal, reciprocating, and rotary pumps. The pump selection charts show the type of pump most suitable from a head-capacity standpoint. The pump selection charts can be used for initial selection of a pump type. Generally, select a centrifugal pump unless:

(a) Capacity at pumping head falls within area labeled "Positive Displacement Pumps."
(b) Viscosity is >1000 cp at pumping temperature.
(c) Percentage of undissolved gas by volume is $>5\%$ or gas is not well dispersed.
(d) The percentage of solids by volume is $>50\%$ or solids are not well dispersed.
(e) The pump is expected to run dry without automatic shutdown provisions, for example, pump out pumps.

2.3.2.2 Pump selection guide

The *pump selection guide*, Table 2.3, is used to identify other factors affecting the pump selection. The local priority column on the Pump Selection Guide weighs

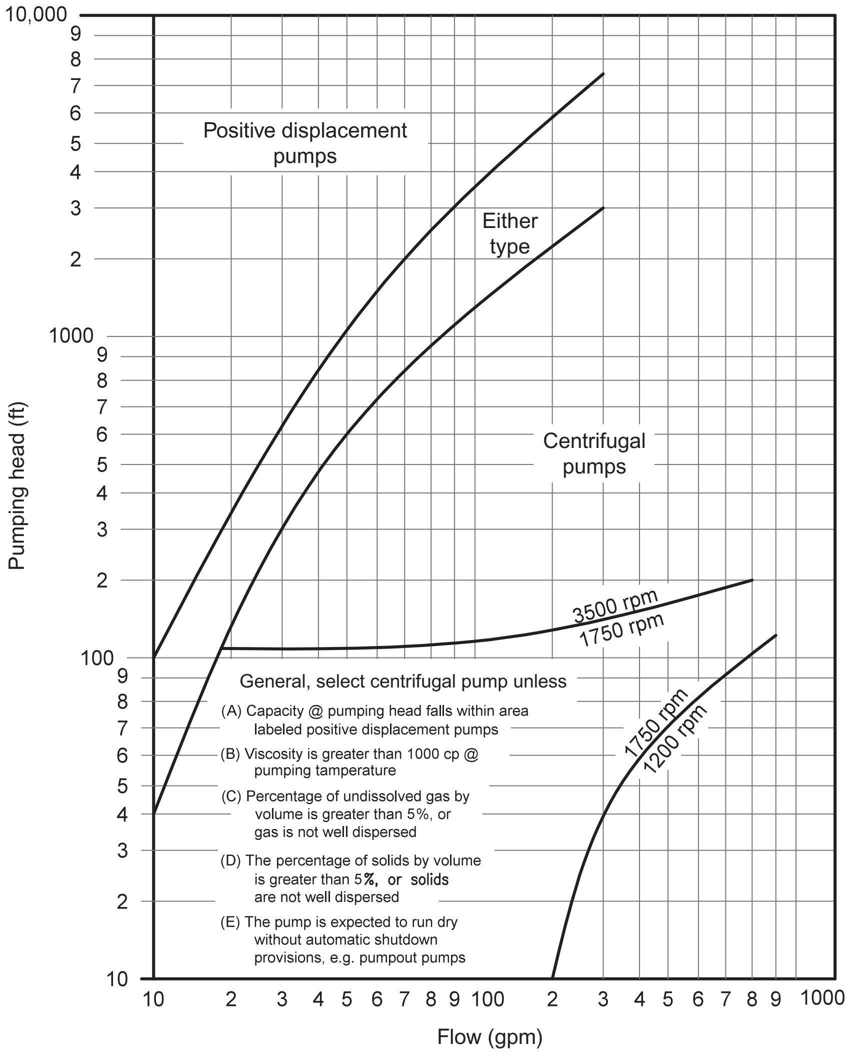


Fig. 2.23 Pump selection chart (1 of 2).
Courtesy of Ingersoll-Rand.

factors by their importance in a specific application. The designer should also consider operations preference, spare parts availability, and local maintenance capabilities.

2.3.3 Centrifugal pumps

Centrifugal pumps are used in approximately 80% of all production operations. There are many reasons why centrifugal pumps are used more often than any other pump type. Some of these reasons are as follows:

Table 2.3 Pump selection guide

Selection criteria	Local priority			Reciprocating			Centrifugal			Rotary			Comments
	Critical	Import	Not imp	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	
Flow rate vs developed head				—See head/capacity selection curve—									
Avoids emulsifying the fluid				X					X	X			
Reliability						X	X				X		
Self-priming				X					X	X			Self-priming centrifugal designs are available
Handling abrasives					X			X				X	"Progressive cavity" rotary pumps handle abrasives
Handling low $NPSH_A$						X	X				X		$NPSH_A$ on Recip's must include allowance for "acceleration head"
Handling entrained gas					X			X		X			
Pump viscous stock					X				X	X			
Pump low viscosity stock				X			X					X	
Energy efficiency				X				X			X		Centrifugal efficiency varies with type of pump, and flow rate relating to BEP
Installed cost					X		X			X			
Maintenance cost						X	X				X		

Table 2.3 Continued

Selection criteria	Local priority			Reciprocating			Centrifugal			Rotary			Comments
	Critical	Import	Not imp	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	
Controls leakage to atmosphere						X	X (IF SEALED)			X (IF SEALED)			Air quality permits may dictate pump selection
Temperatures above 350°F													Varies with application. review each specific application
Client preferences													Many clients have strong preferences based on operating experience
Compatibility with existing system													Consider maintenance facilities, parts, and operator familiarity
Weight and space													Very important in offshore or other limited space applications. Review each specific application

- Lowest initial Capital Expense (CAPEX) and lowest Operating Expense (OPEX)
- Very reliable
- Capable of achieving a wide range of flow rates (10 to over 100,000 gpm)
- Capable of developing a wide range of heads (20 to over 10,000 ft)
- Ability to operate with relatively low $NPSH_A$
- Wide temperature and pressure capabilities
- Minimal environmental impact with proper seals
- Flow variation flexibility
- Easily controlled
- Available from many manufacturers
- Available in many different materials of construction
- Available in a wide range of designs and types
- Experiences low vibration
- Generates low pressure pulsations
- Capable of handling suspended solids

Limitations of centrifugal pumps include:

- *When a self-priming pump is needed*. Conventional centrifugal pumps must be primed and supplied with adequate NPSH for proper application. There are some centrifugal pumps available with a self-priming design. However, the available sizes are limited and performance penalties are substantial.
- *Low flow, moderate to high head applications*. Operating centrifugal pumps to the left of their best efficiency point (BEP) causes many problems. For example, impeller eye recirculation and high levels of vibration result when operating below the recommended minimum flow. The effects of low flow operation are shaft breakage, seal and bearing failures.

The definition of "minimum flow" is pump specific. Some of the factors that affect the minimum flow are fluid density, $NPSH_A$, impeller design suction specific speed, casing volute design, piping and control systems designs. Sundyne vertical in-line pumps are designed for low-flow, moderate to high head applications. However, these pumps will also have a minimum flow that should be maintained for reliable service.

- *Handling fluids with entrained gas*. Centrifugal pumps can lose suction due to an excess amount of gas accumulation in the impeller eye. The vapor or gas accumulation can be caused by excessive cavitation, recirculation, or entrained gas.

The centrifugal pump impeller is a good centrifuge. The heavy material (liquid) is expelled through the impeller while the light fluid (gas) will collect in the eye of the impeller. If the gas or vapor volume fills the impeller eye, the pump may lose suction.

A worst-case example of this would be a centrifugal pump selected for an application with little $NPSH_A$, some entrained gas, and operated at reduced flow rates. At reduced flows, the fluid velocity in the suction pipe may not be capable of pushing the entrained gas or vapor through the impeller.

- *Pumping viscous fluids*. Although centrifugal pumps are capable of pumping fluids with a viscosity of 880 cSt (4000 SSU) and higher, the performance penalties are substantial. As presented in Table 2.4, pump efficiency can be drastically reduced. The pump's capacity and head capability are also reduced.

Table 2.4 Centrifugal pump selection and application

		Typical capacity range (gpm)	Typical head range feet	Common applications
1	Vertical inline (single stage)	20 to 1200	15 to 600	General processing and transfer service at temperatures below 350°F. Minimum space application
2	End suction, frame mounted ANSI	35 to 4000	30 to 700	General processing and transfer service at temperatures below 350°F
3	Horizontal, single-stage, horizontal suction between bearings API	100 to 20,000	40 to 900	Hydrocarbons in the low to moderate flow and moderate head ranges. Cooling tower water circulation
4	Horizontal, multistage axially split	200 to 1500	200 to 4500	Used to pump crude oil, high-pressure boiler feed water, seawater, gasoline, and other hydrocarbons and also in waterflood operations
5	Horizontal barrel (double case)	200 to 1700	up to 9000	Used principally for process plant high-pressure reactor charge and waterflood applications. Pump speed may approach 7500 rpm
6	Radially split vertical can	20 to 2000	55 to 2000	Used principally for improving $NPSH_A$ when pumping bubble point hydrocarbon mixtures. Pumping end usually encased in a pressure vessel (can)
7	Vertical turbine lineshaft	100 to 30,000	10 to 1500	Used for lift application such as seawater and fire water
8	Electric submersible	100 to 30,000	10 to 2000	Used to eliminate long shaft lengths. Same application as vertical turbines and crude oil production

The selection of rotary or reciprocating pumps may be a better choice for low-flow, viscous services. However, if the required flow rate exceeds the capability of rotary or reciprocating pumps, centrifugal may be the only choice.

- *When fluid emulsification must be avoided.* Centrifugal pumps are good agitators and mixers. Rotary or reciprocating pumps are a better choice to minimize fluid emulsification.

2.3.4 Reciprocating pumps

Reciprocating pumps are used most often for low-capacity, high-pressure service. The initial cost of small reciprocating pumps is competitive with centrifugal pumps. However, larger reciprocating pumps (>200 gpm) are usually more expensive (initially and to maintain) than other pump types. High-speed centrifugal pumps should not be overlooked when low-capacity high head services are encountered. Table 2.5 compares typical positive displacement applications.

When service requirements allow using either a centrifugal or a reciprocating pump, one should carefully consider both operating and maintenance costs. For most services, the operating costs of motor- or turbine-driven centrifugal pumps are less than the costs for reciprocating pumps. Maintenance costs usually exceed those of a centrifugal pump because of the many moving parts, including valves and sliding contacts.

Pulsating flow may limit the use of reciprocating pumps. However, pulsating flow is usually not the decisive factor for determining if reciprocating pumps are the best selection. The effects of pulsation can be minimized but not eliminated by using pulsation dampers. Pulsating flow may cause problems in the application of automatic control flow measurement or process.

Typical applications for reciprocating pumps include:

- *Low to moderate capacity with high differential pressure*. Fig. 2.23 show the head-capacity range for which reciprocating pumps are normally considered in nonviscous services. The division shown is intended as a guideline only.
- *Relatively high viscosity*. The efficiency of a centrifugal pump drops rapidly with increasing viscosity. The most economical applications of these pumps are normally limited to viscosities under about 110 cSt (500 SSU). Reciprocating pumps can efficiently handle stocks up to about 1760 cSt (8000 SSU).

Higher viscosity oils can be delivered by reciprocating pumps operating at slower speeds, but such applications usually are not economical. Rotary pumps are more appropriate.

- Relatively constant capacity with widely varying discharge pressures. Reciprocating pumps are particularly suited to this application.
- Highly variable capacity with either constant or varying discharge pressure. Direct-acting, gas-driven pumps are well suited to this application because the speed is easily controlled by the gas driver.
- Where a self-priming pump is needed.

Limitations of reciprocating pumps include:

- *When pulsating flow is undesirable*. An example of such a service is a fuel oil feed to boilers. Ordinarily, rotary pumps are preferred for this service because of their smooth discharge pressure and better efficiency at higher viscosities.
- *Medium capacity and medium differential pressure with low viscosity*. Centrifugal pumps ordinarily are more economical. Examples of such services are water or hydrocarbons pumping about 100 gpm with differential pressures up to about 700 ft or 300 psi.

Table 2.5 Positive displacement pump selection and application

		Maximum flow (gpm)	Maximum pressure (psi)	Application
1	Plunger	300	6000	Used primarily as glycol pumps, steam generator feed pumps, condensate pumps, drilling mud pumps, and additive injection pumps, especially where solids are present
2	Piston	800	1500	Same as plunger, but without solids present
3	Diaphragm	10	1000	Used mainly in controlled volume applications as a metering pump or where low flow rates and high solids concentrations are present. Temperatures limited to <500°F maximum
4	Cam-and-piston			Vacuum services
5	Rotary gear	150	700	Used to transfer recovered oil from a drain separator to a process oil/water separator. Also used as a diesel transfer pump and in high-viscosity applications, and for circulating lubrication oil services for large machines and other clean, high- and low-pressure services
6	Single screw or progressive cavity	450	200	Used for high-viscosity stocks (non-Newtonian), stocks with up to 30% (entrained) gas, and often for abrasive services. Not good for temperatures over 300°F and in carbon dioxide entrained gas services
7	Three-screw	1000	4500	Used for low-temperature, high-viscosity stocks (lower then progressive cavity) without abrasives and circulating lubrication oil for large machines
8	Two or twin-screw	8000	4500	Used for low- and high-temperature liquids with and without abrasives (up to 600°F), and as a multiphase pump (90% plus gas, the rest liquid) with and without abrasives with special design
9	Sliding vane			Vacuum services

(1) For reciprocating pumps, pumps 1, 2, and 3 above. The recommended rpm is a function of stroke length (inches). See API Standard 674.
(2) For reciprocating pumps, pumps 1, 2, and 3 above, handling liquids with viscosities of 300 SSU at pumping temperature, the speeds are normally reduced to a percent of the basic speed. See API Standard 674.

- *High capacity*. For capacities above about 200 gpm, reciprocating pumps are seldom the best selection, regardless of the discharge pressure. In this range, reciprocating pumps become so large that they are more expensive than centrifugal pumps. An exception may be high-pressure water injection services.
- *Minimum packing leakage required*. Some hazardous or toxic services require absolute minimum stock leakage. Reciprocating pumps are subject to packing leaks and in such services must be fitted with a double stuffing box to provide an enclosed leakage disposal system. Centrifugal pumps with mechanical seals would be preferred.

2.3.5 Rotary pumps

- The most common rotary pumps are gear, multiple screw, and single screw. Cam-and-piston and sliding vane pumps can be considered for special services. The following discussion applies to all rotary pumps. Refer to Table 2.5 for positive displacement applications.

Typical applications for rotary pumps include

- *Viscous fluids*. Rotary pumps can deliver high-viscosity fluids with a smaller reduction in efficiency than other pump types. Rotary pumps can handle fluids with viscosities varying from LPG (not recommended for continuous service) to very viscous greases. For fluids with viscosities >2200 cSt (10,000 SSU), rotary pumps are usually the most economical selection.

Special reciprocating pumps operating at greatly reduced speeds can handle viscosities as high as those handled by rotary pumps, but these pumps must be so large, they become prohibitively expensive. Under specific conditions, centrifugal pumps can deliver viscosities up to about 1100 cSt (5000 SSU), but their efficiency above 110 cSt (500 SSU) is so poor that such applications are not economical.

- *Lubricating and hydraulic oils*. Rotary pumps are most commonly used to circulate lubricating oil in mechanical equipment or to provide pressure for hydraulic operating systems. The oil used in these systems is usually cleaned by filtering. The pumped fluid lubricates the pump's internal gears and bearings.
- *Self-priming*. Rotary pumps work well when services require self-priming in moderate capacities, such as barrel, small tank, and sump unloading. However, when these services involve high capacities and low-viscosity fluids, vertical centrifugal pumps are usually used.
- *Vacuum services*. Rotary pumps, lubricated by special oils, are often used in vacuum services to pump air or other gases and vapors. Low vapor pressure oils are used to lubricate the pumps and seal the clearance spaces. Oil separators on the discharge of such pumps remove the oil from the gas. Cam-and-piston type and sliding vane-type rotary pumps are commonly used in this manner. Pressures as low as 2×10^{-4} mm mercury absolute are attainable with the cam-and-piston type.

With either the oil or water seal, the vacuum obtainable is limited by the vapor pressure of the sealing liquid.

- *Intermittent low-capacity services*. Small internal-bearing rotary pumps can sometimes be used economically in intermittent services where rotary pumps might seem to be unsuited. In these cases, it is cheaper to periodically replace inexpensive pumps than it is to buy pumps not subject to the same rate of wear.

An example of this application is pumping out small tanks or vessels where a certain amount of scale and grit is expected. Another example is intermittent handling of nonlubricating fluids, such as LPG and gasoline on tank trucks.

- *Nonpulsating flow*. Hydraulic operating systems and fuel oil systems usually require nonpulsating flow. Rotary pumps work well in these services, especially when high viscosity renders centrifugal pumps uneconomical.
- *Handling "wet" oil*. "Wet" oils (>3% water by volume) should be used only with pumps that will not cause oil to emulsify. Rotary pumps operated at slow speed (300 to 400 rpm) work well in such services.

Limitations of rotary pumps include:

- *Nonlubricating fluids in continuous service*. Internal parts of rotary pumps must be adequately lubricated. Fluids with poor lubricating qualities, such as LPG, gasoline, and water are not usually satisfactory for rotary pumps in continuous service.
- High differential pressures and large capacity. Rotor deflection usually limits the differential pressure produced by a rotary pump. For standard designs, the larger the pump, the lower the maximum allowable differential pressure.
- *Medium-capacity and medium-head services*. Except for high viscosities, medium-capacity and medium-head services usually can be handled more economically by centrifugal or reciprocating pumps than by rotary pumps. Because of the rotary pump's close clearances and the possibility of mechanical damage, reciprocating and centrifugal pumps are usually recommended unless a rotary pump promises significant savings.
- *Abrasive material or possibility of running dry*. Rotary pumps are ordinarily not recommended for fluids containing appreciable quantities of abrasive material. However, under certain conditions, single-screw pumps with rubber liners (progressive cavity) can be used, although they may have very high maintenance requirements. In any case, rotary pumps should never be allowed to run dry.

Rotary pumps do not handle corrosive liquids well. Pumps constructed of brass and bronze are used occasionally. Stainless steel rotary pumps are not practical because of the possibility of galling or seizure.

2.3.6 Miscellaneous pumps

Table 2.6 lists pumps for special applications that do not fit into the typical centrifugal or positive displacement type.

2.4 Exercises

1. For classification purposes, all pumps can be classified as: (a)______________________
(b)__________________
2. Match the following terms:
___velocity head
___hydrostatics
___acceleration head
___static lift

Table 2.6 Miscellaneous pumps

Pump type	Characteristics
Axial flow	Also called propeller pump. Used where large capacity and low head are required. Generally with a vertical configuration for lifting wastewater, effluent, and so on
Disc friction	Also called regenerative turbine pumps. Similar to centrifugal pumps except liquid is pressured by recirculation in the impeller vanes. A low-capacity, moderate to high head pump that can handle large amounts of gas or vapor. Pump efficiency is greatly affected by internal clearances. These pumps are usually unsatisfactory where abrasives are present
Metering	Small reciprocating plunger or diaphragm pumps used for accurate pumping of chemicals and additives. Pumping rates are normally measured in gallons per hour and are adjustable from zero to full pump rate. Capable of high discharge pressures
Diaphragm	An air-operated, versatile, utility pump normally using compressed air as the driving fluid. Useful in handling hazardous or abrasive materials, and in explosive environments. Smaller units are occasionally used in metering service
Jet	Also called eductors or ejectors. Jet pumps have no moving parts and use the venturi action of high velocity fluids through a nozzle to create suction. The driving fluid and pumped fluid are mixed at the discharge. Typical applications are moving granulated solids with water, deep-well water pumping, and shipboard bilge pumping with water
Archimedes screw	Used in lifting effluent and waste water at relatively low flows where agitation and mixing are undesirable. Limited to lifts of approximately 25 ft. Similar to a screw conveyor
Peristaltic	Also called hosepumps. Peristaltic pumps are used for pumping fluids such as waste sludges, lime and cement mortar, adhesives, and shear-sensitive fluids such as latex paints. In the petrochemical industries use is limited to shear-sensitive services. The pumps have few moving parts, no seals, and can be run dry. Life is limited due to the life of the elastomer hose

___$NPSH_A$
___head
___NPSH margin
___static head
___$NPSH_R$
___hydrodynamics
___total dynamic head

- **(a)** the study of fluids at rest
- **(b)** the vertical distance from the surface of the fluid at rest to the pump datum line when the supply is below the datum line
- **(c)** $V^2/2g$
- **(d)** describes the losses associated with pulsating flow rates
- **(e)** difference between discharge head and suction head

(f) suspection head minus vapor pressure
(g) the study of fluids in motion
(h) determined by test or calculation by pump manufacturer for a particular pump
(i) the vertical distance from the surface of a fluid at rest to the pump datum line, when the supply is above the datum line
(j) $NPSH_A$ less $NPSH_R$
(k) energy content of fluid per unit of weight referred to any arbitrary datum

3. Viscosity has a significant effect on pumps ________________
   (a) warm-up
   (b) efficiency
   (c) head capability
   (d) all the above
   (e) A and B only
4. Describe the sound made by a cavitating pump.

   ____

5. When an existing pump exhibits insufficient NPSH margin, the _____________ should be checked.
   (a) discharge backpressure valve
   (b) lube oil level
   (c) suction line
   (d) low rate
   (e) A, B, and D only
6. For high TDHs and low flow rates a ________ pump is preferred, and for high flow rates and low TDHs, a _________ pump is advisable.
7. Head losses are due to:
   (a) frictional flow resistance in the pipe
   (b) valves
   (c) fittings
   (d) all the above
   (e) none of the above
8. To increase $NPSH_A$
   (a) use a booster pump
   (b) raise the liquid level in the suction vessel
   (c) raise the pump elevation
   (d) use several suction sources
   (e) A and B only

Questions 9–12

A centrifugal pump is used to pump salt water from an atmospheric storage tank with a required discharge pressure of 25 psig downstream of a filter. Use Fig. 2.24.

9. What is the minimum pump suction head? (Hint: at lowest tank level.)
   (a) 0.5 ft
   (b) 0.9 ft
   (c) 6.3 ft
   (d) 18.5 ft
   (e) none of the above

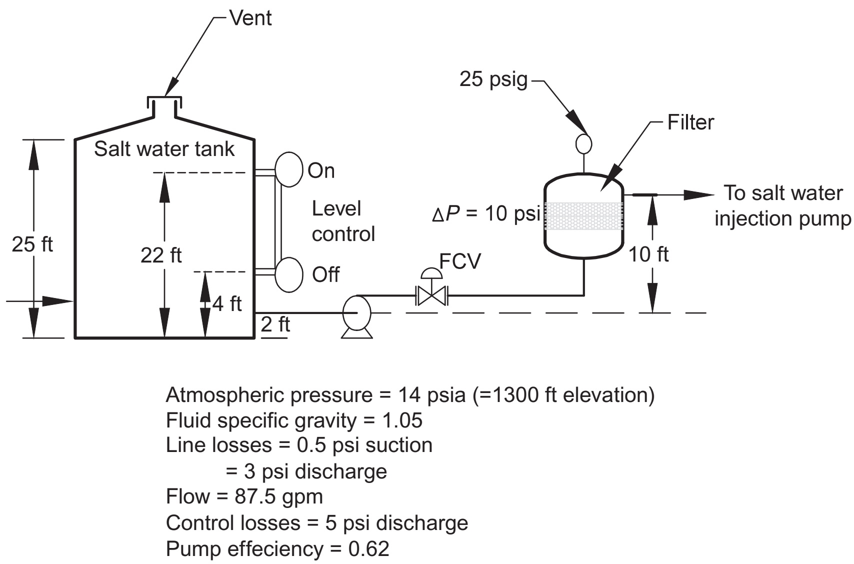


Fig. 2.24 Schematic for Problem 12.

10. What is the pump discharge head?
- **(a)** 3.1 ft
- **(b)** 23.7 ft
- **(c)** 82.6 ft
- **(d)** 104.6 ft
- **(e)** none of the above

11. What is the total dynamic head?
- **(a)** 16.7 ft
- **(b)** 42.6 ft
- **(c)** 59.9 ft
- **(d)** 103.7 ft
- **(e)** none of the above

12. What is the required pump horsepower?
- **(a)** 3.06 BHp
- **(b)** 3.88 BHp
- **(c)** 5.37 BHp
- **(d)** 8.24 BHp
- **(e)** 11.32 BHp

13. What are the two major groups that pumps fall into? Name an example from each group.

14. What are pumps commonly rated by?

15. True or false. Centrifugal pumps use pressure to build velocity following Bernoulli's 18th century hydraulic system laws.

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Centrifugal pumps

3

Nomenclature

A	plunger or piston cross-sectional area, mm^2 ($in.^2$)
a	piston rod cross-sectional area, mm^2 ($in.^2$)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
D	pump displacement, m^3/h (gpm)
d	displacement per pumping chamber, m^3 (gal)
d'	pump piston or plunger diameter, mm (in.)
d_w	diameter of wrist pin, mm (in.)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/s
g	acceleration due to gravity, 9.81 m/s^2 (32.2 ft/s^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
H_f	pipe friction loss, m (ft)
H_p	total head required for pump, m (ft)
HHP	hydraulic horsepower, kW (hp)
K	a factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in.)
m	number of pistons, plungers, or diaphragms
NPSH	net positive suction head, m (ft)
$NPSH_A$	net positive suction head available, m (ft)
$NPSH_R$	net positive suction head required, m (ft)
n	stroke rate or crank revolutions per min, rps (rpm)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
ΔP	static pressure, kPa (psi)
Q	flow rate, m^3/h (ft^3/s)
Q'	flow rate, m^3/h (BPD)
q	flow rate, m^3/h (gpm)
S	stress, kPa (psi)
S'	valve slip, percent
s	stroke length, mm (in.)
SG	specific gravity of liquid relative to water
V	velocity in suction line, m/s (ft/s)

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00003-X

Vol volume of surge tank, m^3 (ft^3)
(Vol)$_g$ required gas volume, m^3 (ft^3)
Z elevation above or below pump centerline datum, m (ft)
ρ density of fluid, kg/m^3 (lb/ft^3)

3.1 Engineering principles

3.1.1 Background

Fig. 3.1 is a classification of kinetic energy pumps. Centrifugal pumps are the most widely used kinetic energy type of pump. Centrifugal pumps are categorized as follows:

- Axial flow
- Mixed flow
- Radial flow

A centrifugal pump is a machine consisting of a set of rotating vanes enclosed within a housing or casing. The vanes produce pressure by accelerating the fluid to a high kinetic energy (velocity), then converting that energy to pressure. As shown in Fig. 3.2, the incoming fluid is pushed into the low-pressure area of the impeller "eye" by higher pressure in the upstream system. The fluid is then thrown outward from the eye by the vanes of the spinning impeller, slowing as the velocity is converted to pressure in the "diffuser" or "volute" (refer to Fig. 3.3). This momentum exchange provides an increase in pressure or "head." Having sufficient upstream or "suction" pressure (head) to push adequate flow into the pump is a critical design consideration. The fluid contained in the pump is forced out through the volute or housing through the discharge outlet.

A partial vacuum, at the eye of the impeller, enables atmospheric pressure to force fluid to enter the centrifugal portion of the pump (eye) impeller through the suction inlet. The impeller is rotated within the casing by the driver. Energy is imparted to the fluid by the rotating impeller that causes the fluid to move in the direction of rotation and radially outward.

Fluid is channeled through a volute (a curve that winds around and constantly recedes from a center point) that carries the fluid to a discharge outlet. The volute is a progressively expanding passageway that collects the fluid and causes velocity energy, added to the fluid by the impeller, to be converted into pressure energy that forces the fluid through the connected piping.

Differential head can be increased by either

- Turning the impeller faster
- Using a larger impeller
- Increasing the number of impellers

Impeller and fluid being pumped are isolated from the outside by packing or mechanical seals. Shaft radial and thrust bearings restrict the movement of the shaft and

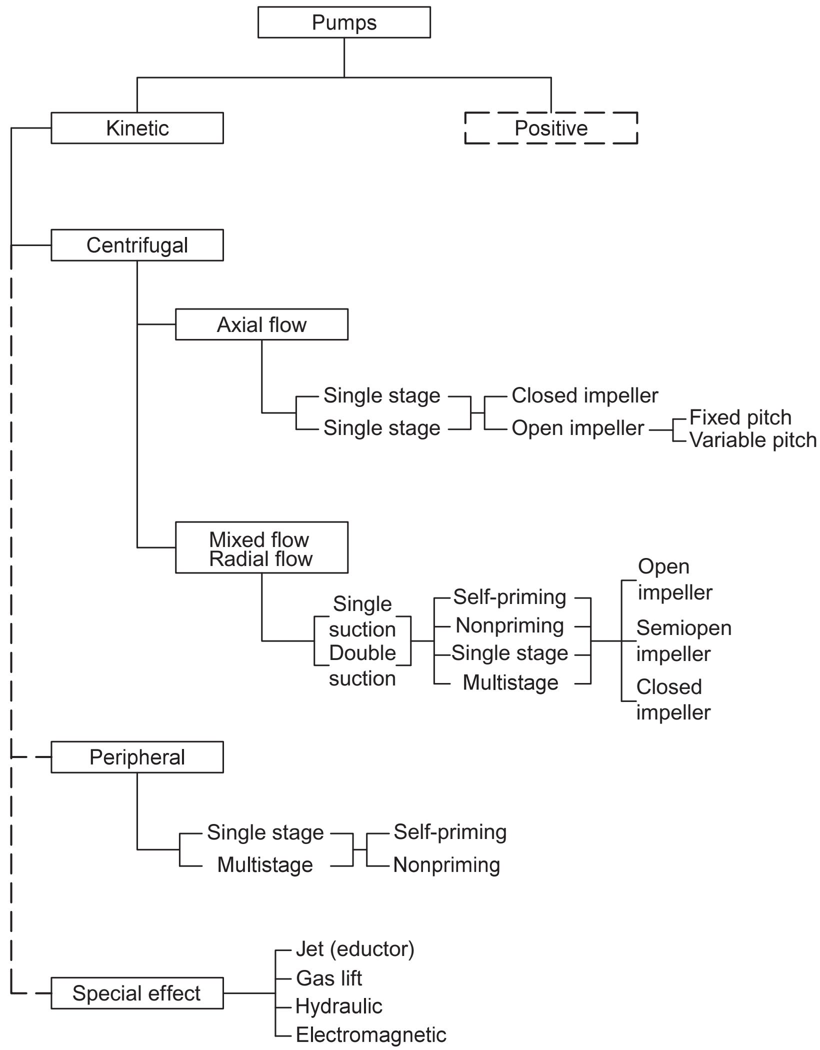


Fig. 3.1 Classification of kinetic energy pumps.
Courtesy of Hydraulic Institute.

reduce the friction of rotation. If downstream flow is restricted, some of the kinetic energy is converted to heat rather than pressure.

As shown in Fig. 3.4, a centrifugal pump has two main parts:

- *Rotating element*—including an impeller and a shaft, and
- *Stationary element*—made up of a casing, stuffing box, and bearings. The structural details of these elements and all refinements are covered in the remainder of this chapter.

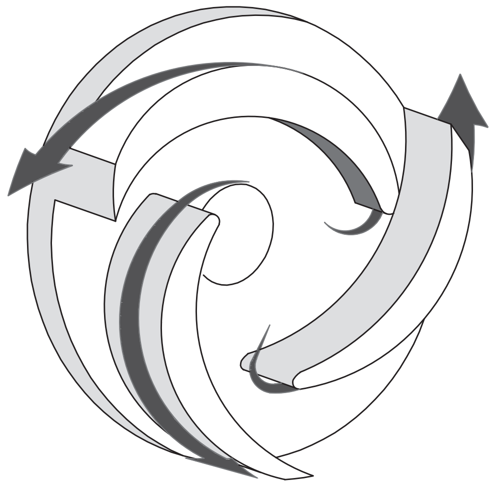


Fig. 3.2 Centrifugal force moves the liquid away from a low-pressure area formed (eye).

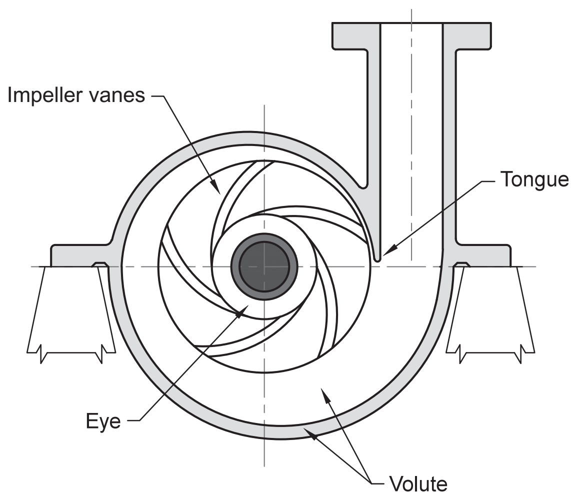


Fig. 3.3 End view of a centrifugal pump.

3.1.2 Service conditions

When selecting a centrifugal pump the most important information required by the manufacturer is the desired capacity and the head against which the pump will be required to operate while delivering the required flow rate.

3.1.2.1 Capacity

The unit of capacity for centrifugal pumps varies with the application of the pump as well as the design standards of the country where the pump will be used, for example:

Field units: Gallons per minute (gpm)
SI units: Cubic meters per hour (m^3/h)

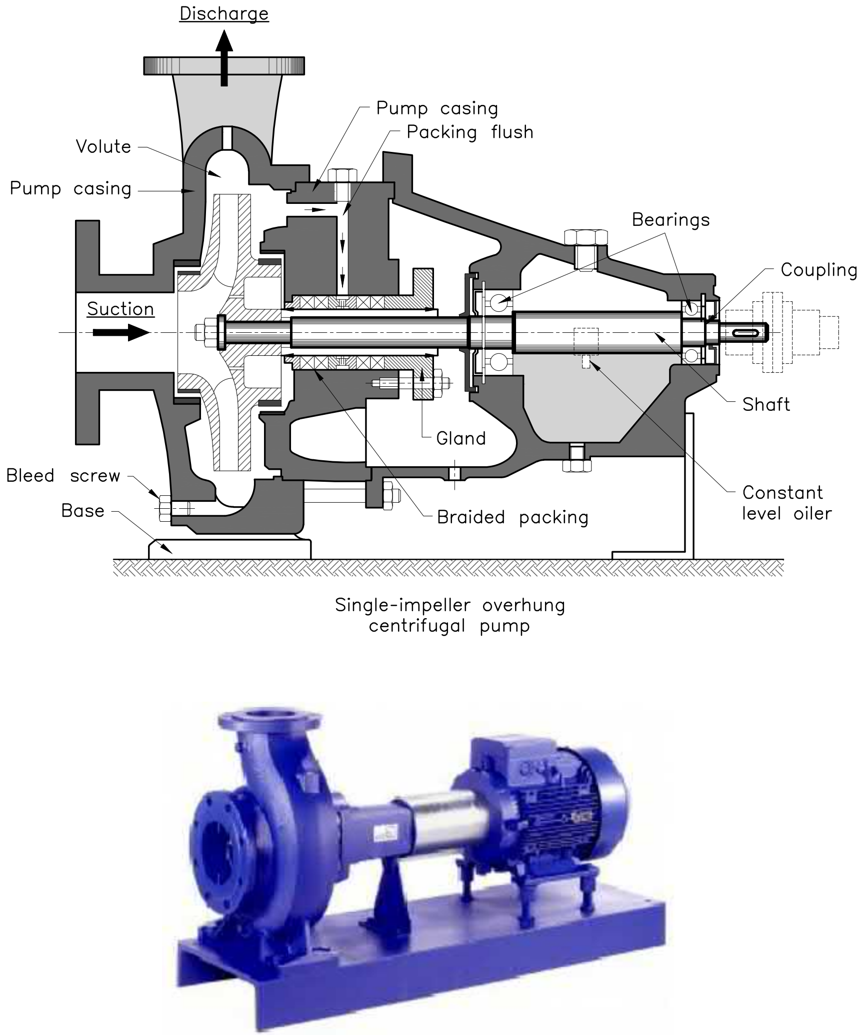


Fig. 3.4 Major components of a horizontal, single-stage, end suction, top-discharge ANSI class centrifugal pump with an overhung impeller.

In upstream production operations capacity is often expressed as cubic feet per second, gallons per hour, barrels per day, and barrels per hour. It is easy to convert the various units into gallons per minute (gpm). Table 3.1 is a conversion chart between the various units. Fig. 3.5 is a direct capacity conversion chart based on the data in Table 3.1.

3.1.2.2 Head

The term "head" is used exclusively to express pressure when designing a centrifugal pump. All pump curves are calibrated to read "feet of head" as a measure of pressure. Suction, discharge pressures, and friction losses are also expressed as feet of head, not psi.

Table 3.1 Capacity equivalents

Various units	gpm
1 second-foot or cubic foot per second (cfs)	448.8
1,000,000 gallons per day (mgd)	694.4
1 imperial gallon per minute	1.201
1,000,000 imperial gallons per day	834
1 barrel (42 gal) per day (bbl/day)	0.0292
1 barrel per hour (bbl/h)	0.7
1 acre-foot per day	226.3
1000 pounds per hour (Ib/h)	2.00[1]
1 cubic meter per hour (m^3/h)	4.403
1 liter per second (1/s)	15.851
1 metric ton per hour	4.403[a]
1,000,000 liters per day = 1000 cubic meters per day	183.5

[a]These equivalents are based on a specific gravity of 1 for water at 62°F for English units and a specific gravity of 1 for water at 15°C for metric units. They can be used with little error for cold water of any temperature between 32°F and 80°F. For specific gravity of water at various temperatures.

One reason the centrifugal pump industry has settled on head, or feet, as a measure of pressure rise is that a pump will develop the same head regardless of the fluid's specific gravity. A pump that develops a column of water (SG = 1.0) 1000 ft (300 m) high will also develop a column of hydrocarbon (SG = 0.7) 1000 ft (300 m) high. Of course, the actual pressure, in psi, would be quite different between water and hydrocarbon. The pressure developed in a pump and the pressure at the bottom of a column of liquid are both proportional to specific gravity.

The concept of head is derived from the fact that a column of cold water approximately 2.31 ft (70 cm) high will produce a pressure of 1 psi (6.9 kPa) at its base. In other words, the pressure of a column of water increases 0.433 psi (3 kPa) for every foot of depth. Thus, at a depth of 10 ft (3 m), the pressure is 4.33 psi (29.41) higher than at the surface; at 100 ft (30 m), 43.3 psi (294.11) higher; at 1000 ft (294.11), 433 psi (2941.1 kPa) higher.

For liquids other than water, the column of liquid equivalent to 1-psi pressure can be calculated by dividing 2.31 by the specific gravity of the liquid. Fig. 3.6 illustrates three pumps, each designed to develop the same pressure head (in psi); consequently the head (in feet of liquid) will be inversely proportional to the specific gravity. In this illustration the friction losses, and so on, are not considered.

Fig. 3.6 illustrates the effect that specific gravity has on the height of a column of various liquids for equal pressures. Thus a pump that must handle 1.2 specific gravity brine against a 50-psi net pressure would be designed for a head of 96 ft. If the pump had to handle cold water against the same net pressure, the head would have to be 115.5 ft, whereas a pump handling 0.70 specific gravity liquid against the same net 50-psi pressure would require a head of 165 ft.

The equivalents for the conversion of various pressure and head units other than feet into feet of liquid are presented in Table 3.2. Fig. 3.7 is a quick conversion chart of pressure and heads into feet of liquid.

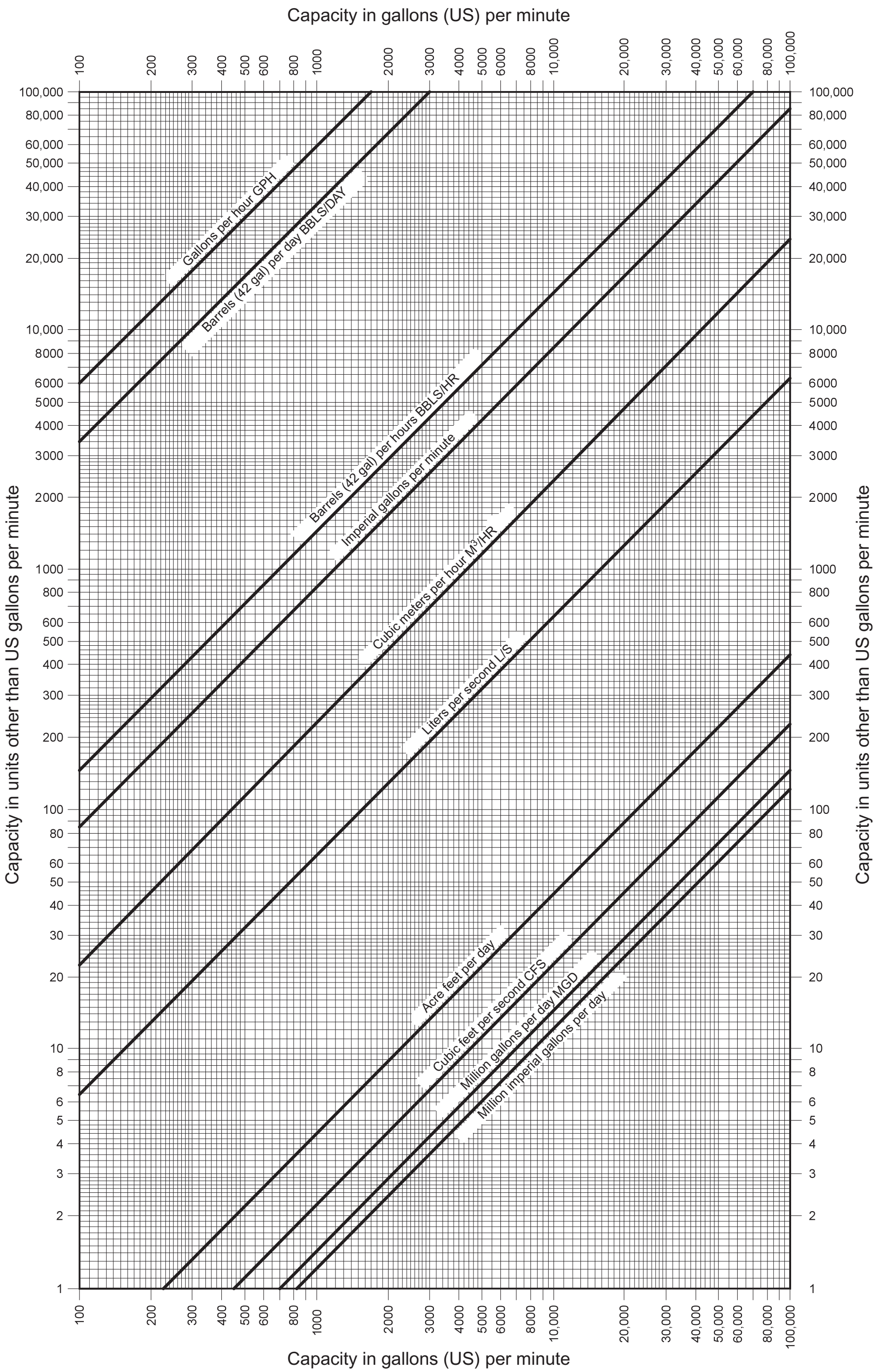


Fig. 3.5 Capacity conversion chart values based on the conversions in Table 3.1.

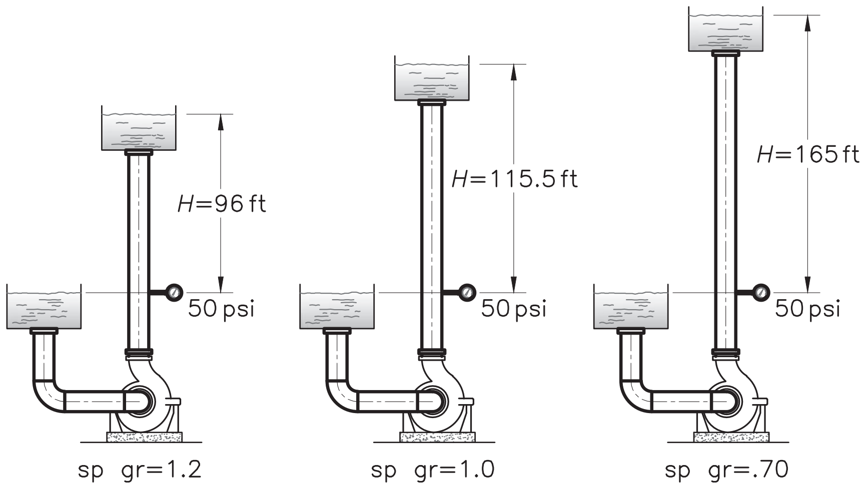


Fig. 3.6 Pressure-head relationship of pumps delivering same pressure handling liquids of differing specific gravity.
Courtesy of Ingersoll-Dresser Pumps.

Table 3.2 Pressure and head equivalents

1 Ib/sq in.	$= \dfrac{2.310}{\text{specific gravity}^a}$ ft of liquid = 2.310 ft of 62°F water
1 in.mercury (32°F)	$= \dfrac{1.134}{\text{specific gravity}^a}$ ft of liquid = 1.134 ft of 62°F water
1 atmosphere[b]	$= \dfrac{33.95}{\text{specific gravity}^a}$ ft of liquid = 33.95 ft of 62°F water
1 kilogram/sq cm	= 1 metric atmosphere
	$= \dfrac{32.85}{\text{specific gravity}^a}$ ft of liquid = 32.85 ft of 62°F water
	$= \dfrac{10.01}{\text{specific gravity}^a}$ m of liquid = 10.01 m of 15°C water
1 meter	= 3.281 ft

[a] These equivalents are based on a specific gravity of 1 for water at 62°F for English units and a specific gravity of 1 for water at 15°C for metric units. They can be used, with little error, for cold water of any temperature between 32°F and 80°F. For the actual specific gravity of water for temperatures to 220°F.
[b] Not used in conjunction with pumps.

3.1.2.3 Altitude effects on atmospheric pressure

There is a decrease in atmospheric pressure of about 1 in. of mercury (or 0.5 psi) per 1000 ft of elevation. At an elevation of 4000 ft, the atmospheric pressure is 4 in. of mercury, or 4.5 ft of water, or 2 psi less than that at sea level. A centrifugal pump will operate satisfactorily for the same maximum capacity only if the suction lift is 4.5 ft less

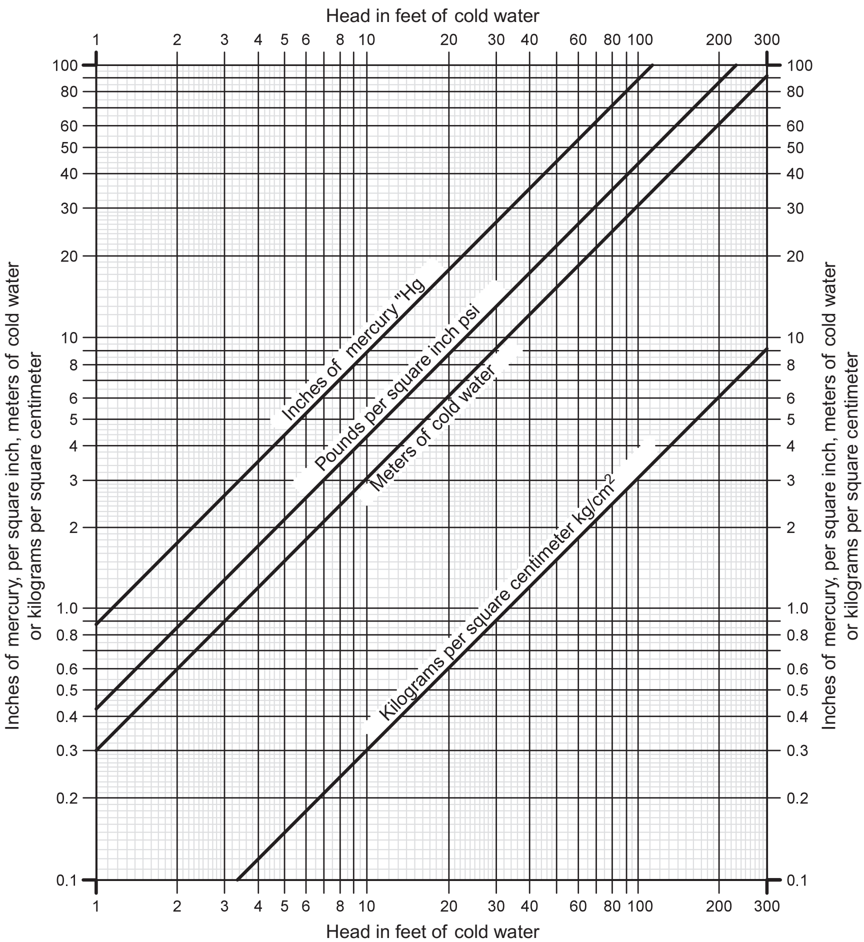


Fig. 3.7 Pressure and head conversion chart.

than that at sea level. The $NPSH_R$ does not change with elevations above sea level. However, the available atmospheric pressure is reduced. For barometric pressures at various altitudes, refer to Table 3.3.

3.1.2.4 Vapor pressure

The vapor pressure of a liquid at a given temperature is that pressure at which it will flash into vapor if heat is added to the liquid. For a single component liquid, such as water, the vapor pressure has a very definite value at any given temperature, and steam tables are available that give the vapor pressure of such liquids over a wide range of temperatures. As an example, Table 3.4 gives the relationship between water vapor pressure and temperature. Mixed liquids, such as crude oil, are made up of several

Table 3.3 Approximate atmospheric pressures and barometer readings at different altitudes

Altitude		Barometer		Atmospheric pressure ib/in.^{2}	Equivalent head of water (75°F) feet	Boiling point of water	
Feet	**Meters**	**Inches of mercury**	**Mm of mercury**			**°F**	**°C**
−1000	−304.8	31.02	787.9	15.2	35.2	2136	101.0
−500	−152.4	30.47	773.9	15.0	34.7	212.9	100.5
0	0	29.921	760.0	14.7	34.0	212.0	100.0
500	152.4	29.38	746.3	14.4	33.4	211.1	99.5
1000	304.8	28.86	733.1	14.2	32.8	210.2	99.0
1500	457.2	28.33	719.6	13.9	32.2	209.3	98 5
2000	609.6	27.82	706.6	13.7	31.6	208.4	98.6
2500	762.0	27.31	693.7	13.4	31.0	207.4	97.4
3000	914.4	26.81	681.0	13.2	30.5	206.5	96.9
3500	1066.8	26.32	668.5	12.9	29.9	205.6	96.4
4000	1219.2	25.84	656.3	12.7	29.4	204.7	95.9
4500	1371.6	25.36	644.1	12.4	28.8	203.8	95.4
5000	1524.0	24.89	632.2	12.2	28.3	202.9	94.9
5500	1676.4	24.43	620.5	12.0	27.8	201.9	94 4
6000	1828.8	23.98	609.1	11.8	27.3	201.0	93.9
6500	1981.2	23.53	597.7	11.5	26.7	200.1	93.4
7000	2133.6	23.09	586.5	11.3	26.2	199.2	92.9
7500	2286.0	22.65	575.3	11.1	25.7	198.3	92.4
8000	2438.4	22.22	564.4	10.9	25.2	197.4	91.9
8500	2590.8	21.80	553.7	10.7	24.8	196.5	91.4
9000	2743.2	21.38	543.1	10.5	24.3	195.5	90.8
9500	2895.6	20.98	532.9	10.3	23.8	194.6	90.3
10,000	3048.0	20.58	522.7	10.1	23.4	193.7	89.8
15,000	4572.0	16.88	428.8	8.3	19.1	184	64.4
20,000	6096	13.75	349.3	6.7	15.2	–	
30,000	9144	8.88	225.6	4.4	10.2	–	
40,000	12192	5.54	140.7	2.7	6.3	–	
50,000	15240	3.44	87.4	1.7	3.9	–	

components, each having its own vapor pressure, and a partial vaporization may take place at various pressures and temperatures.

When determining the heads for pumps, it is important that pressures expressed in pounds per square inch be converted into feet of liquid at the pumping temperature. Do not use conversion factors applying to other temperatures for such conversions. For example, refer to Fig. 3.8, the vapor pressure of water at 212°F (100°C) is 14.7 psia (1 bara) at sea level. The equivalent head in feet of water is 33.9 ft of 62°F water. At 212°F (100°C) water has a specific gravity of 0.959 compared to a gravity of 1.0 for 62°F water, its equivalent head would be 33.9/0.959 = 35.4 ft.

Table 3.4 Temperature-vapor pressure relationship of water

Temperature (°F)	Vapor pressure (psia)	Temperature (°F)	Vapor pressure (psia)	Temperature (°F)	Vapor pressure (psia)
32	0.088	232	21.58	400	247.3
35	0.100	236	23.22	405	261.7
40	0.122	240	24.97	410	276.8
45	0.148	244	26.83	415	292.4
50	0.178	248	28.80	420	308.8
55	0.214	252	30.88	425	325.9
60	0.256	256	33.09	430	343.7
65	0.306	260	35.43	435	362.3
70	0.363	264	37.90	440	381.6
75	0.430	268	40.50	445	401.7
80	0.507	272	43.25	450	422.6
85	0.596	276	46.15	455	444.3
90	0.698	280	49.20	460	466.9
95	0.815	284	52.42	465	490.3
100	0.949	288	55.80	470	514.7
105	1.102	292	59.36	475	539.9
no	1.275	296	63.09	480	566.1
115	1.471	300	67.01	485	593.3
120	1.692	304	71.13	490	621.4
125	1.942	308	75.44	495	650.6
130	2.222	312	79.96	500	680.8
135	2.537	316	84.70	505	712.0
140	2.889	320	89.66	510	744.3
145	3.281	324	94.84	515	777.8
150	3.718	328	100.3	520	812.4
155	4.203	332	105.9	525	848.1
160	4.741	336	111.8	530	885.0
165	5.335	340	118.0	535	923.2
170	5.992	344	124.4	540	962.5
175	6.715	348	131.2	545	1003
180	7.510	352	138.2	550	1045
185	8.383	356	145.4	555	1088
190	9.339	360	153.0	560	1133
195	10.385	364	160.9	565	1179
200	11.526	368	169.2	570	1226
204	12.512	372	177.7	575	1275
208	13.568	376	186.6	580	1326
212	14.70	380	195.8	585	1378
216	15.90	384	205.3	590	1431
220	17.19	388	215.3	595	1486
224	18.56	392	225.6	600	1543
228	20.02	396	236.2		

At sea level, the saturation pressure or vapor pressure (PSIG) = vapor pressure (PSIA—14.7).

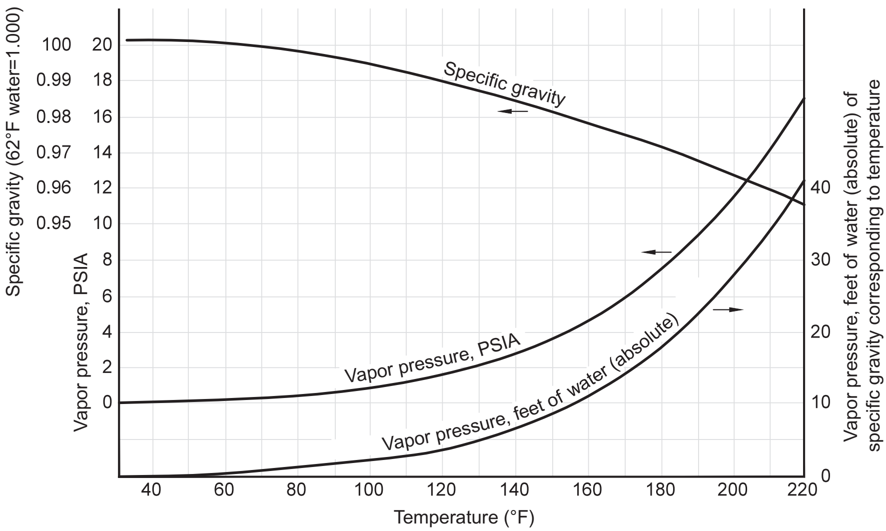


Fig. 3.8 Specific gravity, temperature, and vapor pressure for water.

3.1.3 Temperature rise

The difference between the hydraulic horsepower (HHP) and the break horsepower (BHP) represents the power lost to the bearings and seals (small) and mechanical components. Power losses are transferred to the fluid being pumped causing its temperature to increase. The addition of heat:

- Increases the vapor pressure
- Decreases $NPSH_A$
- Reduces the fluid density
- Causes thermal expansion

To avoid overheating, the temperature rise should be limited to:

- 15°F (−9.4°C): Water
- 10°F (−12.2°C): Hydrocarbons
- 5°F (−15°C): Cryogenic service

The actual temperature rise can be calculated from the following equation:

$$\Delta T = \frac{(33,000)(BHP)(1-e)}{(778)(8.35)(q)(SG)(C_P)} \tag{3.1}$$

Where:

ΔT = Actual temperature rise, °F (°C)
BHP = Break horsepower, Hp (kW)
e = Pump efficiency
q = Pump capacity, gpm (m^3/h)
SG = Fluid specific gravity
C_P = Specific heat of fluid, BTU/lb/°F

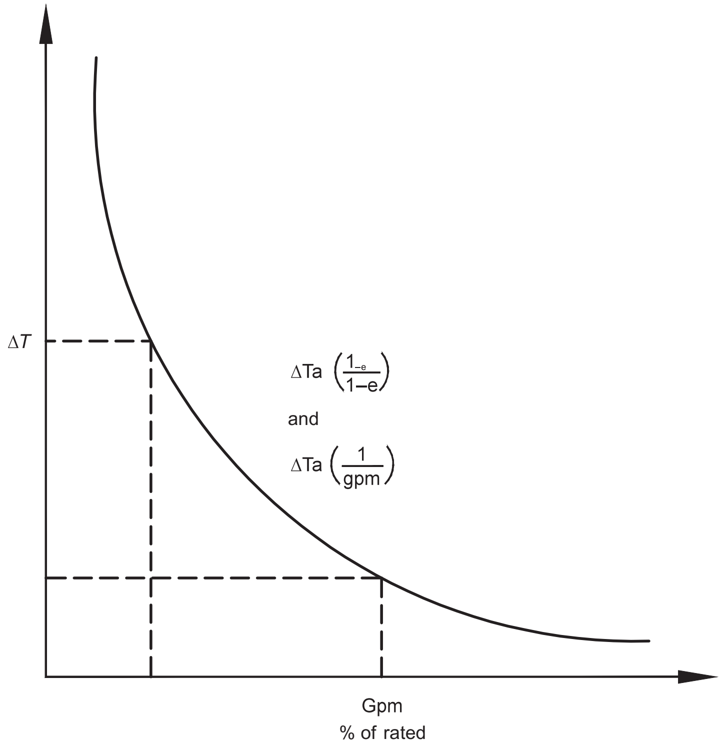


Fig. 3.9 Temperature versus GPM.

Fig. 3.9 illustrates graphically the relationship shown in Eq. (3.1), where temperature rise is directly proportional to the inefficiency and inversely proportional to the flow rate.

3.1.4 Shaft torque

Shaft must be capable of supplying as much torque as required by the pump at each successive speed from zero to full load in order to reach full load. The value of shaft torque can be calculated from the following equation:

$$T_g = \frac{5250(BHP)}{N} \tag{3.2}$$

Where:

T_g = Shaft torque, ft-lbs
BHP = Break horsepower, Hp
N = Pump rotating speed, rpm

Theoretically, at zero speed the torque would be zero, but the shaft must overcome friction and inertia. The diameter of the shaft can be determined by the following equation

$$d_s = 0.75\left(\frac{q}{N}\right)^{0.5} \tag{3.3}$$

Where:

d_s = Shaft diameter, inches (mm)
q = Pump capacity, gpm (m^3/h)
N = Pump rotating speed, rpm (rps)

3.2 Process piping considerations

3.2.1 General considerations

A centrifugal pump is a piece of precision machinery that must not be subjected to external strains beyond those it was designed to encounter. It must be installed in the intended position, carefully aligned, and free from piping forces and moments.

Generally, foundation design is not critical. Vibrations in a centrifugal pump are minimal unless an engine driver is used. As a general rule of thumb, the foundation should be able to handle three times the weight of the pump, driver, and skid assembly. The manufacturer is the best source for determining what size foundation is required.

3.2.2 General piping design

Poor piping design and installation is a common cause of poor centrifugal pump performance or failure. Poor piping can result in the following:

- Cavitation
- Performance dropout
- Impeller failure
- Bearing and mechanical seal failure
- Cracked casings or foundation bolts
- Leaks, spills, and fires

Piping is often dismissed on the assumption that everyone "knows how to do it." Unfortunately, piping cannot be dismissed because inattention, even to small details, can lead to significant problems. Thus the anticipation of the potential problems during the design stage is very important.

Fig. 3.10 shows a typical piping and instrument diagram (P&ID) or mechanical flow diagram (MFD) for two centrifugal pumps installed in parallel, illustrating some of the considerations necessary for a good pump installation. Each installation has different objectives and very few will be exactly like the one shown in Fig. 3.10.

Each pump has isolation valves to allow for maintenance while the other pump is running. Since it is possible that a discharge valve can be left open while a suction

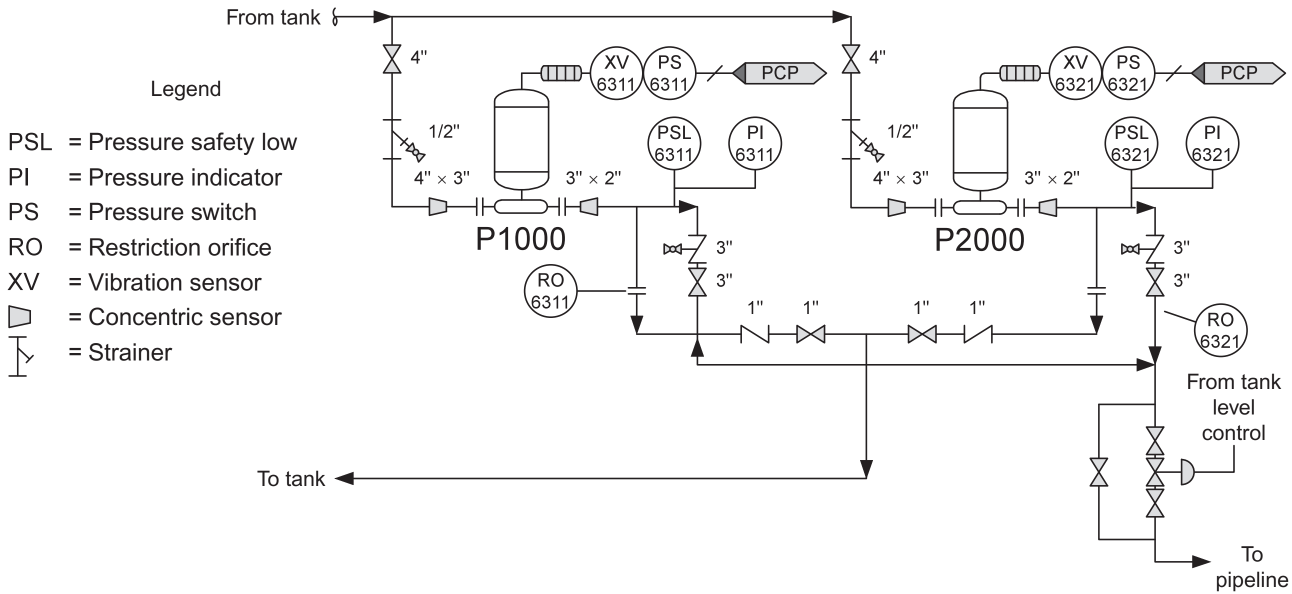


Fig. 3.10 Typical P&ID or MFD of two centrifugal pumps installed in parallel.

valve is closed, the suction line up to and including the block valve should be rated for discharge pressure. As an alternative, a pressure safety valve (PSV) can be installed in each pump's suction piping.

Each pump's discharge has a flow safety valve (FSV) or check valve to prevent the pump's reverse rotation when it is not operating and its discharge valve is open. Selection should take into account the effect of water hammer. Water hammer is the transient change in static line pressure as a result of a sudden change in flow. Items that can initiate the sudden change in flow include the starting or stopping of a pump, or the opening or closing of a check valve. Slow closing check valves are acceptable on systems with a single pump and long lengths of pipe. Fast closing check valves are required with multiple pumps operating in parallel and at high heads. As a general guideline, lift check valves are slow unless they are spring loaded. Tilting-disc check valves are fast closing but are more expensive. When fast reacting check valves are required, pressure drop considerations should be secondary.

A throttling valve is installed to control flow without having to start and stop the pumps. It is also possible to start and stop one or both pumps based on level in the feed tank or to use a variable speed driver to control pump speed.

Since it is possible for the throttling valve to close or for a pump to be started with a blocked discharge, a minimum flow bypass with an orifice sized to provide sufficient flow to avoid overheating the pump installed. The pump manufacturer will specify minimum flow requirements, which are normally 5%–10% of the design flow rate. The bypass is piped back to the tank or vessel for further cooling. It is possible to pipe the recycle directly back to the pump suction. The drawback of this type of bypass is that, with the short loop from discharge to suction, eventually the liquid will overheat. The minimum flow bypass volume must be added to the design flow when a pump capacity is specified.

The continuous bypass effectively reduces the efficiency of the pump. On large installations it can be attractive to install a pressure control valve (PCV) on the pump discharge; this valve would bypass liquid only when the pump discharge pressure approaches the minimum flow head. Other alternatives include minimum flow switches or flow safety low (FSL), pressure safety high (PSH) shutdowns, and so on. There is much debate on which scheme to use.

Operations often install a pressure safety low (PSL) on the suction or the discharge to protect the pump from running dry. The pump is provided with vent and drain connections. The vent should be opened before start-up. A cone-type strainer may be used during initial start-up. Piping must be arranged to allow removal and replacement of the strainer and to allow easy access to the pump for maintenance.

3.2.3 Suction piping

Suction piping is more important than discharge piping. When the fluid source is above the pump (static head), the source vessel or tank should contain the following:

- *Weir plate* to minimize turbulence
- *Vortex breaker* to eliminate "vortexing" (Fig. 3.11) and vapor entrainment, and
- *Nozzle* sized to limit exit velocity to 6–9 ft/s (1.8–2.7 m/s), or less.

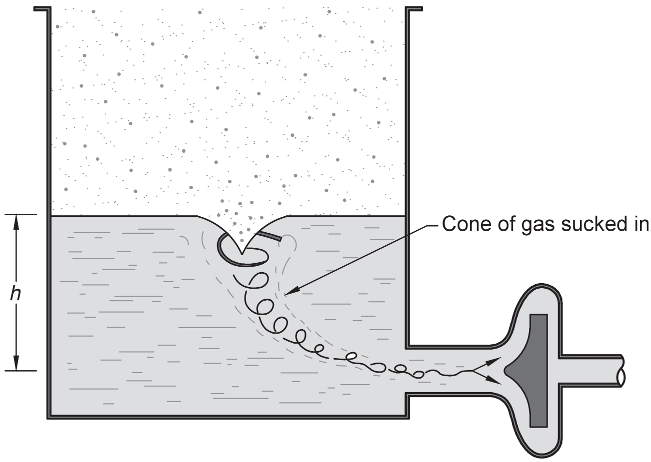


Fig. 3.11 Suction tank experiencing "vortexing."

When the fluid source is below the pump (static lift), the sump, basin, or pit should be:

- Designed to provide even velocity distribution in the approach or around the suction inlet
- Sufficiently submerged to prevent "vortexing"

The suction and discharge piping should be short, with minimum possible number of elbows and fittings. Suction piping should be at least one "nominal" pipe size larger than the pump suction flange. Where possible, pipe should be laid out using 45-degree ells, rather than 90-degree ells, for elevation and plane changes. If 90-degree ells are used, they should be long radius type. Suction pipe diameter changes should be made with eccentric reducers (with the flat side up (FSU)) to eliminate gas pockets where suction is lower than the liquid receiver pump and cause cavitation. Concentric reducers must not be used for horizontal suction lines because they trap vapor that can be pulled into the pump and cause cavitation or "vapor lock." The only time that concentric reducers can be used is for vertical suction lines and horizontal lines with flooded suction. Fig. 3.12 illustrates recommended and nonrecommended suction piping installations.

To avoid abrupt changes in velocity, reducers should be limited to a change in diameter of one line size. Two elbows in series should be avoided if possible. Elbows should not be connected directly to the pump suction flange and a minimum of at least two to five pipe diameters of straight pipe should be between the suction flange and the elbow and between successive elbows. This reduces swirl and turbulence before the fluid reaches the pump. Otherwise, separation of the leading edges may occur with consequent noisy operation and cavitation damage.

Conditions may dictate that permanent strainers be installed in the suction piping. If permanent strainers are not required, temporary cone-type strainers should be installed at least for initial start-ups. Basket trainers should have at least 150% flow area screens.

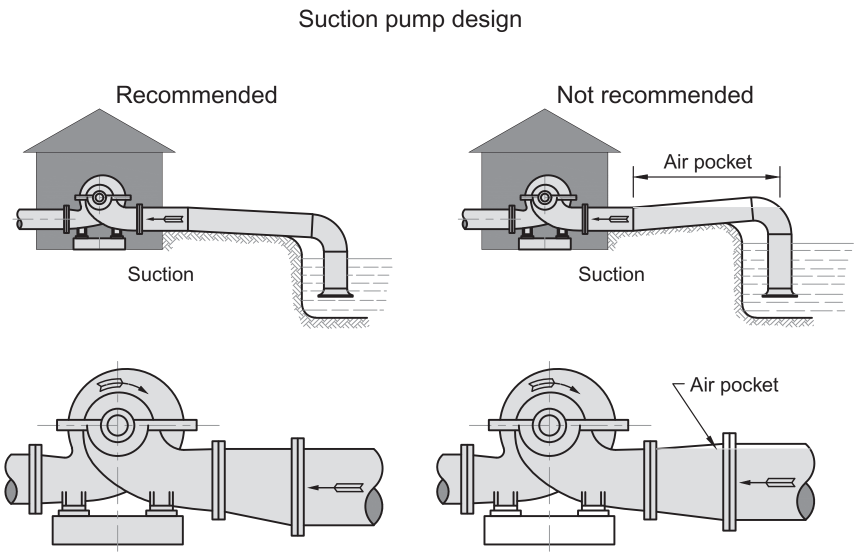
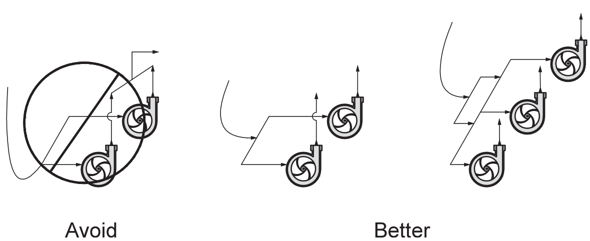


Fig. 3.12 Examples of recommended and nonrecommended suction piping installations.

For flooded suctions, piping should be continuously sloping downward to the pump suction so that any vapor pockets can migrate back to the source vessel. For static lifts, the piping should be continuously sloping upward with no air pockets (install gate valves in horizontal positions). Where air pockets cannot be avoided, the use of automatic vent valves is recommended.

If two or more pumps are installed in parallel it is best to install separate suction lines between the tank and the individual pumps. However, in most cases this is not practical, and the suction lines are often manifolded together. If this is done, the line should be sized so that the velocity in the common feed line is approximately equal to the velocities in the lateral lines feeding the individual pumps. Special attention must be given to successive right angle turns. Fig. 3.13 shows suction piping details when two or more pumps are installed in *parallel*. Two successive 90-degree turns made in planes at right angles to each other should be avoided and replaced with two successive turns made in the same plane.

Fig. 3.13 Suction piping when two or more pumps are installed in parallel.



Pump inlet size has no bearing on the required piping size. The available suction head and the suction requirement and losses must be considered, and piping should be sized on this basis regardless of pump inlet size.

The suction outlet of the supply vessel should be slightly higher than the pump inlet so that gases accumulating in the system may flow back to the vessel rather than through the pump.

The suction piping should be sized for no more than 2–3 ft/s (0.6–1 m/s) flow velocity and head loss due to friction less than one foot (30 cm) per 100 ft (30 m) of equivalent piping length, and the discharge piping for no more than 6–9 ft/s (1.8–2.7 m/s) flow velocity. Although suction and discharge velocities are not as critical for centrifugal pumps as for reciprocating pumps. Field experience indicates lower maintenance when the velocities are kept below these ranges. A low flow velocity for the suction piping is particularly important. Table 3.5 summarizes suction piping design inadequacies.

3.2.4 Discharge piping

When designing a discharge piping system for centrifugal pumps one should avoid sharp bends, reducers, valves with less than a full opening, and so on, near the pump; these may reflect pressure surges back toward the machine. Manifolds in which two or more pumps are tied into a common system should be located as far as possible from the pumps to allow for dampening of surges. For most applications, 100–150 ft (30–45 m) is adequate.

Discharge piping should be securely anchored as close to the pump as practical to prevent system vibrations from acting directly on the pump. Pressure relief valves

Table 3.5 Piping design inadequacies

Root causes	Related problems	Symptoms
Piping nozzle loads	Misalignment	1× and 2× vibration
Concentric elbow in horizontal run	Vapor pocket in suction line	Difficulty priming the pump and/or cavitation
Incorrectly oriented eccentric elbow in horizontal piping run	Vapor pocket in suction line	Difficulty priming the pump and/or cavitation
Upward looping line	Vapor pocket in suction line	Difficulty priming the pump and/or cavitation
Line sloped incorrectly	Vapor pocket in suction line	Difficulty priming the pump and/or cavitation
Flow discontinuity too close to pump suction (e.g., elbow, tee)		Varying degrees of performance impairment depending on type of pump, impeller, and so on
Asymmetric suction piping to pumps running in parallel		Suction stealing, low flow recirculation, cavitation, vibration
Elbow in parallel plane to pump shaft too close to suction of pump with double-suction impeller	Abnormal thrust load	Performance impairment, cavitation, vibration, bearing failures

(PSVs) should be installed in the discharge piping near the pump and certainly upstream of the first block valve.

Discharge/outlet piping should have high enough pressure ratings for potential future needs. Consideration also must be given to discharge PSV flange ratings. PSVs must be set high enough to avoid inadvertent discharges.

3.3 Pump characteristic "performance" curves

3.3.1 Overview

Unlike positive displacement pumps, a centrifugal pump operating at constant speed can deliver any capacity from zero to a maximum value dependent upon the pump size, design, and suction conditions. The total head developed by the pump, the power required to drive it, and the resulting efficiency vary with the capacity.

When a pump manufacturer develops a new pump the pump is tested for performance under controlled conditions. The results are plotted in the form of graphs that show flow rate versus

- Head
- Power Consumption
- Efficiency
- $NPSH_R$

The interrelationships of capacity, head, power, and efficiency are called the pump characteristics. These interrelationships are best shown graphically, and the resulting graph is called the pump's "characteristic" or "performance" or "head-capacity (H-Q)" or "pump curves" (Fig. 3.14). At constant speed, as the head required to be furnished by the pump increases, the flow rate decreases. Fig. 3.14 also shows values to develop the curve from "shutoff" head to maximum flow, and the pump efficiency curve. Fig. 3.14B shows the difference between a theoretical and an actual pump curve due to losses due to shock, turbulence, recirculation, and friction. As shown in Fig. 3.14B, the actual performance is less than design and will always be curved, rather than straight line performance. The losses will vary on each pump, but all will be affected by shock, turbulence, recirculation, and friction. Every pump manufacturer will provide certified test curves on each pump. Most manufacturers make their pumps and impellers to comply with the Hydraulic Institute standard for pump performance.

Some applications require each pump to be provided with a performance test to provide information of actual pump performance across a complete range of given conditions. Each test will provide a large range or performance points across the full range of performance/data points across the full range of values to develop the curve from shutoff head to maximum flow.

For a given impeller shape, the efficiency is a maximum, called the best efficiency point (BEP), at a design throughput rate (refer to Fig. 3.15). As the rate varies upward and downward from this point, the efficiency decreases. Under similar operating conditions, an installed pump is expected to demonstrate the same performance

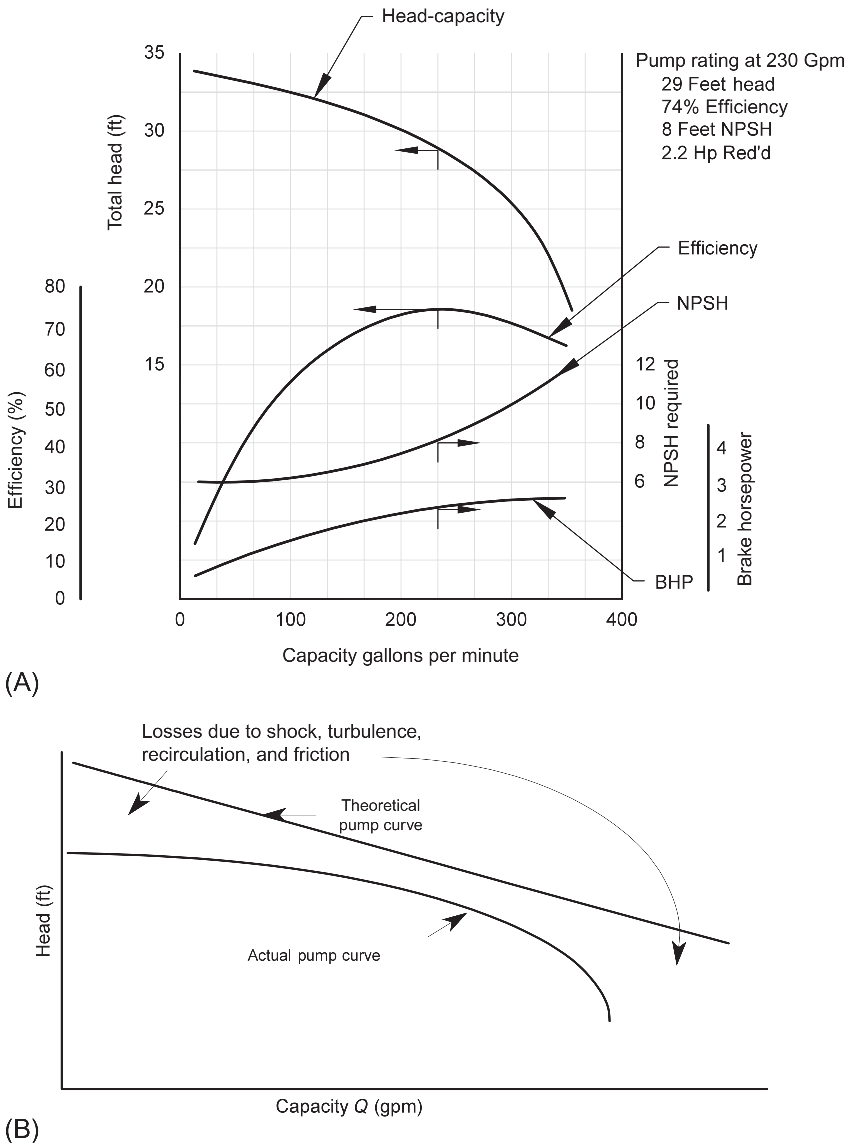
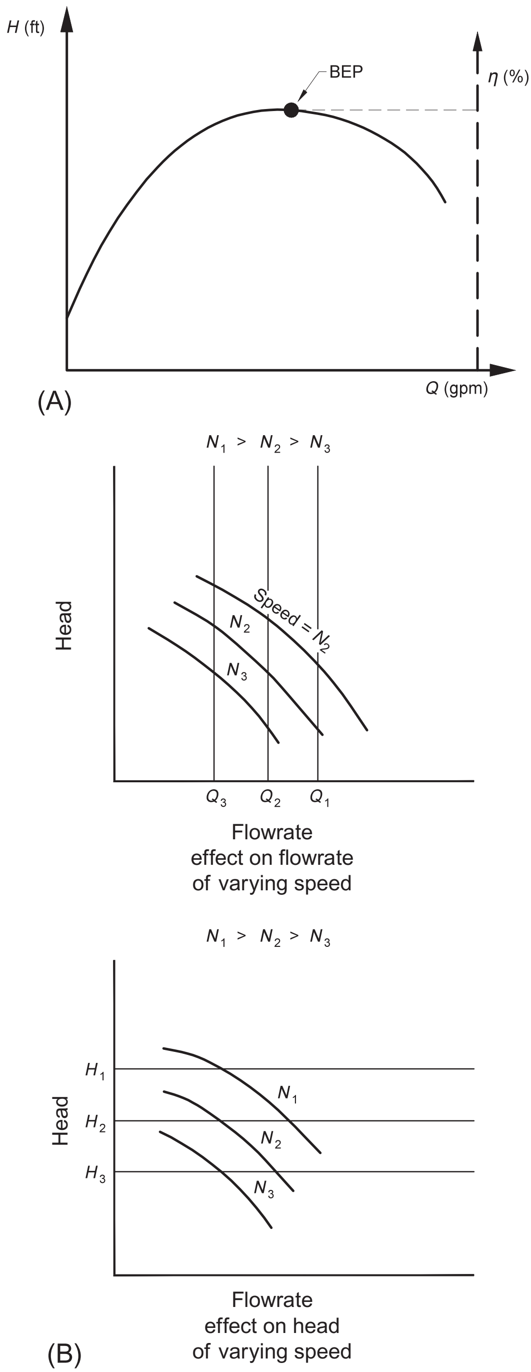


Fig. 3.14 (A) Typical centrifugal pump characteristic or "performance" curves. (B) Difference between a theoretical and an actual pump curve due to the shock, turbulence, recirculation, and friction.

characteristics as shown on its performance curves. If it does not, indicates that something is wrong with the system and/or pump. Comparison of actual pump performance with rated performance curves can aid in the determination of a pump malfunction.

By varying the pump speed the throughput at a given head or the head for a given throughput can be changed. In Fig. 3.15B, as the speed decreases from N_1 to N_2 to N_3,

Fig. 3.15 (A) Head-capacity curve showing the best efficiency point (BEP). (B) Graphs showing the effects on flow rate and head when the pump speed changes.



the flow rate decreases if the head required is constant, or the head decreases if the flow rate is constant.

The centrifugal pump's rotating impeller primarily determines the pump's performance. Head, $NPSH_R$, efficiency, horsepower, and power requirements vary with capacity. Fig. 3.16A and C shows the effects of impeller size on the pump's characteristic curve.

Most performance curves will show a number of important points of performance. Some of these points are as follows:

- Maximum and minimum impeller diameters available
- Efficiency of the impellers at different points
- Brake horsepower requirements at different points
- Model number
- Suction and discharge size
- RPM of driver required
- GPM range or each diameter impeller
- Total head rating of each curve
- NPSH rating at various flows

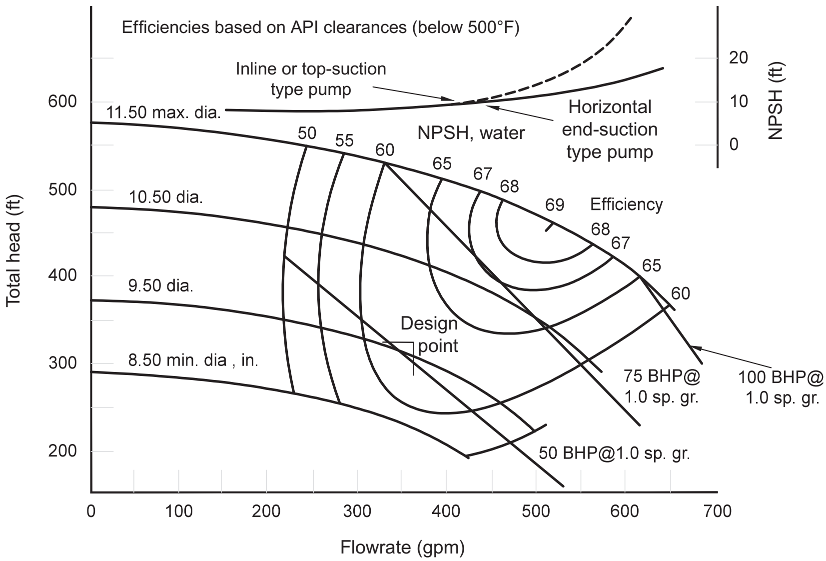
Each manufacturer will provide a performance curve for every pump they manufacturer.

Ideally, it would be best to operate the pump at the BEP, but this is not normally feasible. Alternatively, the pump should operate only in the area of the curve closest to the BEP, and only in the moderately sloping portion of the head curve. Operating in the flat or steeply sloping portions of the curve results in wasted energy and flow control instability. Pumps that run at or near this BEP will run smoother and have better run lives. Any time the actual flow drops below 50% of the BEP flow, it is wise to consult the manufacturer. Shaft deflections may increase dramatically (especially with single-stage overhung design pumps), which may lead to higher maintenance costs and failures.

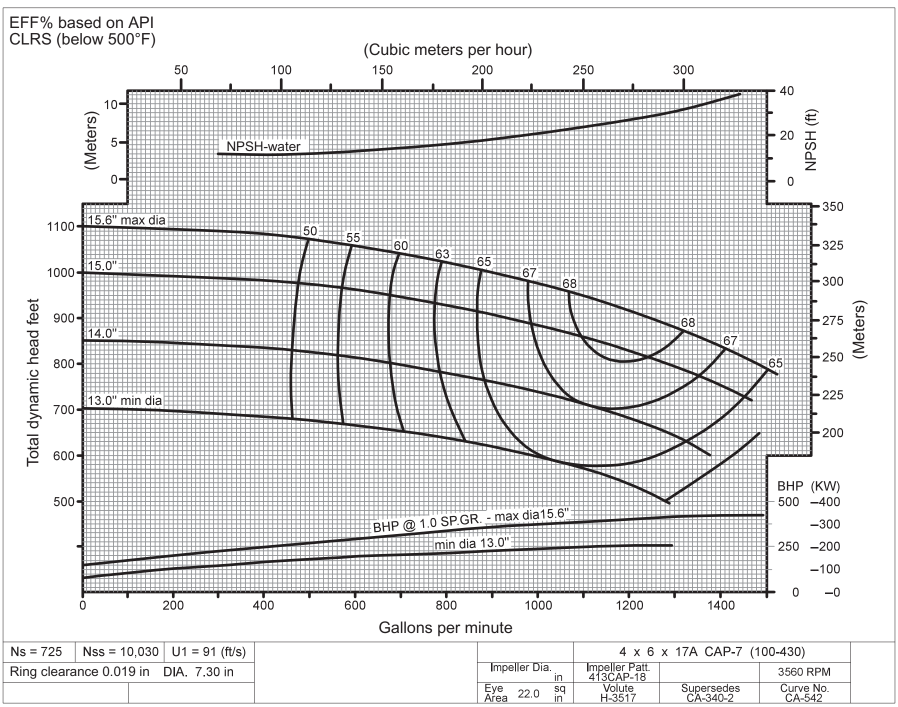
3.3.1.1 Total developed head

The total developed head (TDH) is a measure of the energy a pump delivers to a fluid. It is equal to the total discharge head minus the total suction head in feet (m) of liquid. The word "total" is used because each of these heads is composed of the pressure head, velocity head, static head, and head loss. The TDH can be approximated by measuring the suction and discharge pressures at the pump flanges, subtracting the suction pressure from the discharge pressure (both in units of in units of psig), and then converting to units of head in feet (m). The approximation neglects the velocity head component, which usually results in an error of less than 1%.

The TDH depends on the impeller diameter, pump speed, fluid viscosity, impeller and case design, and pump mechanical condition. It also varies with capacity, largely due to frictional losses in the impeller and casing. Total head is greatest at zero capacity (shutoff head) and falls off with increasing flow rates.



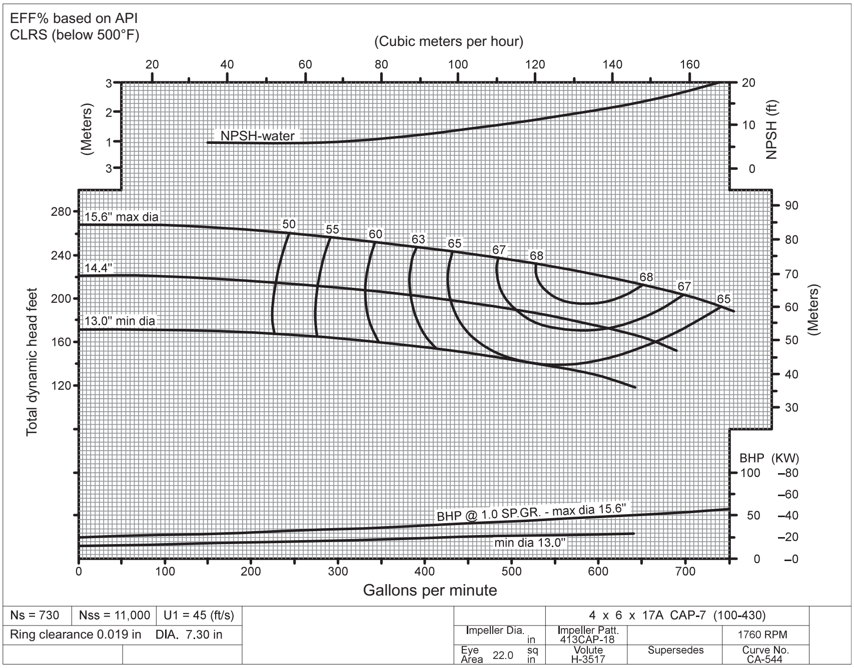
(A)



(B)

Fig. 3.16 (A) Effects of changing the impeller size. (B) Typical manufacturer's performance curves. Courtesy of Sulzer Pumps, United States.

(Continued)



(C)

Fig. 3.16— Cont'd (C) Typical manufacturer's performance curves.
Courtesy of Sulzer Pumps, United States.

3.3.1.2 Power

The power (BHP) curve, starts out at some small value at zero flow, increases moderately up to a point where it reaches a maximum and may even taper off slightly.

3.3.1.3 Efficiency

The pump efficiency curve starts out at zero, increases rapidly as flow increases, levels off at BEP, and decreases in value.

3.3.1.4 $NPSH_R$

The $NPSH_R$ is a finite value at zero flow and increases as a square function of the increase in flow rate.

3.3.2 Head-capacity characteristic curve classifications

Every centrifugal pump will operate on its characteristic curve if there is enough $NPSH_A$ for a given specific gravity and viscosity. For any given capacity, there will be one total head rise, one efficiency, one horsepower, and one $NPSH_R$.

In general, it is desirable to choose a pump that operates at its maximum efficiency point or slightly to the left; however, this is not always possible. Pumps are sold to operate over wide ranges, even at the extreme ends of the curve. Normally, if the $NPSH_A$ is sufficient to prevent cavitation, the pump should operate satisfactory.

The slope and shape of the head-capacity characteristic curve is affected by individual pump design. Centrifugal pump head-capacity characteristic curves are classified as follows:

3.3.2.1 Steady rising

Fig. 3.17 illustrates a head-capacity curve where the head rises continuously as the capacity is decreased. As a rule of thumb, steady "rising" curves are those showing a 10%–25% increase in head between the capacities of peak efficiency and shutoff. Pumps with this type of curve have continuously rising head as capacity is decreased. Pumps with this characteristic curve are often used in parallel operations; because of their stability they continue to operate at lower and lower throughput volumes.

Centrifugal pumps with steady-rise curves are most commonly used. Since the head varies distinctly with a change in capacity, precise flow control can be maintained with this type of curve. The rising curve is a stable curve; for every head, only one corresponding capacity occurs.

3.3.2.2 Drooping

Fig. 3.18 illustrates a head-capacity curve where the head developed at shutoff is less than that developed at some other capacity. These curves are characterized by multiple flow points for a given head. As can be seen from the example curve, there are two stable head points for some capacities. Pumps designed to deliver maximum head per inch of impeller diameter generally have this characteristic. They are almost never used in general oil field pumping applications.

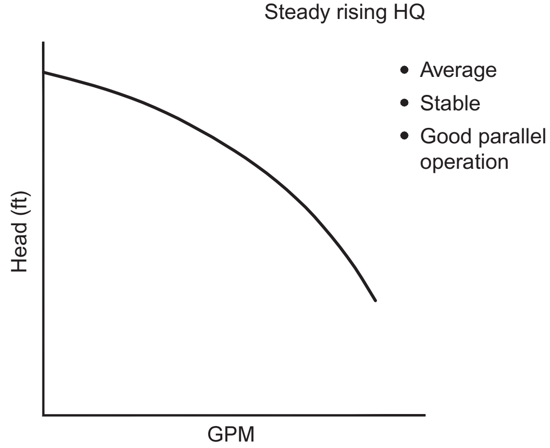


Fig. 3.17 Typical pump curve with a "steady-rising" head versus capacity characteristic.

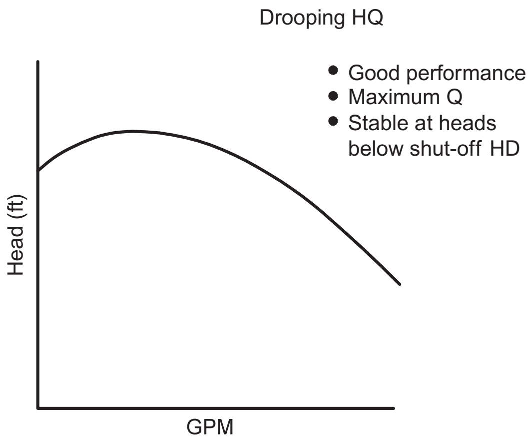


Fig. 3.18 Typical pump curve with a "drooping" head-capacity characteristic.

Pumps with drooping characteristic curves should be avoided because they may exhibit unstable operating characteristics. In some cases, however, such as systems with mostly dynamic loss and no requirements for parallel operation, drooping characteristics could be acceptable.

3.3.2.3 Steep rise

A "steep-rise" head-capacity curve is one in which there is a large increase in head between that developed at shutoff. In this type of pump curve, there is a large increase in head between that developed at design capacity and that developed at shutoff. This pump is stable and will operate in parallel over the entire range. It is best suited for operations requiring minimum capacity changes with pressure, but it is stable over a wide range of capacity vs. head. Sometimes it is applied to a limited portion of the curve. For example, as shown in Fig. 3.19, a pump may have a steep characteristic between 100 % and 50% of the design capacity. As a rule of thumb, "steep-rise" curves typically show a 140% increase in head between the capacities peak efficiency and shutoff.

3.3.2.4 Flat

A "flat" head-capacity curve is one in which the *head varies only slightly with capacity over its entire range* of operation. The characteristic might also be either drooping or rising. All drooping curves have a portion where the head developed is approximately constant for a range in capacity, called the flat portion of the curve. As shown in Fig. 3.20, some curves are qualified as flat either for their full range or for a limited portion of their range. As a rule of thumb, "flat" curves are those with no more than a 5% increase. Pumps with this type of curve are best suited for duties having wide fluctuation of capacity with nearly constant pressure.

Fig. 3.21 summarizes the four typical shapes of head-capacity curves.

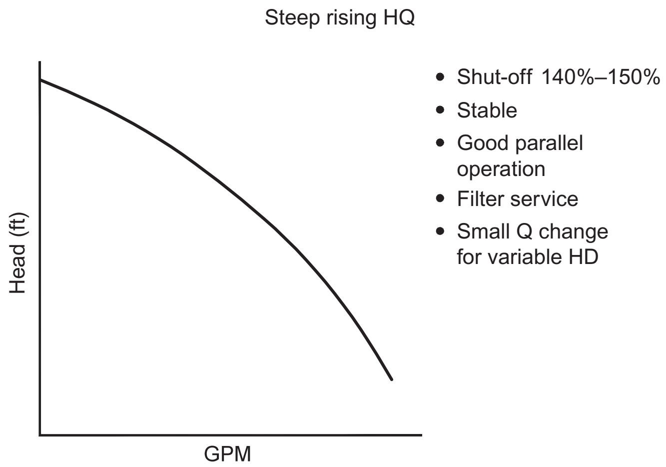


Fig. 3.19 Typical pump curve with a "steep-rise" head-capacity characteristic.

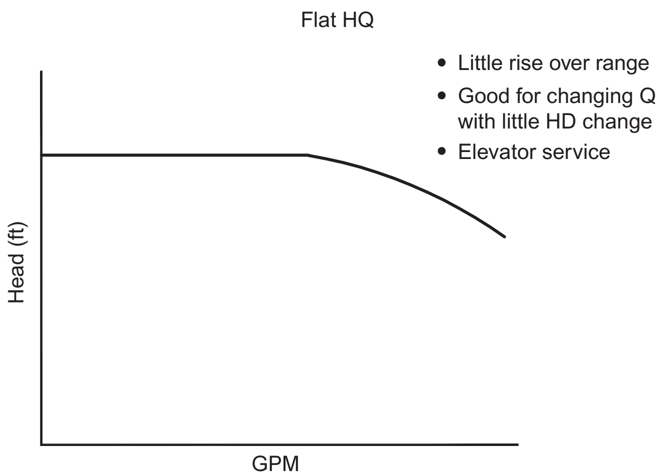


Fig. 3.20 Typical pump curve with a "flat" head-capacity characteristic.

3.3.2.5 Stable

Figs. 3.17 and 3.19 illustrate a head-capacity curve where only one capacity can be obtained at any one head.

3.3.2.6 Unstable

Figs. 3.18 and 3.22 illustrate a head-capacity curve where the same head is developed at two or more capacities.

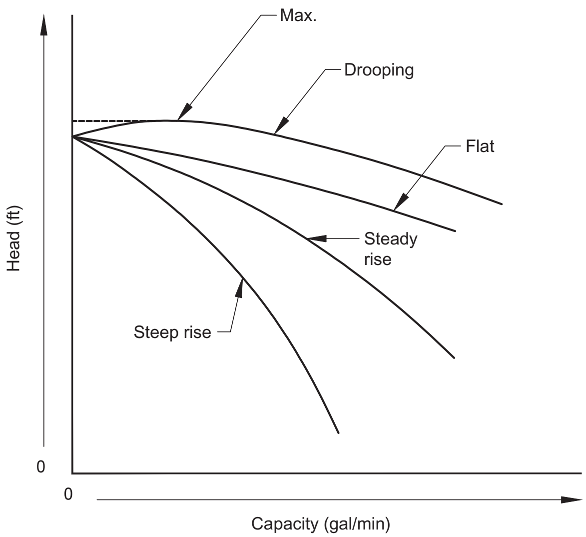


Fig. 3.21 Summary of four typical shapes of head-capacity curves.

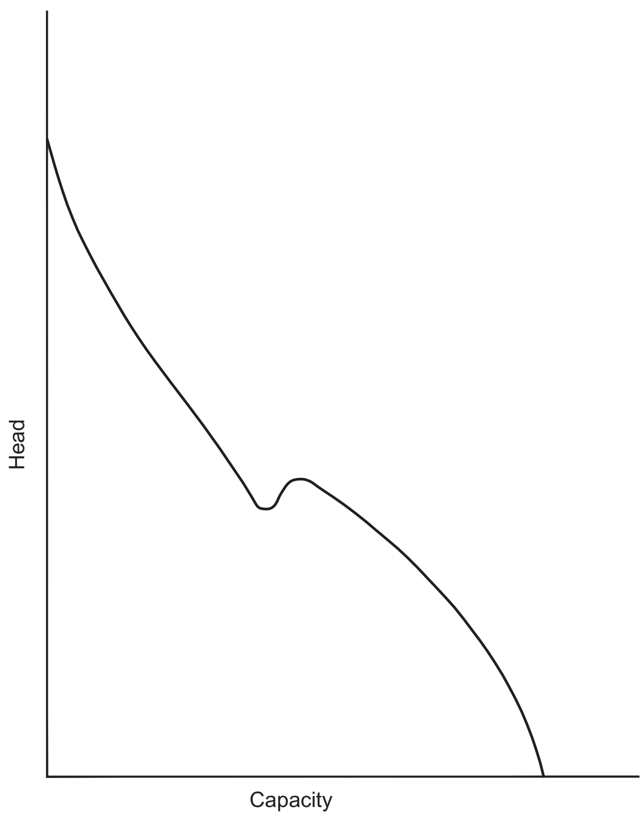


Fig. 3.22 Typical "unstable" head-capacity characteristic curve.

3.3.3 Power-capacity characteristic curve classifications

Head-power characteristic curves are also classified to their shape.

3.3.3.1 Nonoverloading

Fig. 3.23 illustrates a power-capacity characteristic curve that flattens out and then decreases as the capacity increases beyond the maximum efficiency point.

Pumps with nonoverloading power-capacity characteristic curves are preferred because the driver will not become overloaded under any operating condition. That said, they are not obtainable in all specific speed types of pumps.

3.3.3.2 Overloading

Fig. 3.24 illustrates a power-capacity characteristic curve that continues to increase with an increase in capacity. The shape of the power-capacity curve varies with the specific speed type. As a result, the power-capacity curve may have a

- very low value at shutoff (Figs. 3.23 and 3.24),
- high value at shutoff (Fig. 3.25),
- or any value in between.

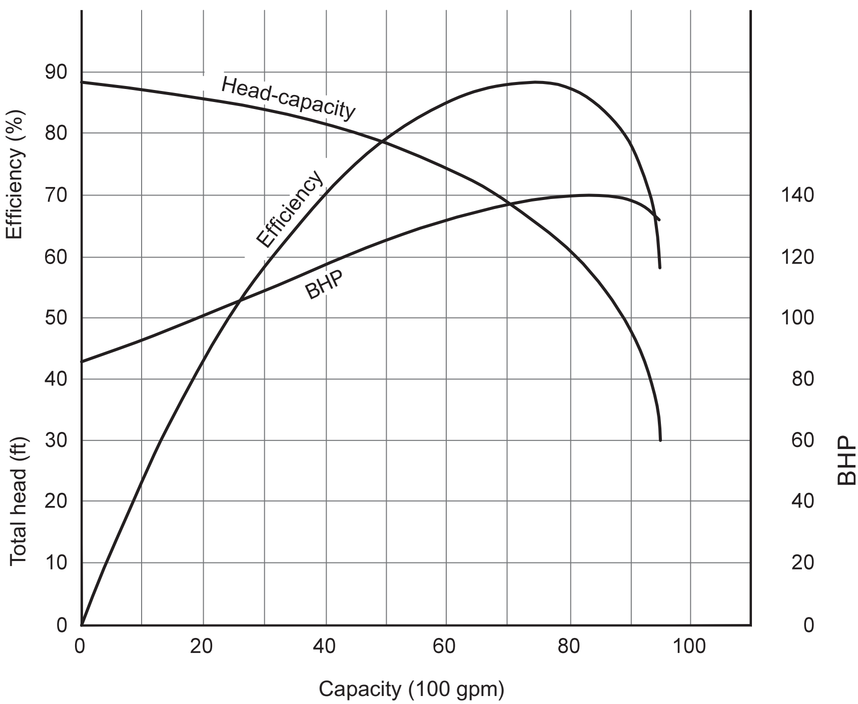


Fig. 3.23 "Nonoverloading" power-capacity characteristic curve with a reduction in head.

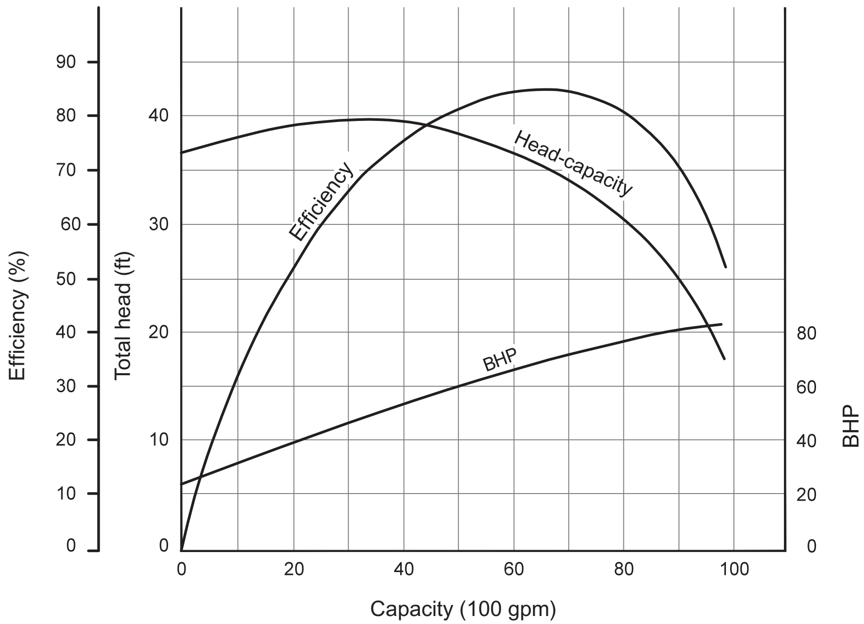


Fig. 3.24 "Overloading" power-capacity characteristic curve with a reduction in head.

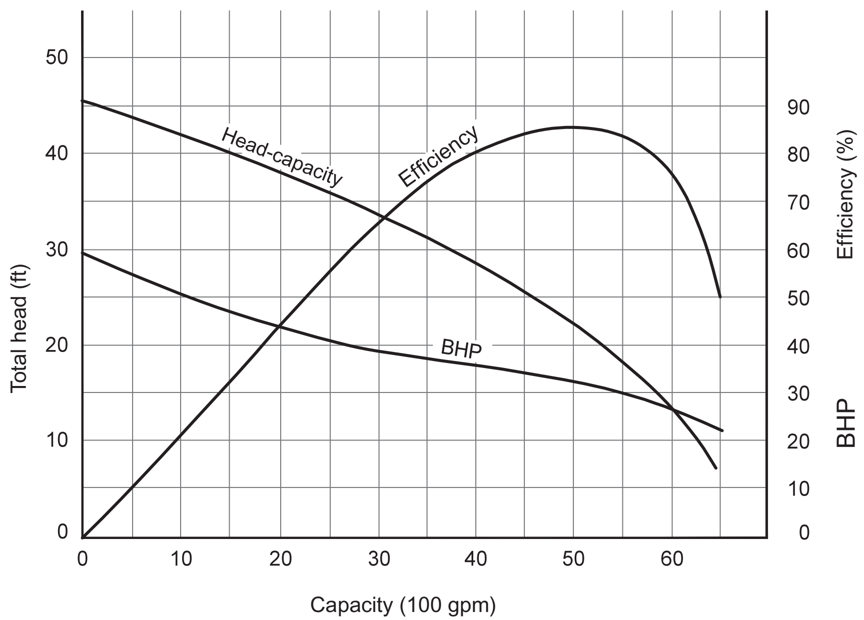


Fig. 3.25 "Overloading" power-capacity characteristic curve with an increase in head.

Fig. 3.24 illustrates an overloading characteristic curve with a decrease in head and increase in capacity, whereas Fig. 3.25 illustrates an overloading curve with an increase in head and decrease in capacity. The power of the driver should always be selected for the range of operating conditions that could possibly occur.

3.3.4 Typical characteristic curves for centrifugal pumps

Pump manufacturers use a number of methods to present centrifugal pump characteristic curves. Three commonly used methods to depict pump performance are shown as follows.

3.3.4.1 Typical centrifugal pump characteristic "performance" curve-speed and impeller diameter fixed

Fig. 3.26 shows a typical 6-in. double-suction centrifugal pump characteristic "performance" curve with speed and impeller diameter fixed. This format results from a pump test at constant speed. Manufacturers commonly use these characteristic curves to predict and guarantee pump performance.

3.3.4.2 Typical centrifugal pump characteristic "performance" curve-speed fixed, impeller diameter variable

Fig. 3.27 shows a typical 6-in. double-suction centrifugal pump characteristic "performance" curve used to express more fully the entire range of a pump, with various impeller diameters at continuous speed. These curves are commonly used in the selection of a pump for a specific service. The curves are generally made up from the average results of tests for various diameter impellers plotted as shown in Fig. 3.26.

3.3.4.3 Typical centrifugal pump characteristic "performance" curve-speed variable, impeller diameter fixed

Fig. 3.28 shows a typical 6-in. double-suction centrifugal pump characteristic "performance" curve driven at variable speeds with a fixed impeller diameter.

Most characteristic "performance" curves furnished by manufacturers are based on water as the pumped liquid. If the pump is handling some other liquid, adjustments must be made for viscosity and specific gravity before flow rate and discharge pressure (psi) can be predicted.

3.3.5 System head curves

3.3.5.1 Overview

A pump operates at the intersection of its head-capacity curve and the system head curve, in which the pump operates. Against a given head, the pump will deliver a given flow rate. A pump must also operate within the envelope of the $NPSH_A$ curve.

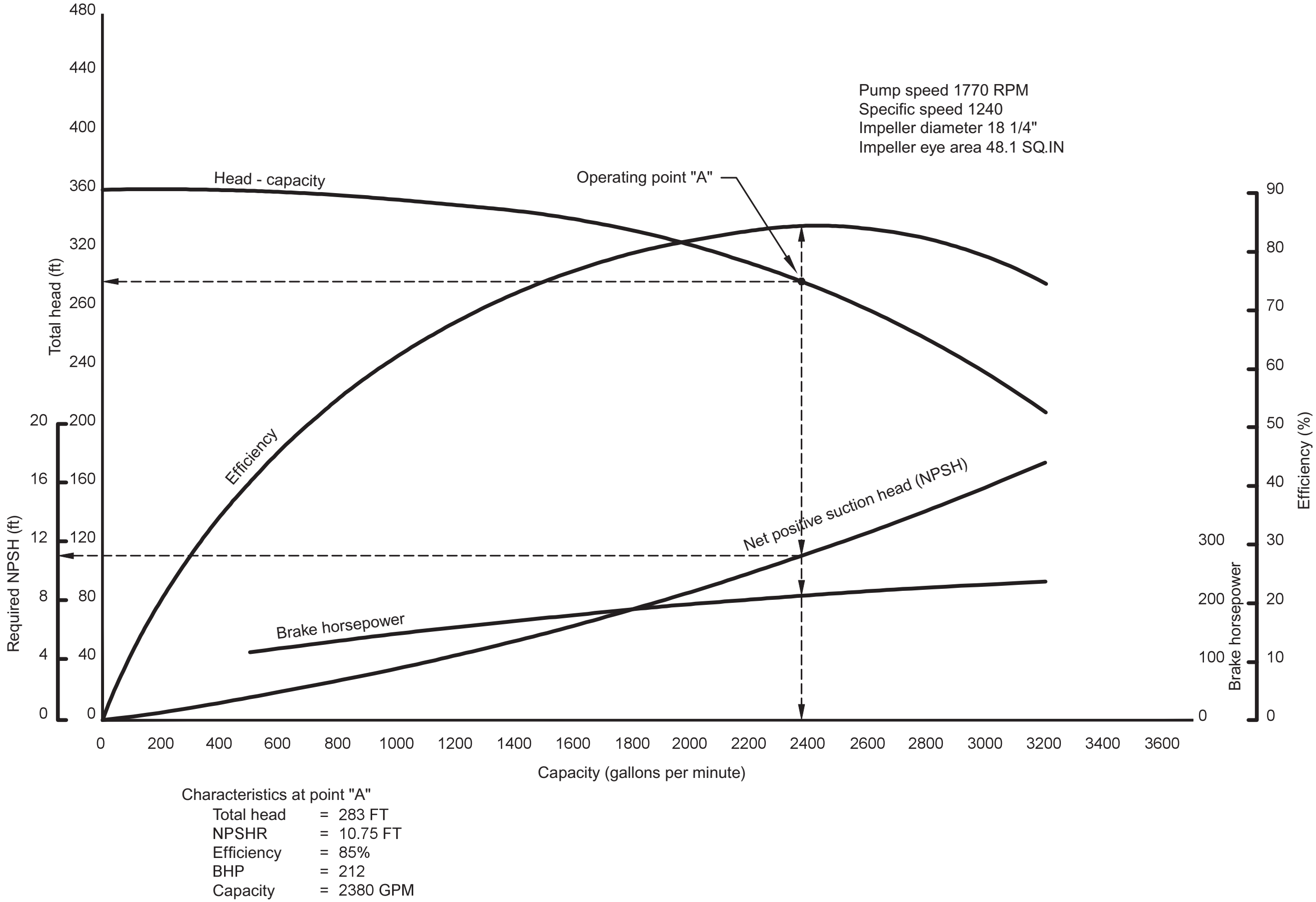


Fig. 3.26 Typical 6-in., single-stage, double-suction centrifugal pump characteristic “performance” curve—speed and impeller diameter fixed.

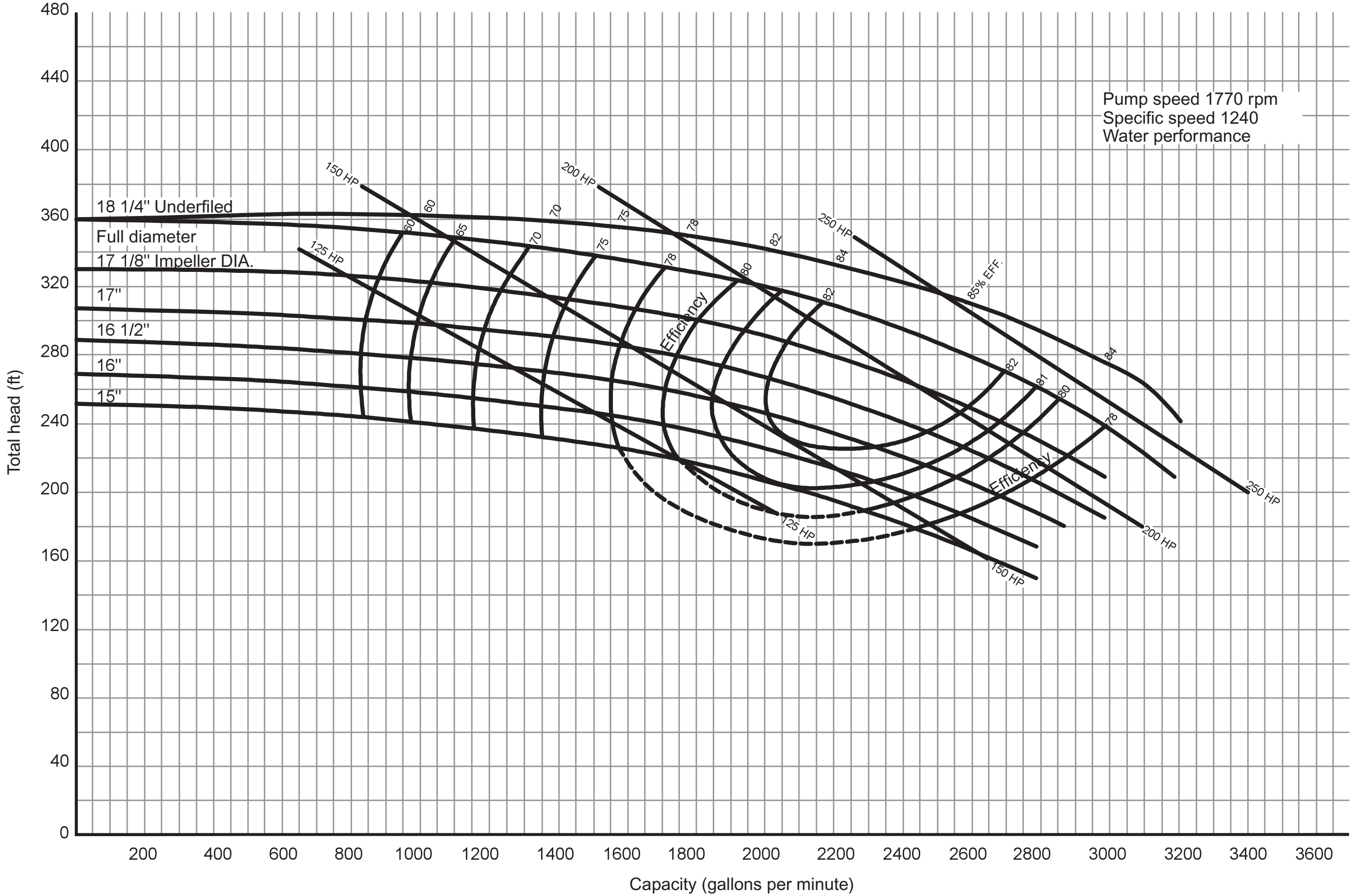


Fig. 3.27 Typical 6-in. double-suction centrifugal pump characteristic "performance" curve—speed fixed, impeller diameter variable.

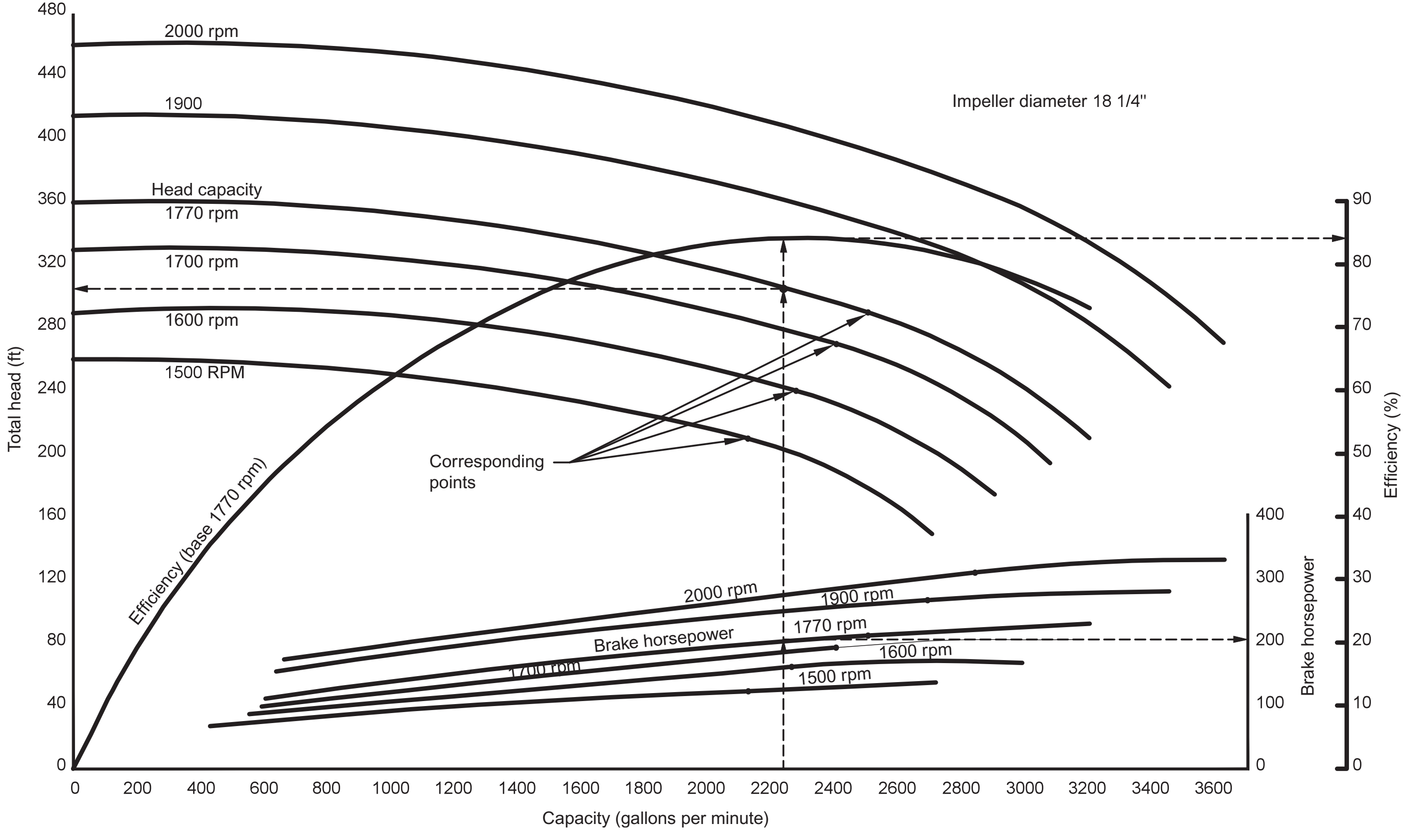


Fig. 3.28 Typical 6-in. double-suction centrifugal pump characteristic “performance” curve—speed variable, impeller diameter fixed.

The system in which a pump operates is composed of pipe, valves, fittings, and any number of components such as heat exchangers, filters, boilers, and so on.

In most piping systems both the head and the flow rate vary because the system has its own required pump head for a given flow rate. This can be seen in Fig. 3.29. As shown in the figure, the head required by the system, which is to be provided by the pump, is merely the friction drop in the pipeline between points "A" and "B," assuming the levels in both tanks are identical. This is a function of flow rate; it can therefore be plotted as a "system head curve" on the pump head-capacity performance curve. For this reason, as the pump speed is increased or decreased, a new equilibrium of head and flow rate is established by the intersection of the piping system head curve and pump performance curve.

Fig. 3.30 shows how the throughput can be changed by imposing an artificial backpressure on the pump. Adjusting the control valves orifice may shift the piping system head curve which establishes a new head-flow rate equilibrium point. As the pressure drop across the control valve increases from ΔP_1 to ΔP_2 to ΔP_3, the flow rate through the system decreases from Q_1 to Q_2 to Q_3.

The system head represents a complete piping system, for example, the friction losses of all the piping, elbows, valves, and so on, and the total static head vs. flow rate.

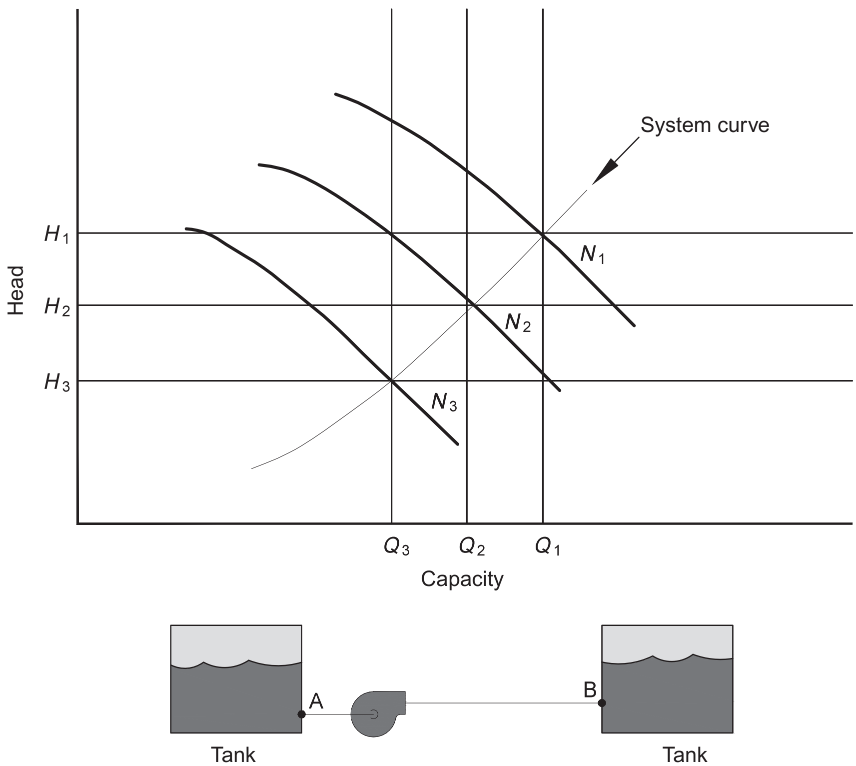


Fig. 3.29 Typical piping system head curve.

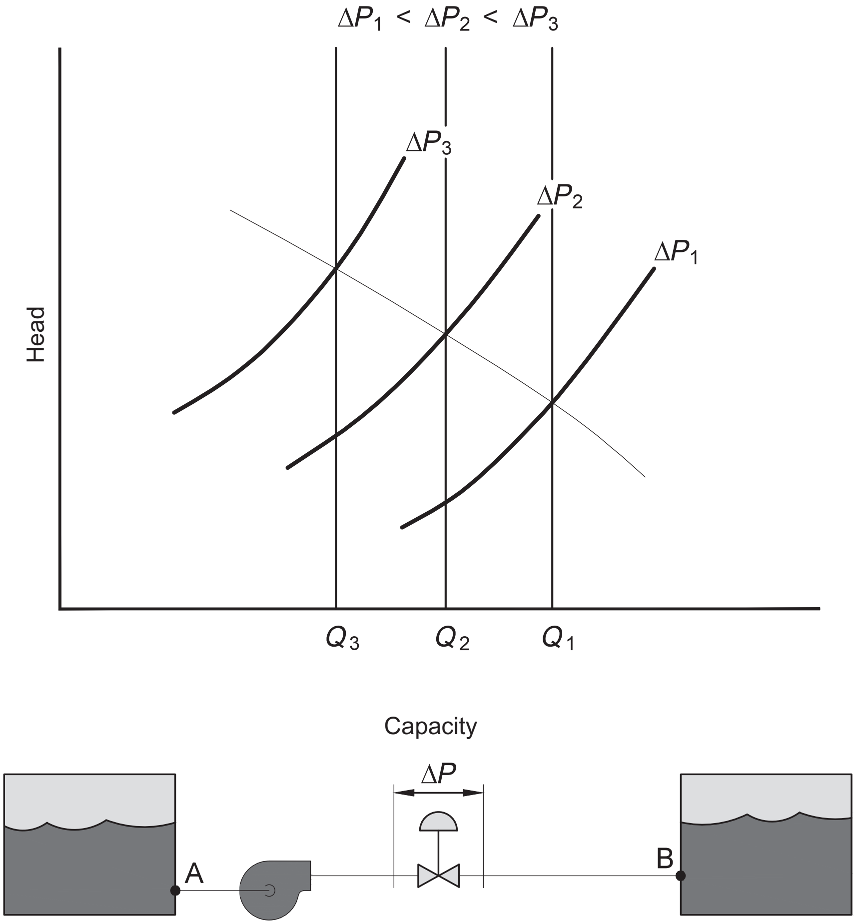


Fig. 3.30 Typical system curve showing the effects of adding a control valve.

It is a graphical representation of how much TDH, in feet, is required to pump various capacities through the piping system.

The system head curve consists of a constant (static) and an increasing (variable) portion (Fig. 3.31).

- *Static portion*: Instantaneous elevation differential between the source and termination, including pressures that exist on the surface of the liquid. As the liquid levels in the system can change over time, this change must be accounted for in the overall review of the system. This value is constant and is not dependent on flow. The static portion is plotted as a straight line at the appropriate head. The constant (static) portion represents the static head difference between the suction and the discharge at zero flow and is equal to:

$$H_{Static} = (P_2 - P_1)\frac{2.31}{SG} + H_2 - H_1 \tag{3.4}$$

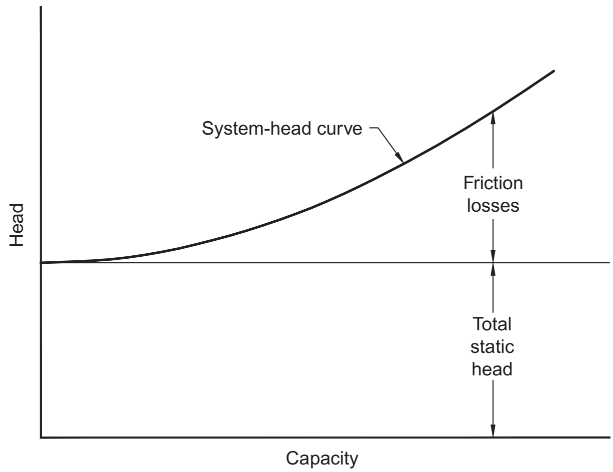


Fig. 3.31 System head curve.

- *Variable portion*: Head of motion or friction. The Hydraulic Institute has compiled a book of tables showing pipe and fitting frictions for various pipe components and flows. Fitting friction is generally expressed in equivalent lengths of straight run pipe. Pressure drops for process components such as heat exchangers must be obtained from the equipment manufacturer. These pressure drops can be converted in feet of head loss across them. Once a system is laid out and all components are identified, including changes in pipe size, the variable head can be calculated for a number of flow rates. One must remember that different pipe and fitting sizes will have different friction losses for identical flow rates. For ease of calculation, all lengths and fittings of the same size should be lumped together. This will minimize the number of calculations. The variable portion represents the head required to overcome friction as a result of flow, and it varies as the square of the flow and is equal to:

$$H_{Variable} = \frac{(P_{f1} + P_{f2} + P_C)2.31}{SG} \tag{3.5}$$

The system head curve (Fig. 3.31) is obtained by combining the system friction curve (Fig. 3.32) with a plot of the TDH.

3.3.5.2 Determination of system friction

The determination of friction losses is usually rough approximations at best as the roughness of the pipe is not known. The friction loss will increase as pipe deteriorates with age. Therefore it is common practice to base friction loss calculations on old pipe, thus allowing for friction losses in excess of those that will be obtained when the pipe is new. As a result, the pump is generally designed for excess head and delivers overcapacity when installed in a new system.

The flow of any liquid is accompanied by two types of friction: Internal friction (viscosity) caused by the rubbing of the fluid particles against one another; and external friction caused by the rubbing of the fluid particles against the pipe walls or against

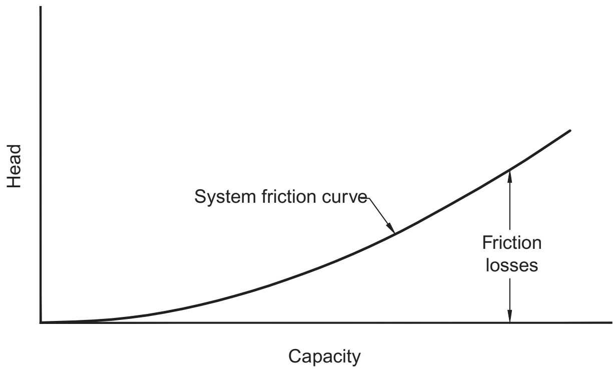


Fig. 3.32 System friction curve.

the static layer of liquid adhering to the walls. Energy must be expended to overcome this friction.

If the flow is turbulent, the friction developed is partially dependent upon the roughness of the walls. Since the interior surfaces of pipes of the same material are practically the same irrespective of pipe diameter, small pipes are relatively rougher than larger ones. Thus, for equal velocities, the larger the pipe, the smaller will be the friction loss. The roughness of the pipe wall also depends upon the material from which the pipe is made and, after the pipe has been in service, upon any change that occurs at the inner surface.

3.3.5.2.1 Friction loss in piping

Volume 3, Chapter 6 of this series presents detailed calculation methods for determining the friction loss in piping systems. Table 3.6 lists the friction, in feet of water, for the flow of water through new Schedule 40 steel pipe. This table is based on the Darcy-Weisbach formula:

$$h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) \tag{3.6}$$

where

h_f = loss in head, feet of liquid (m)
L = length of pipe, ft (m)
D = inside diameter of the pipe, ft (m)
V = velocity, ft/s (m/s)
g = acceleration due to gravity
= 32.2 ft/s^2 (9.8 m s^2)
f = conversion factor dependent upon the relative roughness of the pipe, the velocity of the liquid, the size of the pipe, and the viscosity of the liquid

Table 3.6 Velocity and friction head loss in new piping

gpm	v[a]	f[b]	v	f	v	f	v	f	v	f	v	f	v	f	v	f	gpm
					1-in. pipe (1.049-in. ID)		1 1/4-in pipe (1.380-in. ID)		1 1/2-in. pipe (1.610-in. ID)								
1					0.37	0.11											1
2					0.74	0.39	0.43	0.10									2
3					1.11	0.82	0.64	0.21	0.47	0.10							3
4					1.49	1.37	0.86	0.36	0.63	0.17	2-in. pipe (2.067-in. ID)						4
5					1.86	2.08	1.07	0.54	0.79	0.26							5
													2 Vi-in. pipe (2.469 in. ID)				
6					2.23	2.83	1.28	0.76	0.95	0.35	0.57	0.1					6
															3-in. pipe (3.068 in. ID)		
8					2.97	4.88	1.72	1.29	1.26	0.61	0.76	0.17					8
10					3.71	7.12	2.14	1.95	1.57	0.90	0.96	0.26	0.67	0.11			10
	3′/i-in pipe (3.548-in. ID)																
15					5.56	15.0	3.21	4.06	2.36	1.87	1.43	0.54	1.00	0.23			15
20					7.41	25.6	4.28	6.80	3.15	3.12	1.91	0.92	1.34	.38	.87	.13	20
			4-in. pipe (4.026 in. ID)														
25	0.81	0.10					5.35	10.3	3.94	4.70	2.38	1.39	1.67	.58	1.08	.20	25
30	0.97	0.13					6.43	14.4	4.72	6.60	2.86	1.92	2.00	.81	1.30	.28	30
40	1.30	0.23	1.01	0.12					6.30	11.2	3.82	3.35	2.68	1.36	1.73	0.47	40
					5-in. pipe (5.047-in, ID)												
50	1.62	0.34	1.26	0.18					7.87	16.6	4.77	5	3.34	2.06	2.16	.72	50
60	1.94	0.48	1.51	0.25							5.72	7	4.02	2.85	2.60	.99	60
							6-in. pipe (6.065-in. ID)										
70	2.27	0.63	1.76	0.34	1.12	0.1 1					6.68	9.4	4.68	3.80	3.03	1.33	70
80	2.59	0.82	2.01	0.43	1.28	0.15					7.62	11.9	5.35	4.95	3.46	1.72	80
90	2.91	1.00	2.27	0.54	1.44	0.18					8.6	14.7	6.02	6.05	3.89	2.13	90
100	3.24	1.24	2.52	0.65	1.60	0.22	1.11	.09			9.56	18.7	6.70	7.47	4.83	2.58	100
125	4.05	1.82	3.15	1.00	2.00	0.33	1.39	.13					8.37	11.1	5.41	3.90	125

150	4.86	2.55	3.78	1.39	2.40	0.47	1.66	.18	8-in. pipe (7.981-in. ID)				10.0	15.4	6.50	5.44	150
175	5.66	3.40	4.40	1.90	2.80	0.62	1.94	.24					11.7	20.8	7.58	7.30	175
200	6.48	4.35	5.04	2.40	3.20	0.80	2.22	.31							8.66	9.18	200
225	7.30	5.44	5.66	2.98	3.60	0.97	2.50	.38	1.44	0.10					9.75	11.6	225
250	8.10	6.59	6.29	3.68	4.00	1.19	2.77	.47	1.60	0.12					10.8	14.0	250
	3 1/2-in. pipe (3.548-in. ID)	4-in. pipe (4.026-in. ID)	5-in. pipe (5.047-in. ID)	6-in. pipe (6.065-in. ID)	8-in. pipe (7.981-in. ID)					3-in. pipe (3.068-in. ID)							
275	8.91	7.9	6.92	4.35	4.40	1.43	3.05	0.56	1.76	0.15	10-in. pipe (10.020-in. ID)			11.9	16.9	275	
300	9.72	9.3	7.55	5.04	4.80	1.65	3.32	0.66	1.92	0.17				13.0	19.6	300	
350	11.3	12.2	8.80	6.85	5.60	2.21	3.88	.88	2.24	0.23							350
400	13.0	15.9	10.1	8.67	6.40	2.89	4.44	1.12	2.56	0.29	1.62	0.10					400
450	14.6	20.0	11.3	10.9	7.20	3.56	4.99	1.40	2.88	0.37	1.82	0.12					450
500			12.6	13.3	8.00	4.36	5.54	1.72	3.20	0.45	2.03	0.15					500
550			13.9	16.0	8.80	5.17	6.10	2.06	3.52	0.55	2.23	0.18					550
600			15.1	19.1	9.60	6.16	6.65	2.42	3.84	0.63	2.44	0.21					600
650					10.4	7.22	7.20	2.78	4.16	0.73	2.64	0.24					650
700					11.2	8.29	7.75	3.25	4.47	0.85	2.84	0.28					700
750					12.0	9.40	8.31	3.63	4.80	0.97	3.04	0.31					750
800					12.8	10.3	8.87	4.11	5.11	1.11	3.25	0.35					800
900					14.4	13.0	9.96	5.12	5.75	1.33	3.65	0.44					900
1000					16.0	15.8	11.1	6.17	6.40	1.64	4.06	0.55					1000
1100					17.6	19.0	12.2	7.45	7.04	1.98	4.46	0.64					1100
1200							13.3	8.73	7.67	2.36	4.87	0.75					1200
1300							14.4	10.2	8.31	2.71	5.27	0.88					1300
1400							15.5	11.9	8.95	3.10	5.68	1.02					1400
1500							16.7	13.2	9.60	3.49	6.09	1.18					1500

Continued

Table 3.6 Continued

gpm	v[a]	f[b]	v	f	v	f	v	f	v	f	v	f	v	f	v	f	gpm
1600							17.8	15.0	10.2	3.92	6.49	1.31					1600
1800							20.0	18.5	11.5	4.99	7.30	1.60					1800
2000									12.8	5.96	8.11	1.97					2000
2500									16.0	9.00	10.2	2.95					2500
3000									19.2	12.50	12.2	4.15					3000
3500									22.4	16.60	14.2	5.60					3500
4000											16.2	6.90					4000
4500											18.3	8.80					4500
5000											20.3	10.8					5000
5500											22.3	13.0					5500
6000											24.4	15.3					6000

Friction values apply to Schedule 40 (standard weight) steel pipe carrying water.
[a]Velocity, in feet per second.
[b]Friction head loss, in feet of water per 100 feet of pipe.

3.3.5.2.2 Friction loss in valves and fittings

The pressure drop through fittings and valves can be much greater than that through straight run of pipe itself. The additional frictional effects can be determined using an extension of the Darcy-Weisbach equation. The extension involves the determination of either of the following:

- Resistance coefficients for fittings
- Flow coefficients for valves
- Equivalent lengths for both valves and fittings

For a detailed discussion of each of the earlier methods, refer to *Volume 3, Chapter 6* in this series. Table 3.7 summarizes the equivalents of various commonly used valves and fitting.

3.3.5.3 Operating point

Fig. 3.33 shows an example of a system head curve superimposed on the pump characteristic "performance" curve. Superimposing the pump characteristic "performance" curve on the piping system head curve gives the point at which a particular pump will operate. Assuming a centrifugal pump's characteristic curve rises steadily to shutoff, the centrifugal pump will operate at one point. This point is the intersection of the piping system head curve and the pump's characteristic curve.

Changing the resistance of the piping system by partially closing the flow control valve (FCV) in the discharge line produces a change in the system head curve which shifts the operating point to a higher head but lower flow rate. This concept illustrates the most common basis for centrifugal pump control is by throttling the discharge. As a control valve in the discharge line varies the total pressure drop in the system, the Head Curve varies. This variance in the system head curve causes the operating point to shift right or left on the pump characteristic curve, with a resulting increase or decrease in flow rate.

Fig. 3.34 is an example illustrating how throttling the flow control valve (FCV) changes the system head curve which changes the intersection of the two curves which in turn changes the operating point. As shown in Fig. 3.34, with the FCV fully open the pump characteristic curve intersects the piping system head curve at point "A" which yields an operating point of 500 gpm. Partially closing the FCV increases the system friction and produces a new or "artificial" system head curve. The intersection of the pump characteristic curve and the new piping system head curve yields the new operating point of 375 gpm.

Sometimes, pumps with low specific speeds, a portion of the pump's characteristic head-capacity curve is unstable which could cause fluctuations in the pump speed, capacity, and power requirements. Fig. 3.35 illustrates a "drooping" pump characteristic head-capacity curve that can cause unstable operation. Point "F" represents the normal operating condition of the pump when the FCV is fully open. As the FCV is throttled the system head curves change (B, C, D, and E). As the system resistance (friction) is increased (by throttling the FCV) the pump is able to supply the additional head until point C is reached on the pump head-capacity characteristic curve. Additional system resistance causes the operating point to move into the part of the pump

Table 3.7 Equivalent length of valves and fittings in feet

Nominal pipe size in.	Globe valve of ball check valve	Angle valve	Swing check valve	Plug valve	Gate or ball valve	45° all	Short red. all	Long red. all	Branch of tee	Run of tee	90° miter bends			Enlargement					Contraction				
														Sudden			Std. red.		Sudden			Std. red.	
														Equiv. *L* In terms of small *d*									
						Weld thrd	Weld thrd	Weld thrd	Weld thrd	Weld thrd	2 miter	3 miter	4 miter	*d*/*D* =1/4	*d*/*D* =1/2	*d*/*D* =3/4	*d*/*D* =1/2	*d*/*D* =3/4	*d*/*D* =1/4	*d*/*D* =1/2	*d*/*D* =3/4	*d*/*D* =1/2	*d*/*D* =3/4
1½	55	26	13	7	1	1 2	3 5	2 3	8 9	2 3				5	3	1	5	1	3	2	1	1	–
2	70	33	17	14	2	2 3	4 5	3 4	10 11	3 4				7	4	1	6	1	3	3	1	1	–
2½	80	40	20	11	2	2 ..	5 ..	3 ..	12	3 ..				8	5	2	8	2	4	3	2	2	–
3	100	50	25	17	2	2	6	4	14	4				10	6	2	10	2	5	4	2	2	–
4	130	65	32	30	3	3	7	5	19	5				12	8	3	14	3	6	5	3	3	–
6	200	100	48	70	4	4	11	8	28	8				18	12	4	19	4	9	7	4	4	1
8	260	125	64	120	6	6	15	9	37	9				25	16	5	24	5	12	9	5	5	2
10	330	160	80	170	7	7	18	12	47	12				31	20	7	28	7	15	12	6	6	2
12	400	190	95	170	9	9	22	14	55	14	28	21	20	37	24	8	–	8	18	14	7	7	–
14	450	210	105	80	10	10	26	16	62	16	32	24	22	42	26	9	–	–	20	16	8	–	–
16	500	240	120	145	11	11	29	18	72	18	38	27	24	47	30	10	–	–	24	18	9	–	–
18	550	280	140	160	12	12	33	20	82	20	42	30	28	53	35	11	–	–	26	20	10	–	–
20	650	300	155	210	14	14	36	23	90	23	46	33	32	60	38	13	–	–	30	23	11	–	–
22	688	335	170	225	15	15	40	25	100	25	52	36	34	65	42	14	–	–	32	25	12	–	–
24	750	370	185	254	16	16	44	27	110	27	56	39	36	70	46	15	—	—	35	27	13	—	—
30	–	–	–	312	21	21	55	40	140	40	70	51	44										
36	–	–	–		25	25	66	47	170	47	84	60	52										
42	–	–	–		30	30	77	55	200	55	98	69	64										
48	–	–	–		35	35	88	65	220	65	112	81	72										
54	–	–	–		40	40	99	70	250	70	126	90	80										
60					45	45	110	80	260	70													

(Courtesy of GPSA)

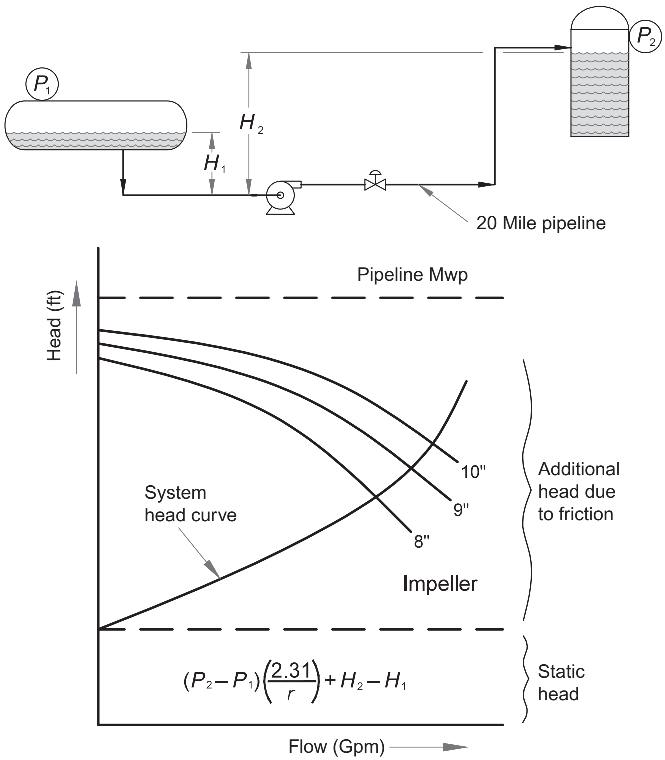


Fig. 3.33 Example of a system head curve superimposed on pump characteristic "performance" curve.

curve where the head decreases as the flow decreases. Operation in this region of the head-capacity curve may result in an unstable surging discharge pressure. Pumps with "Drooping" head-capacity curves should not be installed in parallel as one pump may operate at a lower flow rate than the other and could fail if operating below the minimum flow rate.

3.3.5.4 Cutoff point

The cutoff point is the maximum quantity of liquid that the available suction head can force into the impeller. It depends on the relationship between $NPSH_R$ and $NPSH_A$. Fig. 3.27 shows that the greatest possible capacity obtainable with this pump is about 3200 gpm, which may be obtained at a head of 240 ft using the 18-1/4-in. unfiled impeller. Pumps should not be selected with a cutoff close to the required rating. Pumps operating above cutoff will vibrate excessively and fail prematurely.

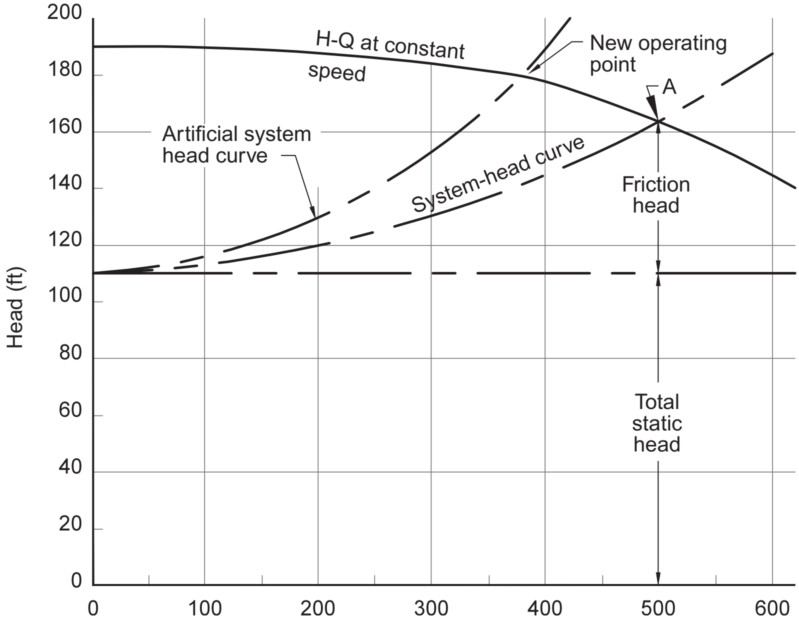


Fig. 3.34 Operating point "A" is where the piping system head curve intersects the pump's characteristic curve.

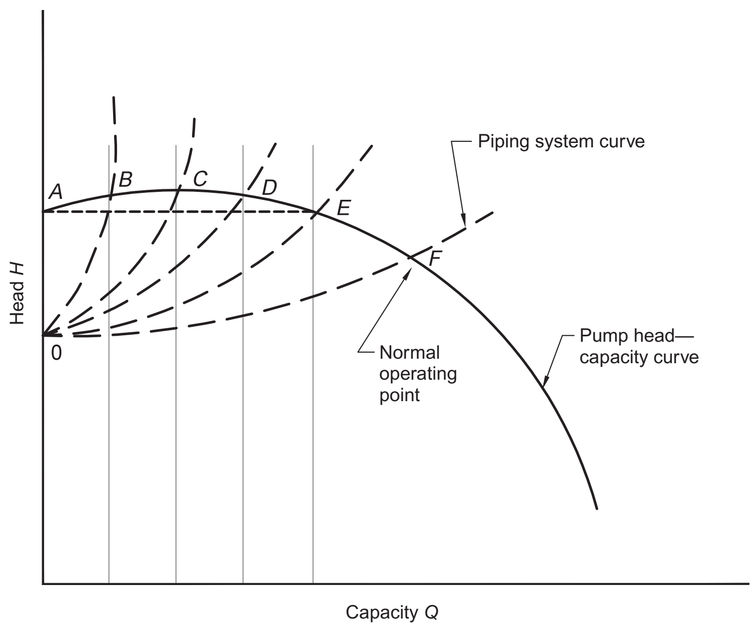


Fig. 3.35 Typical "drooping" head-capacity characteristic head curve that may indicate unstable operation.

3.4 Parallel and series operation

3.4.1 Parallel operation

Parallel pumping is used where systems require capacity to double while the head increases only slightly and is often used where the system may require operation at a lower flow. The head and flow rate will change when operating only one pump. The shutoff head of each pump must be the same for successful parallel pump installation. If one pump has a higher shutoff head than the other, it will keep the check valve closed in the discharge of the pump with the lower head until the system head drops below the lower head pump's shutoff. This could result in the lower head pump running at shutoff for extended periods of time. Check valves installed in the system will prevent backflow and pumps surging to maintain equal performance.

When designing multiple centrifugal pump installations it is necessary to keep in mind the interaction between the pump curves and the piping system head curve. That is, throughput cannot be doubled by adding an identical pump in parallel, and head is not doubled by adding an identical pump in series. The effect of adding two identical pumps in parallel can be seen in Fig. 3.36. As shown in Fig. 3.37, curve "A" is the

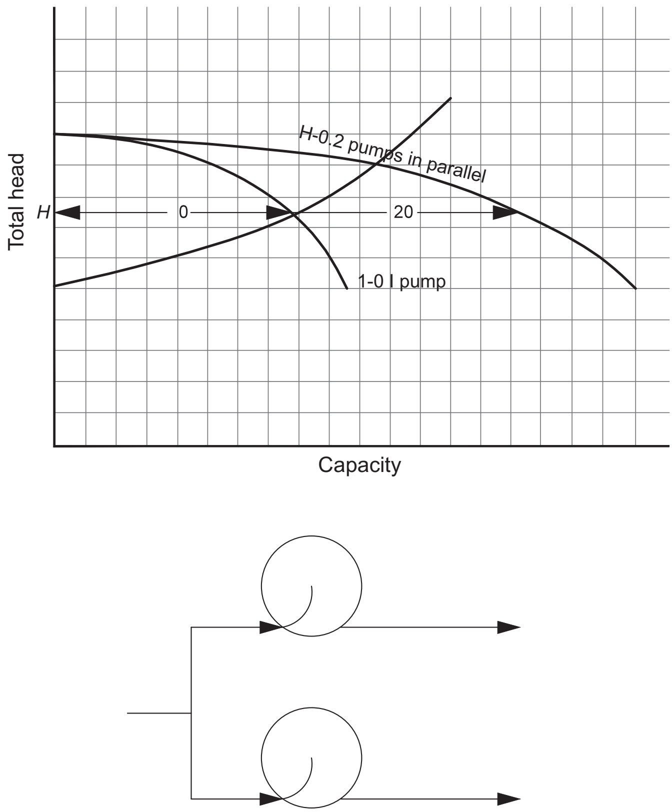


Fig. 3.36 Head-capacity curves for two identical pumps installed in parallel.

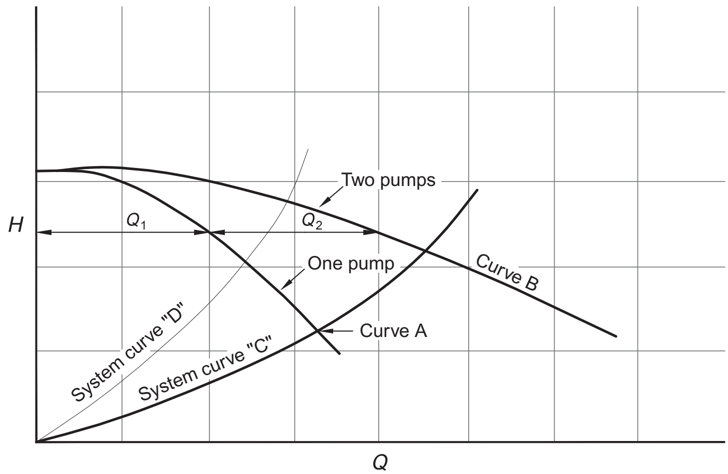


Fig. 3.37 Head vs flow rate graph showing the effect of installing two identical pumps in parallel.

pump performance curve for one pump. Curve "B" is constructed by doubling the flow rate at a given head to show how the pumps behave in parallel operation. Curve "C" shows a piping system head curve to which the addition of the second pump adds only about 50% to the system throughput. Curve "D" shows a steeper piping system head curve. In this case the system throughput is increased only about 20%.

Centrifugal pumps are used frequently in multiple pump installations where the pumps are piped to the same suction and discharge lines. The combined flow rate is the summation of the individual pump flows at the same TDH. *In parallel operation the capacities for each pump are added at any given head*. Specific rules when using two pumps in parallel are as follows:

- Head-capacity curves of the parallel pumps should be the same, or nearly so, at all points along their respective curves (they should not intersect)
- All centrifugal pumps discharging to an elevated or pressurized tank, and all centrifugal pumps operating in parallel should have check valves in the event of a pump shut down to keep the pump from spinning backward (the danger being a sheared shaft upon restart attempt)
- Driver size should be selected so that overloading at any point across the entire pump curve for the selected impeller does not occur
- Flow orifices or meters should be provided in each pump's discharge line to allow verifications of flow rates
- Suction and discharge piping should be arranged as symmetrically as practical so that all pumps have the same $NPSH_A$

When two or more pumps are connected to the same suction header and operated in parallel, the total volume is often much less than proportional to the number of pumps used. One pump seems to take all the liquid from the other pump or pumps. This effect is called "suction stealing" and arises from unequal suction pressures at the impeller inlets of the various pumps. It is most pronounced where the pressure in the suction header is low, so that the inequalities in friction between the inlet to the header and inlets to the various pump impellers greatly influence the volume of flow into the pump. The solution is to provide equal head losses between the inlet to the header and the inlets to the pump suction nozzles and adequate $NPSH_A$ to both pumps at the total flow rate. Independently matched pump curves give the same effect, especially if they are "flat," permitting minor inlet piping variances to produce major effects. Actual cases of suction stealing can usually be traced to flat or unstable curves.

3.4.2 Series operation

Series pumping can be thought of as a boost in pressure while maintaining the same flow rate. Flows for pumps operated in series must be equal. If one pump has a lower flow rate than the other, it could restrict throughout for the system. Fig. 3.38 shows the effect of installing two pumps in series. As shown in Fig. 3.39, curve "A" is the head vs. flow rate curve for one pump. The combined curve for both pumps, curve "B," is constructed by doubling the head of Curve "A" at each value of flow rate. The benefit of the additional pump can be seen by inspecting the intersection of the system head curves, "C" and "D," with the pump performance curves. The choice of whether to add an additional pump in series or in parallel is illustrated in Fig. 3.40. If the piping system head curve is shallow, more throughput is obtained from parallel operation. If the piping system head curve is steep, more throughput can be obtained by series operation.

Centrifugal pumps installed in series are used less than parallel installations. An example would be a low $NPSH_R$, low TDH pump taking suction from an atmospheric tank or vessel operating at its bubble point, and discharging to a higher $NPSH_R$, high

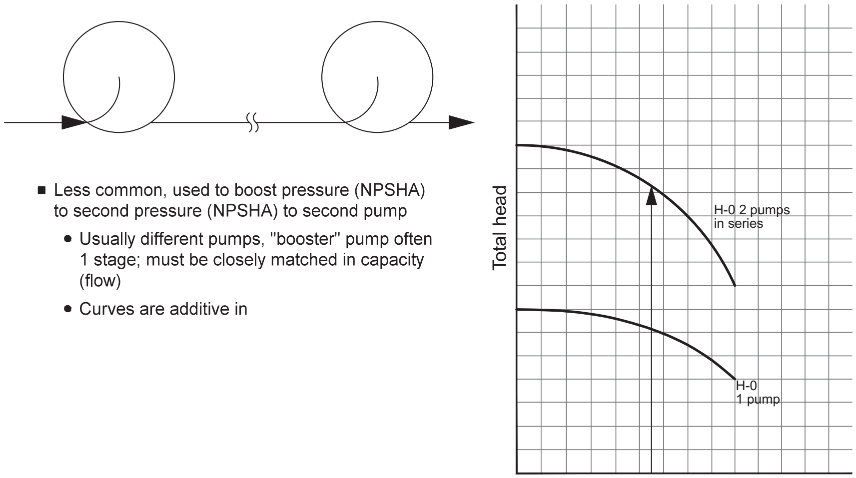


Fig. 3.38 Head-capacity curves for two identical pumps installed in series.

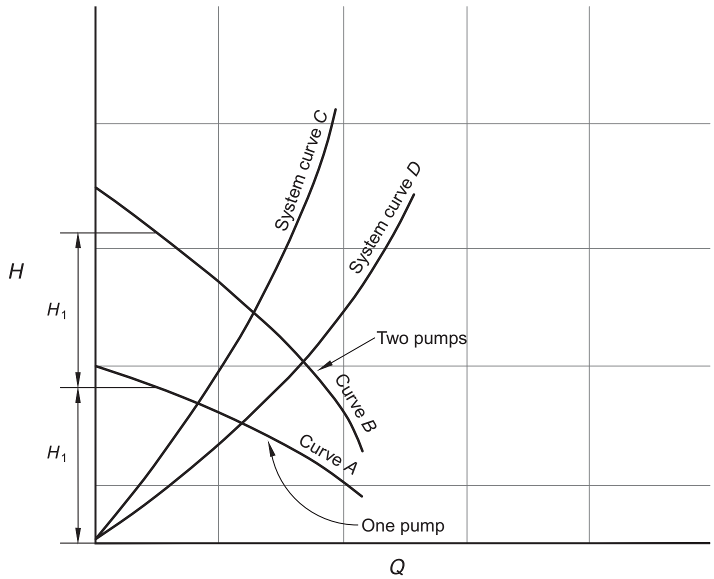


Fig. 3.39 Head flow rate graph showing the effect of installing two identical pumps in series.

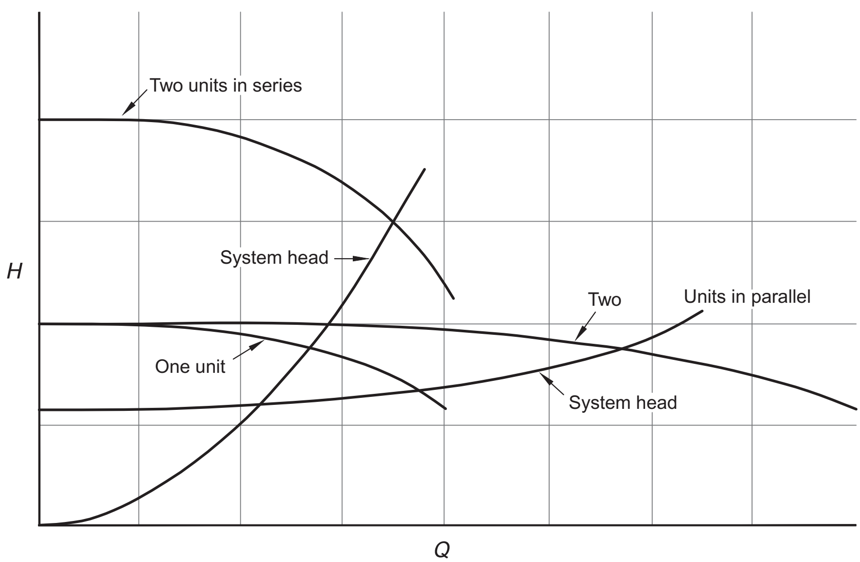


Fig. 3.40 Head vs flow rate graph showing the difference between installing two pumps in series or in parallel operation.

TDH pump. The combined head is the summation of the individual pump TDHs at the same flow. *In series operation the heads for each pump are added at any given capacity.*

It is important that pumps operated in series have adequate suction pressure. Occasionally, pumps in series operation have not developed the anticipated total differential head. This is usually the result of one pump operating under cavitating conditions because of insufficient $NPSH_A$.

3.4.2.1 Head-capacity characteristic "performance" curves for pumps in parallel and series

Fig. 3.41 illustrates the shape of the Head-Capacity Curves for both parallel and series operation for two pumps under various discharge conditions. Two pumps, P1 and P2, have Head-Capacity Characteristic Curves as shown and are to pump through pipe systems with characteristic Curves shown by System Curves 1, 2, 3, 4, and 5. The intersection of the piping system head characteristics with the pump head-capacity characteristics shows the quantities and heads at which the pumps will operate either singly, in series, or in parallel. Adequate suction pressure is assumed.

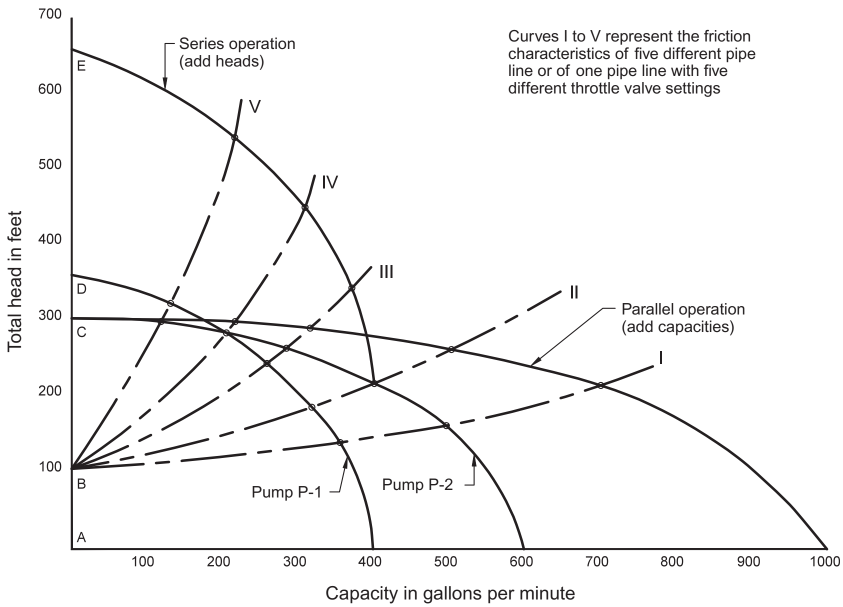


Fig. 3.41 Typical parallel and series operation of two centrifugal pumps pumping through a pipe system throttled at the discharge end.

3.5 Viscosity effects on centrifugal performance

3.5.1 General effects on performance

Centrifugal pump's performance deteriorates as viscosity of the fluid increases. An increase in liquid viscosity

- Increases BHP
- Decreases head
- Reduces capacity

As the physical size of the pump increases, the maximum viscosity that it can handle also increases. For example:

- 2 in. discharge nozzle can handle 43.5 cSt (200 SSU)
- 3 in. discharge nozzle can handle 154 cSt (700 SSU)
- 4 in. discharge nozzle can handle 220 cSt (1000 SSU)

Pumps can actually handle higher viscosities but only at increased efficiency penalties. When exact performance is essential, performance tests should be conducted with the specific viscous fluid to be handled.

3.5.2 Estimation of viscous performance

Since most manufacturers' curves are based on pumping water it is important to have a method for estimating the effect of viscosity upon water performance curves. There a number of methods to estimate the viscous performance when the water performance is known. Two commonly used estimation methods are provided.

3.5.2.1 Ingersoll-Rand method

The following equation can be used to estimate the viscous performance of a pump when the water performance of the pump is known:

$$Q_{VIS} = C_Q Q_W \tag{3.7}$$

$$H_{VIS} = C_H H_W \tag{3.8}$$

$$E_{VIS} = C_E E_W \tag{3.9}$$

Field units

$$BHP_{VIS} = \frac{\lfloor (Q_{VIS})(H_{VIS})(SG) \rfloor}{[(3960)(E_{VIS})]} \tag{3.10a}$$

SI units

$$BHP_{VIS} = \frac{[(Q_{VIS})(H_{VIS})(SG)]}{[(367.7)(E_{VIS})]} \tag{3.10b}$$

Where:

Q_{VIS} = Viscous capacity, gpm (m^3/h)
H_{VIS} = Viscous head, ft (m)
E_{VIS} = Viscous efficiency, percent
BHP_{VIS} = Viscous BHP, hp (kW)
C_Q, C_H, C_E = Viscous correction factors
= From Fig. 3.42

Fig. 3.42 is used to determine the correction factors used in Eqs. (3.7)–(3.9). The correction curves are not exact for any particular pump and should be used with judgment. Fig. 3.42 is subject to the following limitations:

(1) Use only within the scales shown. Do not extrapolate.
(2) Use only for pumps of conventional hydraulic design in the normal operating range with open or closed impellers.
(3) Do not use for mixed- or axial-flow pumps or for pumps of special hydraulic design for either viscous or nonviscous liquids.
(4) Use only where adequate NPSH is available in order to avoid cavitation.
(5) Use only on Newtonian liquids. Gels and slurries, paper stock, and other nonuniform liquids may produce widely varying results, depending on the particular characteristics of the liquids.

3.5.2.2 Hydraulic institute method

Fig. 3.43 shows the effect of viscosity on pump performance (solid lines—water, dashed lines—viscous fluid). The Hydraulic Institute developed the data provided in Fig. 3.44 which provides viscosity corrections to pump performance. The curves convert the pump's water performance to that of the viscous fluid. Fig. 3.43 is subject to the following limitations:

(1) Curves do not apply to mixed- or axial-flow pumps, nor to pumps handling non-Newtonian liquids.
(2) Use only on Newtonian liquids. Slurries and similar non-Newtonian liquids may produce widely different results depending on their characteristics.
(3) Curves only apply to single-stage pumps operating at the best efficiency flow rate for the impeller.
(4) For multistage pumps the head per stage should be used to obtain the proper correction factors, which should then be verified with the original equipment manufacturer.

The viscosity correction curves provide factors to be applied at the best efficiency point (BEP) to arrive at the viscous performance curve. Efficiency is the parameter affected most severely by viscosity, followed by capacity, then head. Since efficiency

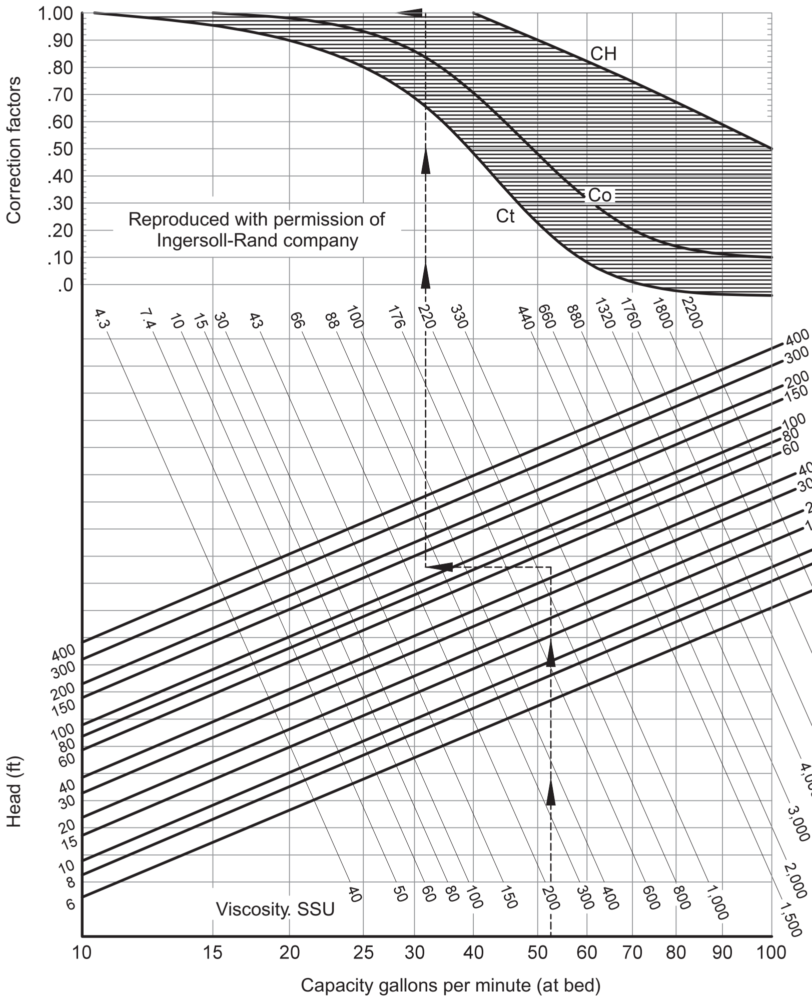


Fig. 3.42 Ingersoll-Rand viscosity correction factors used to estimate viscous performance. Courtesy of Ingersoll-Rand Company.

has the greatest effect, power cost should be evaluated as it may impact the pump selection.

Positive displacement reciprocating, screw or gear pumps are very efficient in handling viscous fluids. They should be considered when fluid viscosity exceeds 43.2–110 cSt (200–500 SSU) and when there are very few suspended solids present.

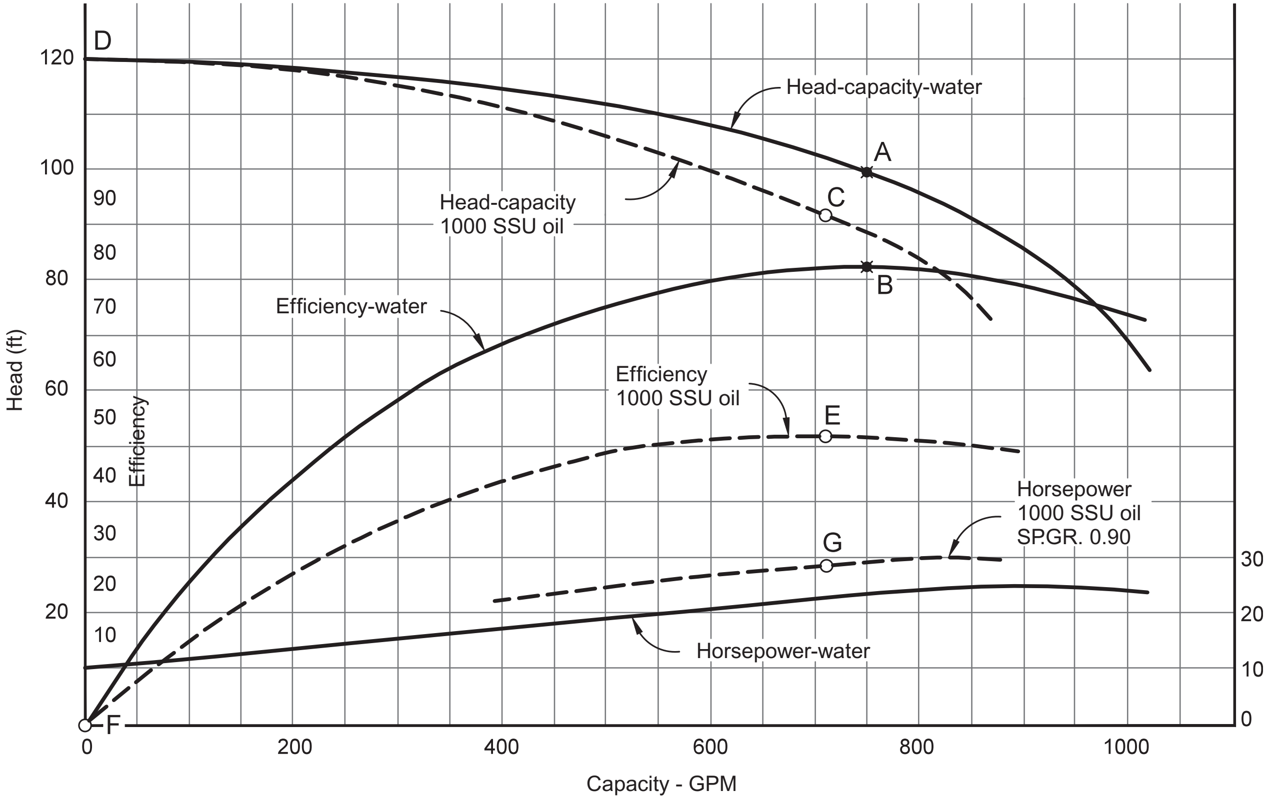


Fig. 3.43 Effect of viscosity on centrifugal pump performance.

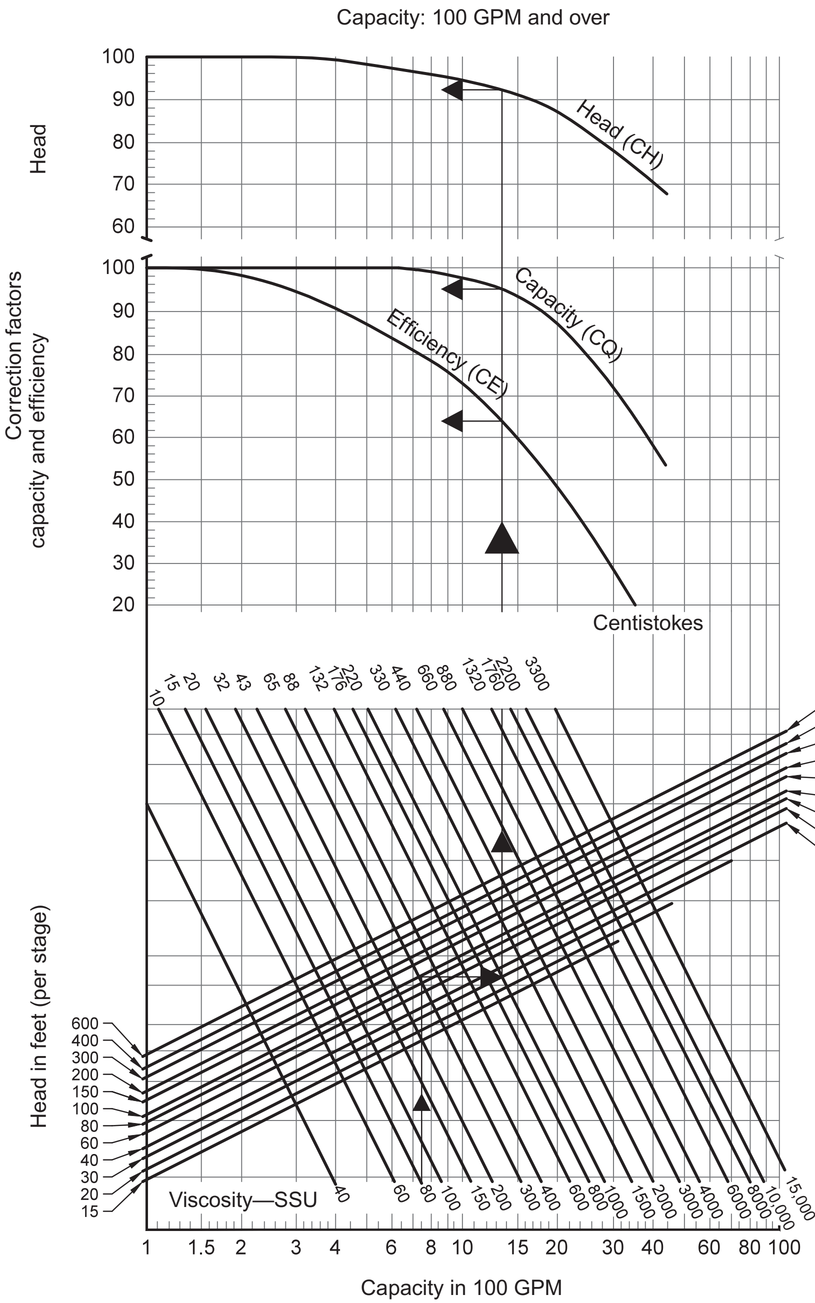


Fig. 3.44 (1 of 2) Hydraulic Institute Viscosity Corrections for centrifugal pumps handling viscous fluids 100 gpm and over. Courtesy of Hydraulic Institute.

(Continued)

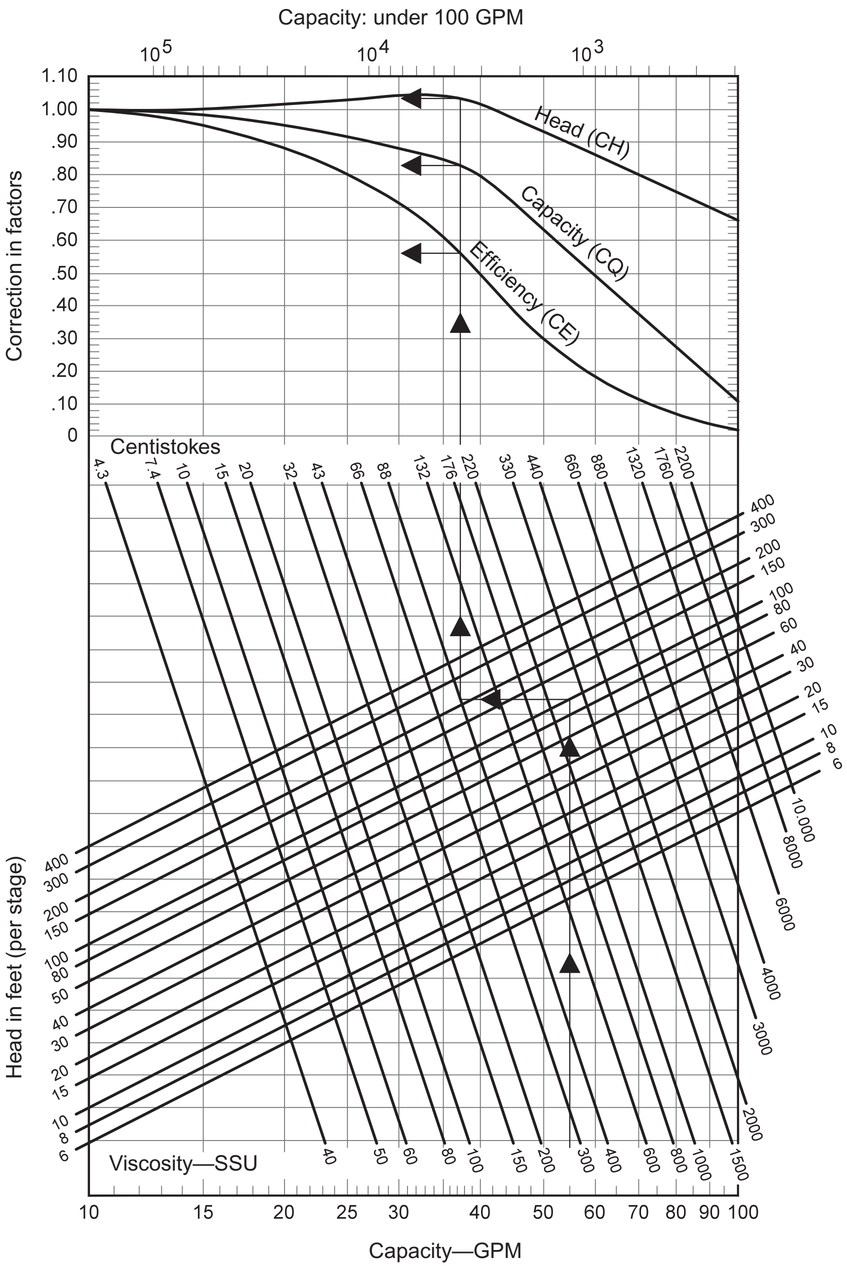


Fig. 3.44— Cont'd (2 of 2) Hydraulic Institute Viscosity Corrections for centrifugal pumps handling viscous fluids under 100 gpm.
Courtesy of Hydraulic Institute.

Example 3.1. Calculation of viscosity effects on a centrifugal pump (field units)
Given:

Pump performance obtained by test on water is shown in Fig. 3.43

Oil SG	0.9
Viscosity	220 cSt (1000 SSU)
Head @ BEP	100 ft (Point A)
Capacity (Q) @ BEP	750 gpm
Efficiency @BEP	82% (Point B)

Determine:

Pump performance pumping 0.9 oil with viscosity 220 cSt (1000 SSU)
Solution:

(1) Determine the viscosity correction factors from Fig. 3.44 (1 of 2).
- Enter Fig. 3.44 (1 of 2) with 750 gpm, 100 ft and 220 cSt (1000 SSU) and read the following correction factors:
- Capacity correction factor (C_Q): $C_Q = 0.95$
- Head correction factor (C_H): $C_H = 0.92$
- Efficiency correction factor (C_E): $C_E = 0.635$

(2) Multiply the respective correction factors by the water capacity, head, and efficiency at the BEP values as follows:

$$Q_V = (Q_W)(C_Q) = (750)(0.95) = 712\,\text{gpm}$$

$$H_V = (H_W)(C_H) = (100)(0.92) = 92\,\text{ft}$$

$$E_V = (E_w)(C_E) = (82)(0.635) = 52\%\text{eff}$$

- The point for viscous capacity and head can now be located below the water curve (Point C, Fig. 3.43). The viscous head-capacity performance curve is drawn from the water head to zero capacity (Point D) through the viscous head-capacity point (Point C) with approximately the same shape as the water curve. The efficiency at the BEP for viscous performance can be plotted as Point E and the viscous efficiency curve plotted from zero (Point F) through Point E; the shape of the curve is similar to that obtained for water efficiency.

(3) Determine BHP at the BEP for viscous conditions. The horsepower (BHP) for any capacity can now be calculated from the head and efficiency at the capacity designed. The BEP for the viscous performance is:

$$\begin{aligned}\text{BHP} &= \frac{[(Q_V)(H_V)(SG)]}{[(3960)(E_V)]}\\ &= \frac{(712)(92)(0.9)}{(3960)(0.52)}\\ &= 28.6\,\text{BHP}\end{aligned}$$

The horsepower can now be plotted as Point G and the horsepower curve for viscous performance drawn through Point G approximately parallel to the break horsepower for water.

3.6 Changing centrifugal pump's performance

3.6.1 Overview

The maximum head that a centrifugal pump can develop is determined by:

- Speed
- Impeller diameter, and
- Number of stages

Thus, to change head of a pump, one or more of the earlier factors must be changed.

Speed can be changed by

- Using different gears, belts or pulleys, or
- Installing a variable speed driver

Impeller diameter can be altered for small performance changes or replaced by larger changes. The number of impellers can be changed by replacing existing impellers with spacers or by inserting dummy impellers.

Small increases can be obtained by "under-filing" the impeller vanes without changing impeller diameter. Under-filing the impeller vanes on the inside will change performance. This means that the exit end of the vanes is filed back, without cutting the shroud (Fig. 3.45).

Fig. 3.27 illustrates the effect on the pump curve of "under-filing" the impeller. The head-capacity characteristic curve for the under-filed condition is for the full diameter vanes. Similar effects are obtained by under-filing any other usable diameter. Under-filing should only be adopted in cases where the standard impeller does not attain the required rating and changing the impeller or using a larger pump is not warranted.

Under-filing pumps with flat curves can lead to a "drooping" (unstable) curve. This would not happen on pumps with "steep" curves. This is a good example of why under-filing should be carefully considered.

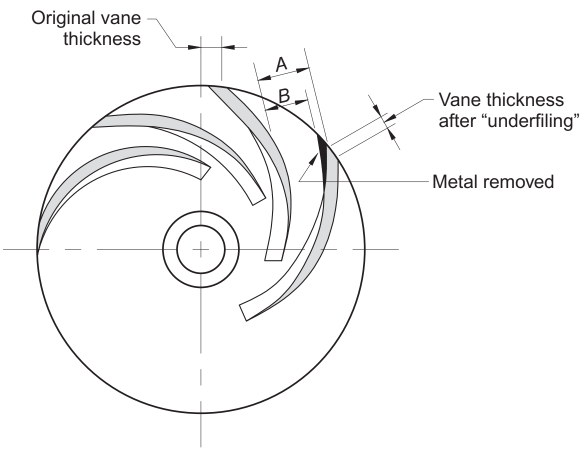


Fig. 3.45 Centrifugal pump impeller with "under-filed" vanes.

3.6.2 Affinity laws

The affinity laws express the mathematical relationships between the several variables involved in pump performance. They are used to predict what effect speed or impeller diameter changes have on centrifugal pump performance. One can trim an existing impeller and the affinity laws will apply to the new conditions. Rather than putting a false head on a pump to make it perform at design conditions, one can trim the impeller to meet the operating conditions. This will save operating costs which can be calculated using the affinity laws. If a system changes, one can retrim the impeller and receive additional energy savings based on the affinity laws. The laws can be summarized as follows:

- Capacity varies directly as the speed or impeller diameter
- TDH varies as the square of the speed or impeller diameter
- BHP varies as the cube of the speed or impeller diameter

The laws are based on dimensional analysis of rotating machines that shows, for dynamically similar conditions, certain dimensionless parameters remain constant. The relationships apply to all types of centrifugal and axial machines.

3.6.2.1 Effects of changing pump speed

Knowing the effects of varying a centrifugal pump's speed is helpful in many situations, such as:

- Adjusting to new service requirements
- Sizing a new driver
- Turning down to avoid excessive flow or pressure

The following affinity law holds for any corresponding points on the head-capacity characteristic curve when the speed is changed. Thus, for a change in pump speed, the following changes in pump performance can be determined:

$$q_2 = q_1 \left(\frac{N_2}{N_1}\right) \tag{3.11}$$

$$TDH_2 = TDH_1 \left(\frac{N_2}{N_1}\right)^2 \tag{3.12}$$

$$BHP_2 = BHP_1 \left(\frac{N_2}{N_1}\right)^3 \tag{3.13}$$

Where:

N_1 = Old speed, rpm
N_2 = New speed, rpm

For the earlier relationships to be true, the efficiency must remain relatively constant for the corresponding point. Efficiency tends to rise slightly as speed increases, because neither hydraulic nor mechanical losses increase as fast as the square of the speed.

The pump characteristic curves illustrated in Fig. 3.28 are marked to show a set of corresponding points for the same impeller at different speeds.

Fig. 3.46 shows a graphic example of reduced operating parameters due to speed reductions.

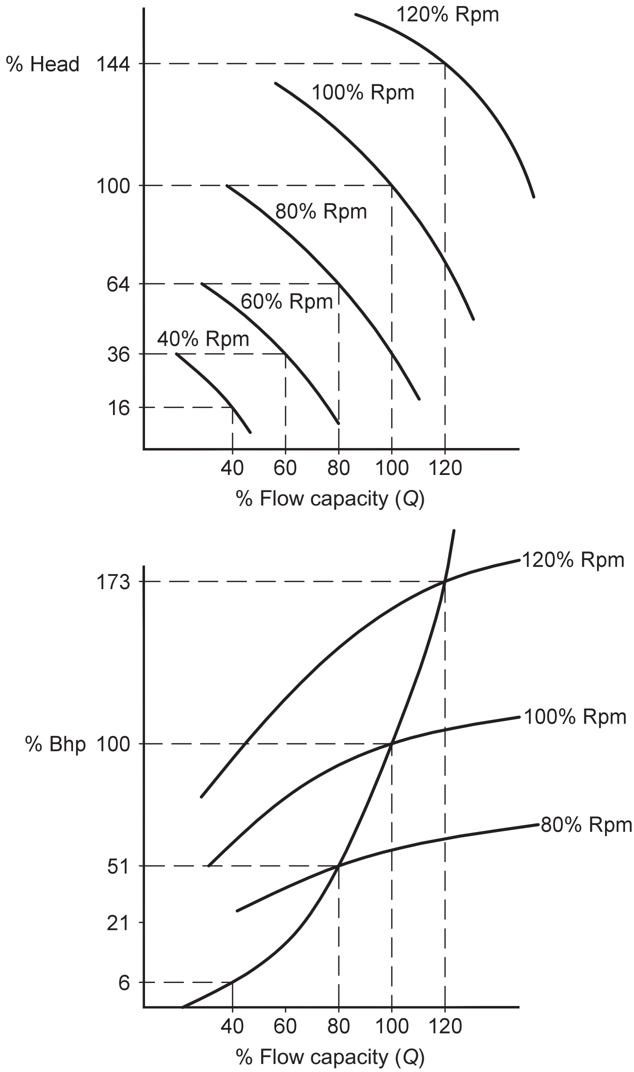


Fig. 3.46 Changed operating parameters due to speed variations.

3.6.2.2 Effects of changing pump impeller diameter

The pump characteristic curves illustrated in Fig. 3.27, except the under-filed curve, may be approximated from a single curve by the following rules, which apply to reducing impeller diameter to the stated design minimum without other changes in design. They are applicable to minor changes (5%–15%) in impeller diameter.

The following affinity laws may be applied for any corresponding points on the characteristic curves when the impeller diameter is changed. Thus, for a change in diameter, the following performance changes can be determined:

$$q_2 = q_1 \left(\frac{D_2}{D_1}\right) \tag{3.14}$$

$$TDH_2 = TDH_1 \left(\frac{D_2}{D_1}\right)^2 \tag{3.15}$$

$$BHP_2 = BHP_1 \left(\frac{D_2}{D_1}\right)^3 \tag{3.16}$$

Where:

D_1 = Old diameter, (in.)
D_2 = New diameter (in.)

These laws are essentially the same as the affinity law for speed change but do not apply with the same accuracy over as wide a range.

For the earlier relationships to be true, the efficiency must remain constant for the corresponding point. Since this is not exactly what happens, the head calculated by the earlier relationship will usually be low. The efficiency will usually drop. Engineering judgment and field experience should be used to estimate how much deviation from the relationship should be expected.

The affinity law for impeller diameter applies not only to the BEP, but to any corresponding points on the original and calculated new head–capacity characteristic curve, provided they are not affected by suction conditions.

3.6.2.3 Effects of changing both pump speed and impeller diameter

For a change in both impeller diameter and speed, the following changes in pump performance can be determined:

$$q_2 = q_1 \left(\frac{D_2}{D_1}\right)\left(\frac{N_2}{N_1}\right) \tag{3.17}$$

$$TDH_2 = TDH_1 \left(\frac{D_2}{D_1}\right)^2 \left(\frac{N_2}{N_1}\right)^2 \tag{3.18}$$

$$BHP_2 = BHP_1 \left(\frac{D_2}{D_1}\right)^3 \left(\frac{N_2}{N_1}\right)^3 \tag{3.19}$$

Predictions for speed changes are fairly accurate throughout the range of speed changes. However, they tend to be accurate for diameter change of ±10%, because changing the diameter also changes the relationship of the impeller to the pump casing. Thus, for a 10% increase in either diameter or speed, the flow will increase by 10%, TDH by 21%, and the BMP by 33%. Because of the cube relationship between impeller diameter and/or speed and power, a pump's speed cannot be arbitrarily increased, but these conversions for slowing a pump down or decreasing the diameter can be used.

Example 3.2. Application of affinity laws
Given:

$D_1 = 6$-in. (150 mm) diameter
$q_1 = 100$ gpm (22.7 m^3/h)
$TDH = 100$ ft (30 m)
$N = 1800$ rpm (rps)
$BHP = 20$ hp (14.9 kW)

Determine:

q_2, *TDH,* and *BHP* when impeller diameter is increased to 8 in. (200 mm) at the same rpm.

Solution:

Field units

$$q_2 = q_1 \left(\frac{D_2}{D_1}\right) = 100 \left(\frac{8}{6}\right) = 133 \text{ gpm}$$

$$TDH_2 = TDH_1 \left(\frac{D_2}{D_1}\right)^2 = 100 \left(\frac{8}{6}\right)^2 = 177 \text{ ft}$$

$$BHP_2 = BHP_1 \left(\frac{D_2}{D_1}\right)^3 = 20 \left(\frac{8}{6}\right)^3 = 47 BHP$$

SI units

$$q_2 = q_1 \left(\frac{200}{150}\right)(22.7) = 30.3 \text{m}^3/\text{h}$$

$$TDH_2 = \frac{(200)^2}{(150)^2} = 53.3 \text{m}$$

$$BHP_2 = \frac{(200)^3}{(150)^3} = 35.3 \text{kW}$$

3.6.3 Effects of specific gravity on pump's performance

Keeping speed and impeller diameter constant, the specific gravity has the following on pump performance:

- Flow rate is unchanged by specific gravity
- Pressure varies directly with specific gravity
- Power (BHP) varies directly with specific gravity

The earlier relationships are important when converting a pump from one service to another or if significant changes to fluid gravity are anticipated, for example, converting from a light hydrocarbon service to produced water service may significantly overload an existing driver.

3.7 Pump specific speed

3.7.1 Definition

The *pump specific speed* (N_S) is a dimensionless term used to compare the performance and shape of impellers of two centrifugal pumps that are geometrically similar. The *pump specific speed* is the speed (rpm) at which a geometrically similar impeller would run if it were of such size as to discharge one gallon per minute (gpm) against a TDH of one foot. It is a tool pump designers use to describe geometry of the impeller and to classify them as to their type: radial flow, mixed flow, or axial flow.

In practice, *pump specific speed* is used to relate the three main parameters (gpm, TDH, and rpm) to the performance of the pump. *Pump specific speed* can be calculated from the following equation:

Field units

$$N_S = \frac{N\sqrt{q}}{(TDH')^{0.75}} \tag{3.20a}$$

SI units

$$N_S = \frac{51.65N\sqrt{q}}{(TDH')^{0.75}} \tag{3.20b}$$

Where:

N_S = Pump specific speed
N = Pump speed, rpm (rps)
q = Pump capacity, gpm (m^3/h)
THD' = TDH per stage at best efficiency, ft (m)

Eq. (3.20a) and (3.20b) is derived such that it is independent of pump dimensions. The *pump specific speed* is always calculated at the pump's point of maximum efficiency. The number is used to characterize a pump's performance as a function of its flowing parameters (Fig. 3.47).

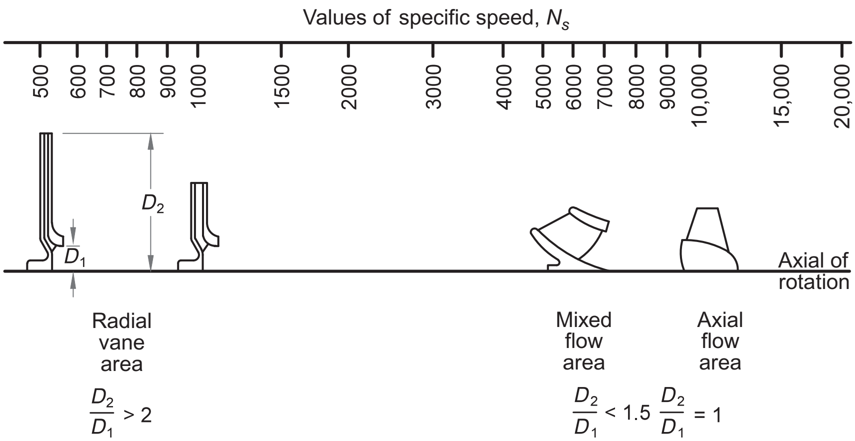


Fig. 3.47 Impeller design vs pump specific speed.

Pump specific speed is a dimensionless number derived from Eq. (3.20a) and (3.20b). This number is not very useful for equipment engineers, operators, or maintenance personnel. However, it is an extremely valuable tool for pump designers in the development of impellers. Impeller design must take into account these parts of the impeller:

- *Hub and eye*: Determine the inlet velocity and NPSH required by the pump.
- *Shrouds*: Affect the overall efficiency.
- *Inlet vane angle*: Determines recirculation flow and pumps capability to operate at off-design conditions.
- *Vane curvature and thickness*: Affects capacity and efficiency.
- *Vane width*: Affects capacity in that the width affects the cross-sectional area.
- *Number of vanes*: Affects efficiency, head capability, and capacity.
- *Outlet vane angle*: Determines head and efficiency. Under-filling vane tips will increase head and efficiency slightly. However, the cross-section of the vane tip will be reduced, resulting in premature vane tip wear and premature failure of the vane.

There are two points that must be defined. The first is the design point of a pump. This is the actual head, capacity, and rotative speed that the design engineer uses to design a pump. In theory, there are an infinite number of head-capacity points for which pumps can be designed. To be practical, the pump designer makes some compromises to produce pumps that can operate over wider ranges. Compromises might lower the peak efficiency but give a broader band of relatively high efficiency. For example, a perfectly designed pump might have its best efficiency of 80% at 1000 gpm, where at 950 or 1050 gpm, the efficiency drops to 70%. To compromise the design, the pump designer might lower the peak efficiency to 78% and raise the efficiencies at 950 and 1050 gpm to 77%. If the engineer was designing a pump for one specific operating condition, he would be interested in only one point.

The second point that must be defined is rated condition. This is the actual head-capacity point for which a pump is purchased and should be used by all people subsequent to the design engineer—the salesman who sells the pump, the user who buys the pump, and operations personnel who actually uses the pump.

The lower the pump specific speed the more the impeller shape will approach true radial flow. The higher the pump specific speed the more closely the impeller geometry approaches true axial flow.

Low-pump specific speed impellers have high heads and low flow capacities. Impellers for low heads and high flow rates have high pump specific speeds.

3.7.2 Impellers with low specific speeds (500–4000)

Pumps with low *pump specific speeds* are described as radial flow impellers. The pumped fluid undergoes a 90 degrees turn from inlet to outlet. They are characterized by narrow and relatively large in diameter. They exhibit high TDHs (700–1000 ft) and low flow capability. Shutoff heads range from about 120%–150% of the head at the BEP. Most commonly used in production operations.

3.7.3 Impellers with medium specific speeds (4000–10,000)

Pumps with medium specific speeds are described as mixed flow impellers. The impellers are wider and smaller in diameter than radial flow impellers. They exhibit medium TDH (100–150 ft) and medium flow capability. The shutoff head is 150%–280% of the head at the BEP. Only used in vertical multistage pumps where the diameters are small (down-hole electric submersible pumps).

3.7.4 Impellers with high specific speeds (10,000–16,000)

Pumps with high specific speeds are described as axial flow impellers. The fluid does not turn away from the inlet direction. They exhibit high flow and low TDH capability and a continuously declining head curve. They are used primarily in very high volume, low TDH water irrigation, flood control, pumped storage power generation projects, and as ship propellers.

3.7.5 Selection considerations

Selection should be based on total lifecycle cost not just initial purchase cost. Impeller diameter is one of the primary factors that determine pump purchase costs. Normally, it is desirable to select the impeller with the higher specific speed (smaller diameter). This may be offset by the higher operating cost associated with higher speeds and greater susceptibility to cavitation damage.

3.7.6 Validation of vendor quoted efficiencies

Fig. 3.48 is a graph showing the general relationship between impeller shape, efficiency, and capacity. Each impeller design has a specific speed range for which it is best adapted. These ranges are approximate, without clear-cut demarcations

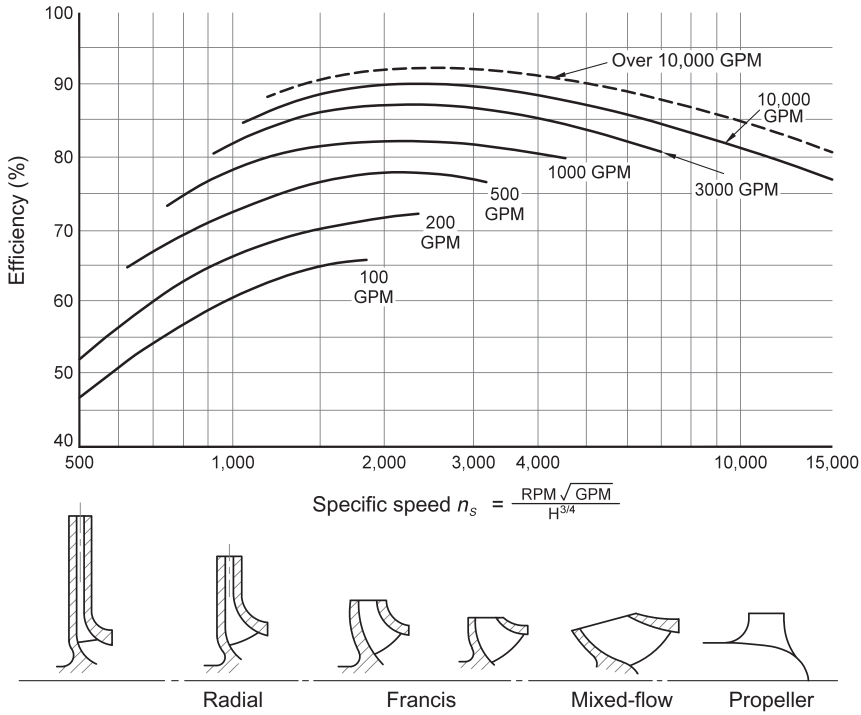


Fig. 3.48 Relationship of impeller shape, efficiency, and capacity.
Courtesy of Pump Handbook, Karassik, Krutzch, Frazer and Messina, McGraw Hill, 1976.

between them. Most pumps used in upstream production operations are designed with impellers that have specific speeds between 800 and 1500 (as calculated from Eq. (3.20a) and (3.20b)).

Pump specific speed is a pump design tool, but may be used in the pump selection process to compare the curve shape and stability. Fig. 3.48 can be used to validate vendor's quoted efficiencies. Generally, low pump specific speeds indicate flat head-capacity curves, with peak efficiency over a wide range of capacity, and a break-horsepower decreasing as the pump is throttled. High pump specific speeds result in steep head-capacity curves, sharply peaked efficiency curves, with brake-horsepower increasing as the pump is throttled.

3.8 Suction specific speed

3.8.1 Definition

Suction specific speed (N_{ss}) is another design index used by pump designers and equipment engineers. It describes the geometry of the suction side of the impeller. The *Suction specific speed* is a dimensionless number that describes the characteristics

of an impeller, not on terms of the conditions existing at the discharge of the pump (head). *Suction specific speed* can be calculated from the following equation:
Field units

$$S_{SS} = \frac{N\sqrt{q}}{(NPSH_R)^{0.75}} \tag{3.21a}$$

SI units

$$N_{SS} = \frac{51.65N\sqrt{q}}{(NPSH_R)^{0.75}} \tag{3.21b}$$

Where:

N_{SS} = Suction specific speed
N = Pump speed, rpm (rps)
q = Pump capacity at BEP for the maximum diameter impeller, gpm (m^3/h)
= Divided by two for double-suction impellers
$NPSH_R$ = Net positive suction head required at point "q," feet (m) (supplied by the manufacturer)

Very similar to pump specific speed, the $NPSH_R$ and q are measured at the point of best efficiency with the largest diameter impeller available. For double-suction pumps q should be one half of the total flow.

N_{SS} is used to consider a pump's ability to operate away from its design point. The pump design engineer theoretically designs a pump for one operating condition which is at its BEP for the maximum diameter impeller that can be installed in the pump casing. How far back on its performance curve a pump can operate without damage is determined by its impellers suction specific speed. N_{SS} is determined for Eq. (3.21a) and (3.21b). Note that if the pump has a double-suction impeller, q is divided by two as half of the flow enters through each impeller eye.

Also note that the larger the $NPSH_R$ value, the lower the N_{SS} value. The lower the N_{SS} value, the better a pump can operate back on its curve without damage to the pump. All pumps at low flow will recirculate liquid in the eye of its impeller, which is an increase in velocity and a decrease in liquid pressure. The higher the N_{SS} value of the pump's impeller, the sooner this recirculation occurs. The damage caused to the impeller, casing, bearings, and seals is similar to that caused by cavitation.

N_{SS} values less than 10 are generally acceptable for water-based liquids while N_{SS} values less than 11,000 are generally acceptable for hydrocarbons. Other factors such as component metallurgy and operating swings must also be considered.

Generally, when specifying a pump these indexes are not necessarily needed. In certain instances, special considerations will make them useful. Always consult the manufacturer, especially when specifying an impeller other than standard design.

- For double-suction impellers, q = ½ BEP flow.
- Based on inlet flow angle.
- Low flow angle for best NPSH, higher for best efficiency.
- N_{SS} = 9000 is typical.

- 5000–8000 also common.
- Double suction use ½ the flow.
- Many specifications limit N_{SS} to below 11,000.
- Higher N_{SS} usually has fewer vanes, less blockage.
- Suction eye is typically large for high N_{SS}.
- Higher N_{SS} pumps typically do not run well over a wide operating range.

The $NPSH_R$ by a pump is largely dependent on the impeller "eye area" and inlet "vane angle" design. These relatively complicated and proprietary design features can easily be evaluated by comparing each pump's suction specific speed (N_{SS}).

Typical values for N_{SS} range between 7000 and 14,000 as determined by pump design. However, conservative impeller designs will have a N_{SS} value less than 11,000. Multistage, high-energy pumps which operate above 3600 rpm should have a first-stage impeller N_{SS} value of less than 9000.

As the N_{SS} increases, the pump $NPSH_R$ decreases, and the pump minimum flow increases. Field experience indicates that pump reliability is directly related to the pump N_{SS}. Pumps with N_{SS} values above 11,000 are less reliable. The lower reliability usually manifests itself as high vibration and shaft deflection due to flow instability in the impeller eye. The shaft deflection and vibration results in reduced mechanical seal and bearing life.

3.8.2 Low suction specific speeds

Pumps with low Suction Speeds have a lower percentage of its best efficiency flow required for its minimum flow (better turndown ratio) and have the least chance of cavitation.

3.8.3 High suction specific speeds

Pumps with high *suction specific speeds* increase the design sophistication. Pumps with *suction specific speeds* are as follows:

- Up to 8500 rpm, pump models are widely available and easy to design.
- From 8500 to 10,000 rpm, pumps require more refined engineering design, but sufficient sources exist for competitive designs.
- From 10,000 to 12,000 rpm, pumps have to be highly engineered and should be secured from those manufacturers with established experience

If a pump's *suction specific speed* is too high, it can be lowered by:

- Reducing pump speed
- Increasing the number of impellers, or
- Using a double-suction pump

3.9 Suction considerations

If a pump is to operate successfully it must have enough suction pressure to push the liquid into the pump without flashing, cavitation, or boiling. This is particularly critical where liquids are near their boiling points (flash separators, boiler feedwater, etc.).

Failure to provide adequate suction pressure will lead to a variety of mechanical and operational problems, including complete pump failure.

3.9.1 Flashing (boiling)

A pump should be selected with inlet velocities sufficiently low to prevent vapor formation in the entering liquid. This may call for:

1. Oversized inlet piping
2. Pumps operating at low speed
3. Pumps designed for such conditions
4. Use of vertical pumps installed in a suction can

The pressure at the suction of the pump must be adequate to accelerate the liquid to the required velocity at the impeller entrance without the pressure in the pump falling below the fluid's vapor pressure. Flashing (boiling) of the fluid in the pump suction eye is called cavitation and can significantly affect pump performance.

3.9.2 Cavitation

The formation of vapor bubbles in the impeller suction eye due to fluid flashing (boiling), with subsequent collapse of the bubbles as the pressure rises, is called "cavitation" (see Fig. 3.49). Cavitation may cause vibration, pitting damage, and impaired performance. Cavitation may or may not be serious depending on the pump, BHP/stage, impeller design, and the fluid being pumped. In small pumps with low differential head per stage, the energy of collapsing bubbles is much less than in larger,

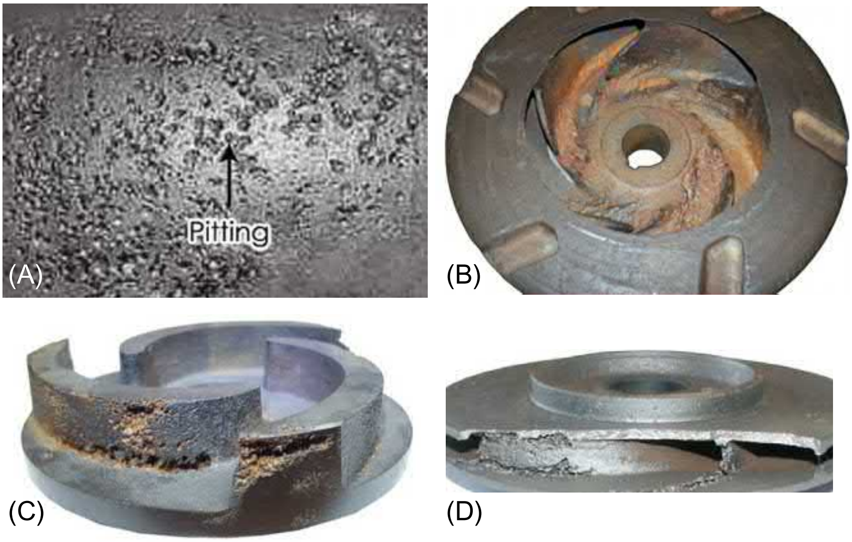


Fig. 3.49 Cavitation damage: (A) impeller pitting; (B) cavitation at impeller inlet; (C) cavitation of inlet vanes; (D) cavitation at impeller eye and inlet vanes.

high-head-per-stage pumps. Cavitation is more severe in a single-boiling point fluid (e.g., water) than with a mixture (e.g., most hydrocarbons) that have a broad boiling range.

3.9.3 Recirculation

Recirculation is a flow reversal at the inlet eye or discharge tip of an impeller. Recirculation at the inlet eye is called suction recirculation. Discharge recirculation occurs at the impeller tip. Recirculation usually occurs when operating centrifugal pumps at flows below their best efficiency flow.

All impellers will begin to recirculate at a certain flow rate. The point at which recirculation begins may not be the same for suction and discharge. Suction recirculation usually will begin at a higher flow than discharge recirculation. The capacity at which recirculation occurs is determined primarily by the impeller design. Most of the problems associated with recirculation can be avoided by selecting pumps with impellers of low N_{SS} designs. Recommended limits for N_{SS} are as follows:

BHP per stage	<250 to 300	>300
N_{SS} limit	11,000	9000

The exact flow at which suction recirculation (the most frequent cause of problems) occurs depends on impeller design. The larger the impeller eye diameter to outside diameter ratio, the lower the $NPSH_R$, and the higher the capacity at which recirculation occurs as a percentage of the capacity BEP.

Suction recirculation causes the formation of very intense vortices. When vortices collapse, severe noise and pressure pulsations of 20% or more in the pump discharge occur. This can be damaging to the operation of the pump and to the integrity of the impeller (Fig. 3.50). Normal cavitation damages the visible portion of the impeller

Fig. 3.50 Cavitation damage as a result of low flow recirculation.

(when held up and viewed from one end). Suction recirculation damages the hidden side which must be viewed with the aid of a mirror.

Critical factors used to classify recirculation are as follows:

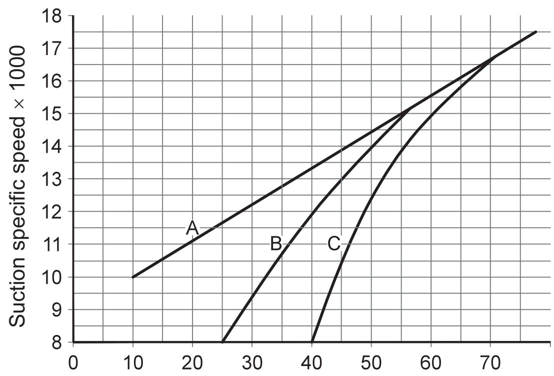
- Suction specific speed (S_{ss})
- NPSH margin
- Pumped fluid
- Power intensity factor (defined as BHP/stage/inch of impeller diameter)

Figs. 3.51 and 3.52 can be used to estimate the point at which recirculation will occur. Fluid characteristics do not affect the flow at which recirculation occurs, but do have a pronounced effect on the severity of the symptoms and the resulting damage. For example, damage to pumps will be less when handling hot water or hydrocarbons than when handling cold water.

The effects of recirculation can be impeller and casing damage, bearing failures, and seal or shaft failures. Symptoms associated with recirculation are listed as follows:

Suction recirculation:

- Cavitation damage to the pressure side of the impeller vanes at the inlet of the vane.
- Cavitation damage to the stationary or splitter vanes in the suction side of the pump casing.
- Random cracking or gravel pumping noise (inadequate NPSH sounds the same except the noise will be constant not random).
- Surging pressure in the suction pipe.



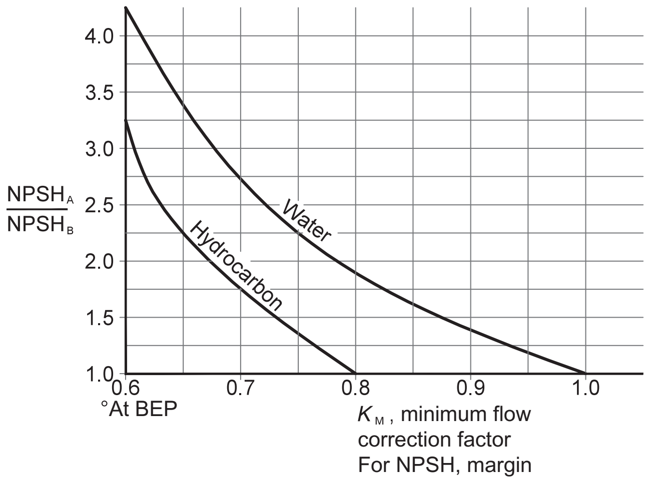
Percent of best efficiency point capacity at maximum diameter impeller correction factors on fig. 3.52

A — Single stage, overhung, 2" discharge and smaller multistage, 2" discharge and smaller

B — Single stage, overhung, 3"–4" discharge (3600 rpm), 6" discharge and larger (1800 rpm and lower) single stage, double suction, 4" discharge and smaller, multistage, 3" discharge and larger

C — Single stage, overhung, 6" discharge and larger (3600 rpm) single state, double suction, 6" discharge and larger

Fig. 3.51 Minimum continuous stable flow vs. suction specific speed for centrifugal pumps in general refinery services.



Note: 1. *N*, is always defined at the pump best efficiency point or maximum design impeller diameter

2. $N = \dfrac{\text{RPM}\sqrt{\text{GPM}}}{(\text{NPSH})^{1/4}}$

3. GPM is per eye: for double suctions pumps, use one-half the total pump flow or BEP

4. When determining NPSH margin, use NPSH, at the BEP FLOW

Fig. 3.52 Minimum flow correction factor for $NPSH_A$ margin.

Discharge recirculation:

- Cavitation damage to the pressure side of the impeller vane and exit shroud at the discharge of the impeller. This may be seen as impeller at the impeller vanes or shroud.
- Higher than normal axial vibration or shaft movement. This may be accompanied by thrust bearing damage.
- Cavitation damage to the "cut water" (casing tongue) or diffuser vanes in the case.

Unless there is a compelling reason to do so, do not specify $NPSH_R$ values much above 9000. When a pump is not expected to operate at flows below its BEP for constant speed and unthrottled flow, higher N_{ss} values can be used. Pumps handling hydrocarbons can operate at lower flows than similar pumps handling cold water. In the case of small pumps ($<$100 HP), the effects of recirculation are not likely to be as damaging. A backflow recirculation or stabilizer can be used. The risks of operating in the area where recirculation exists can best be determined after the pump is in operation. Provision, therefore, should be made to increase the minimum flow bypass capacity.

3.10 Regulating flow rate

It is unusual for a pumping system to require operation at a single fixed flow rate. A pump will only deliver the capacity that corresponds to the intersection of the head-capacity and system head curves. To vary the capacity, one must change the shape of one or both curves. A pump head-capacity curve shape can be changed by altering the pump speed or impeller diameter. The system head curve shape can be changed by the use of a backpressure (FCV) throttling valve.

Centrifugal pumps operate best when flow rate exceeds 40%–50% of the best efficiency flow. Deviation from this range can cause heat buildup, excessive vibration, damage, and failure. The effects of operating at or below significantly reduced capacity may lead to the following undesirable results:

- Operating at much less than the BEP
- High bearing loads
- Temperature rise
- Internal recirculation

The effects of operating less than BEP result in lower power requirements if the BEP flow is not required and higher energy consumption per unit capacity. The earlier effects can be avoided by either using a variable speed driver or several parallel pumps for the total capacity and sequentially shutting down individual units as demand requires.

Higher bearing loads will exist for any flow away from BEP, especially for single-stage, single-suction pumps. This can be anticipated by specifying certain types of heavy duty and long life bearings.

If the temperature of the pumped fluid rises and the flow rate through the pump decreases, the installation of a minimum flow bypass can be used. The manufacturer generally provides the minimum continuous required flow rate for any pump selection.

Operating between BEP and minimum required flow generally avoids all of the earlier problems. Operating at or below the minimum flow is critical for high-speed pumps (such as Sundyne) since vibration can quickly cause gearbox damage. Fig. 3.53

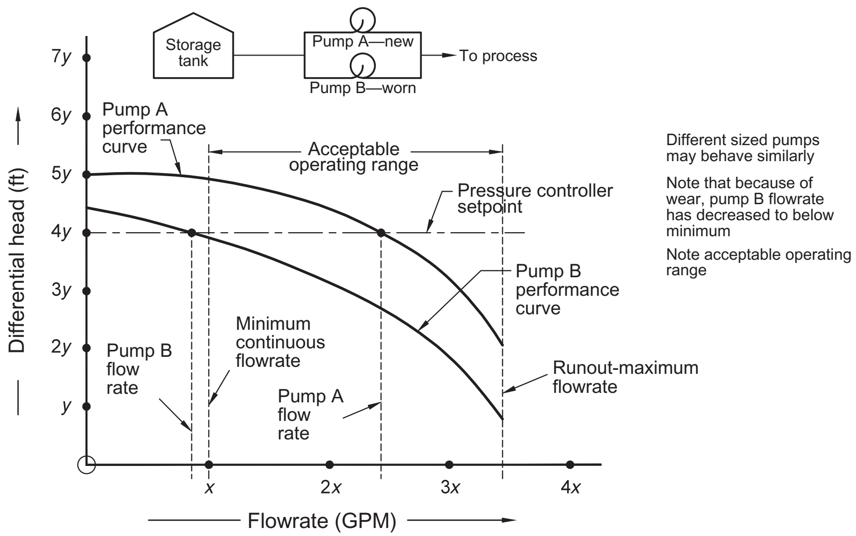


Fig. 3.53 Effect of wear on pump performance.

is an example illustrating the minimum and maximum flow operating range for a pump.

Several methods of controlling the pump flow rate are used to prevent operation outside the designed range. These methods include (refer to Figs. 3.54, 3.55, 3.59 and 3.60):

- pressure control
- flow control
- variable speed

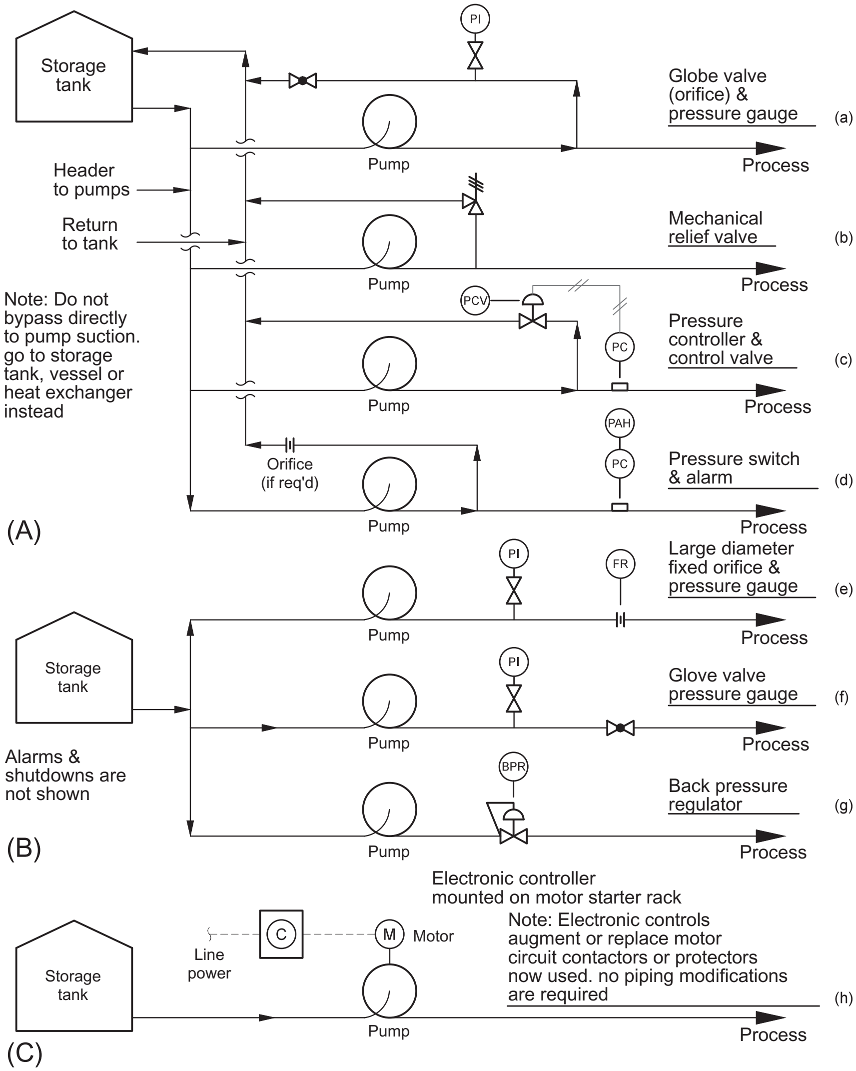


Fig. 3.54 Pressure control methods: (A) maintaining minimum continuous flow; (B) preventing run-out; and (C) electronic controls.

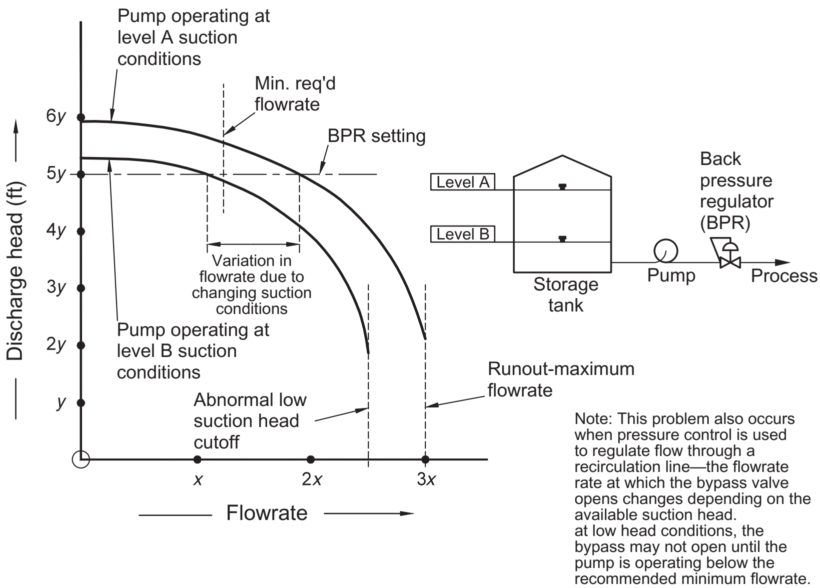


Fig. 3.55 Effect of variable suction head on pressure control in a centrifugal pump.

3.10.1 Pressure control

Pump controls circulate fluid from the pump discharge back to a suction vessel or tank to maintain a minimum flow rate, or they impose backpressure on a pump to prevent run-out. Run-out is defined as operating beyond a pump's maximum recommended flow rate. Run-out is most likely to be a problem when discharge lines are short (no friction loss) or when pumping into a system with low backpressure (e.g., an empty tank).

The minimum flow by pass prevents the buildup of excessive amounts of heat within the casing and reduces the level of variations that would otherwise exist. Cases where a minimum flow bypass valve should be installed include:

- As flow requirements change during normal operations, the backpressure valve may close off to less than the minimum continuous stable flow at which the pump can safely operate
- Pump may have an automatic closing valve that fails in the closed position or a discharge block valve that can be inadvertently closed

Minimum flow bypasses should be located upstream, of the first block, control or check valve. A small continuous recycle of fluid through an orifice is routed from the discharge side (on small pumps) back to the suction source or a second control valve (on medium and large pumps) is directed to open on either low pressure or low flow rate.

Pressure control methods include using a bypass controlled by one of the following:

- a globe valve (or a fixed orifice)
- a mechanical pressure safety valve (PSV)
- a proportional pressure controller (PC) with a pressure control valve (PCV)
- a pressure safety high (PSH) alarm
- self-contained backpressure regulators are used to prevent run-out.

3.10.1.1 *Fixed recirculation through a globe valve (or fixed orifice) (refer to "a" in Fig. 3.54)*

A globe valve or fixed orifice is mounted with a pressure indicator in a bypass line from the pump discharge back to the storage vessel or tank. When using a recirculation bypass, never return fluid directly back to the pump suction as this will cause swirling and heating problems which will raise the vapor pressure and affect the $NPSH_A$. Instead, route the bypass line back to the vessel, tank, or heat exchanger.

Proper recirculation is established by keeping the discharge pressure below that corresponding to the manufacturer's minimum continuous flow rate. This is usually an inexpensive approach but operating costs may be high due to energy losses across the valve or orifice. In addition, continuous recirculation should be specified at the time the pump is purchased to ensure sufficient capacity for both process and bypass flow rates. This is the preferred approach for low-energy pumps, for example, less than 10 HP.

As shown in Fig. 3.56, the difference between the TDH developed by the pump and the head required by the system head curve represents lost energy. Since the majority of centrifugal pumps are driven by constant speed electric motors, throttling is the only practical method of regulating capacity.

The backpressure valve imposes a variable amount of control losses on the system head curve. Closing the valve increases control losses and causes the system head curve to slope up more steeply to intersect the TDH capacity curve at the desired capacity.

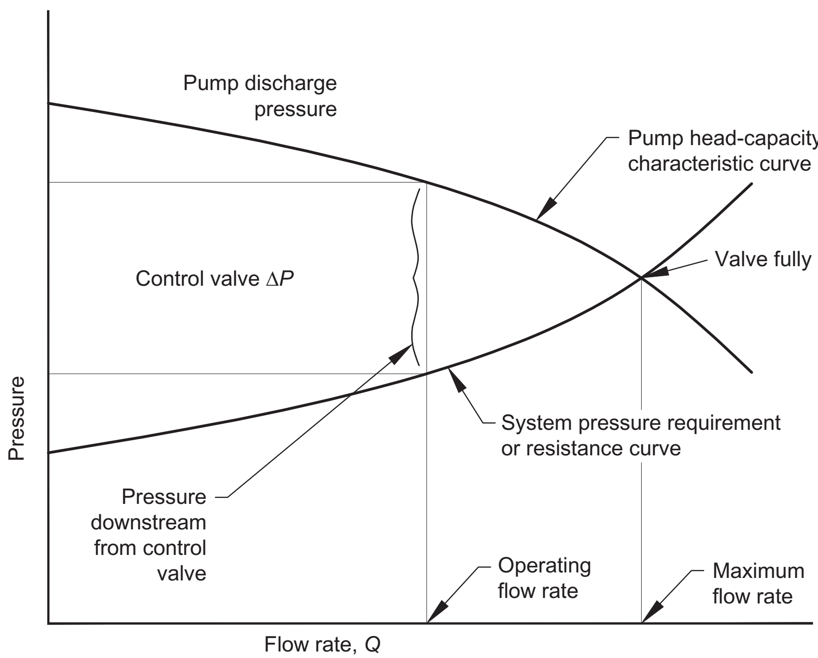


Fig. 3.56 Globe valve (FCV) regulating the flow rate of a constant speed centrifugal pump.

Opening the valve decreases the control losses and causes the system head curve to slope downward and intersect the TDH capacity curve at a higher capacity. With the valve completely open, the capacity is only governed by the intersection of the two curves.

3.10.1.2 Variable recirculation through a pressure safety valve (refer to "b" in Fig. 3.54)

A PSV is installed to bypass discharge back to the storage vessel or tank. The set pressure is the pressure corresponding to the manufacturer's minimum continuous flow rate. As the discharge pressure rises, and flow decreases, the PSV opens, circulating flow back to suction, maintaining the required minimum flow rate.

PSV's reseat below their set point and this may not be acceptable. After it relieves, a PSV may stay open or simmer, even as the pump returns to operation above the minimum flow. A PSV is therefore not recommended except for positive displacement (PD) pumps, and then not for minimum flow protection, only pump casing relief protection.

3.10.1.3 Proportional pressure control (refer to "c" in Fig. 3.54)

A pneumatic or electronic pressure controller is used to sense pump discharge pressure and control a bypass control valve. As discharge pressure rises, pump flow decreases toward the manufacturer's recommended minimum. The controller senses the pressure rise and opens the bypass control valve, maintaining the flow rate above minimum.

3.10.1.4 PSH alarm (refer to "d" in Fig. 3.54)

PSHs installed in the discharge piping identify high discharge pressures (an indication of low flow) and also low discharge pressure (loss of prime or surging). PSHs are usually inexpensive and can be connected to an alarm or automatic shutdown of the pump.

3.10.1.5 Backpressure regulator (refer to "g" in Fig. 3.54)

Run-out can be prevented with a backpressure regulator. Backpressure regulators are self-contained and easily field adjustable. The set-point changes are made through spring replacements or by changing the spring preload.

3.10.1.6 When not to use pressure control

Pressure control is usually less expensive than flow control. That said, pressure control does not work well under the following listed conditions. In the following listed conditions, flow control is usually the best choice, even though it is more expensive.

3.10.1.6.1 Pumps with variable suction conditions

If suction pressure changes, the discharge pressure will also change. Thus, if the suction vessel or tank level is high, the discharge pressure will be higher than if the vessel or tank is nearly empty. Fig. 3.55 illustrates this by plotting the pump discharge head vs. flow for high- and low-suction tank conditions. The control valve, however, tries to maintain a steady pressure. As a result, the flow rate varies as the suction pressure varies. In some cases, the flow rate may drop to below the recommended minimum at very low head conditions. Therefore pressure controls may be used only when the pump suction pressure is stable, or when the suction pressure variations cannot move the flow rate below the manufacturer's recommended minimum.

3.10.1.6.2 Pumps with flat pump curves

Pressure control works best when a large change in pressure affects flow. The opposite occurs with flat pump curves, common for multistage pumps. In this case, using pressure to control flow is inaccurate, because a small pressure change results in a large change in flow (refer to Fig. 3.57).

3.10.1.6.3 Pumps in parallel, each with a dissimilar curve

Dissimilar curves can occur from different pumps or from identical pumps, when one is worn internally. The difference in the performance curves can allow one pump to move more liquid than the other, with the result that one pump may operate below the manufacturer's recommended minimum flow rate.

3.10.1.6.4 Pumps with "drooping" head curves

A "drooping" pump head-capacity curve has a dome at the top of the curve where one differential head value corresponds to two flow rates (Fig. 3.58). The pump may oscillate between the two flow rates while maintaining the same differential head. When two or more drooping curve pumps are connected in parallel, one pump could operate below minimum flow rate while the other operates well out on its own.

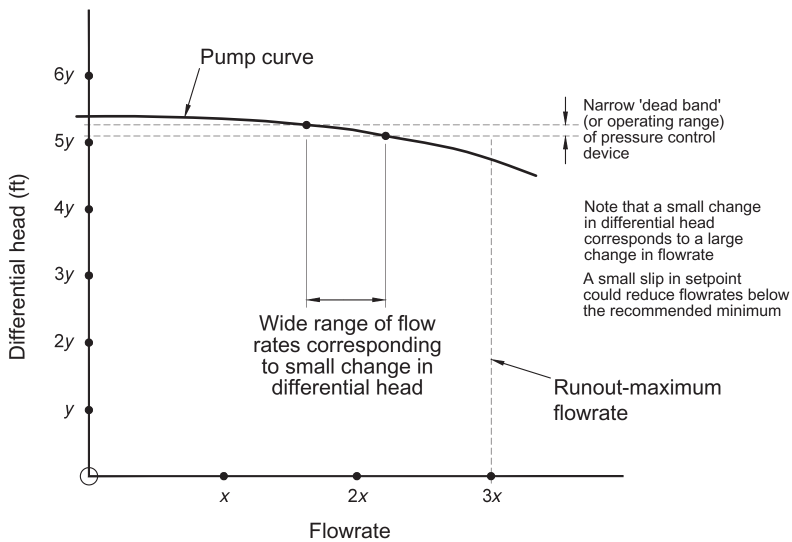


Fig. 3.57 Problems with pressure control on a "flat" centrifugal pump performance curve.

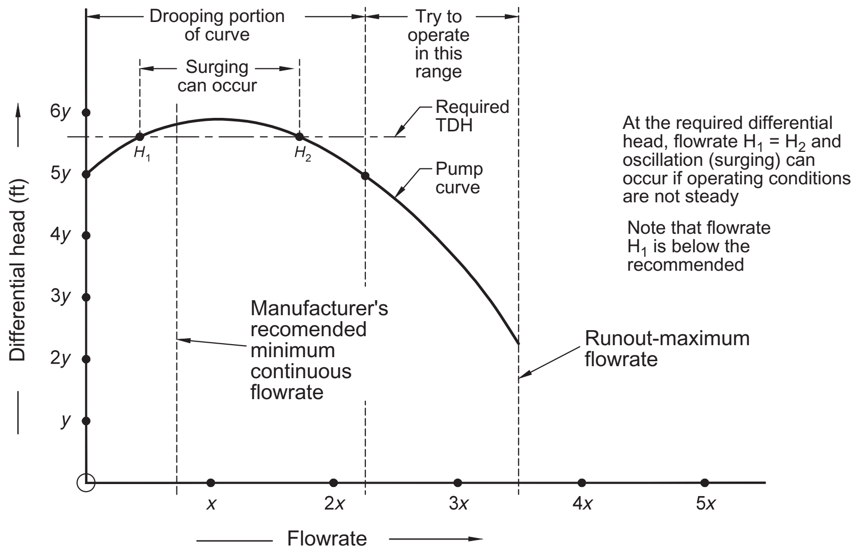


Fig. 3.58 Centrifugal pump "drooping" curve.

3.10.2 Flow control

Flow control is used to maintain minimum flow rates and to prevent pump run-out. The advantages are as follows:

- Better accuracy than pressure control since pump flow is measured directly
- Flow control is unaffected by fluctuating suction pressure. It is effective on pumps with flat or drooping curves or parallel pumps with dissimilar performance curves.

Flow control usually costs more than pressure control, unless flow is to be measured for other process reasons. Methods of flow control include a bypass with one of the following:

- globe valve or orifice
- automatic recirculation (ARC) valve
- proportional control using a flowmeter
- proportional controller/control valve
- snap-acting solenoid valve

Run-out can be prevented by either a self-contained, pilot operated, diaphragm valve, or proportional controller/valve arrangement mounted in the pump discharge piping.

3.10.2.1 Fixed recirculation through a globe valve or orifice (refer to "a" in Fig. 3.59)

A globe valve or orifice is mounted with or without a meter in a bypass line from pump discharge back to the storage vessel or tank. This is an expensive approach. Energy losses occur across the orifice or valve. Continuous recirculation should be specified

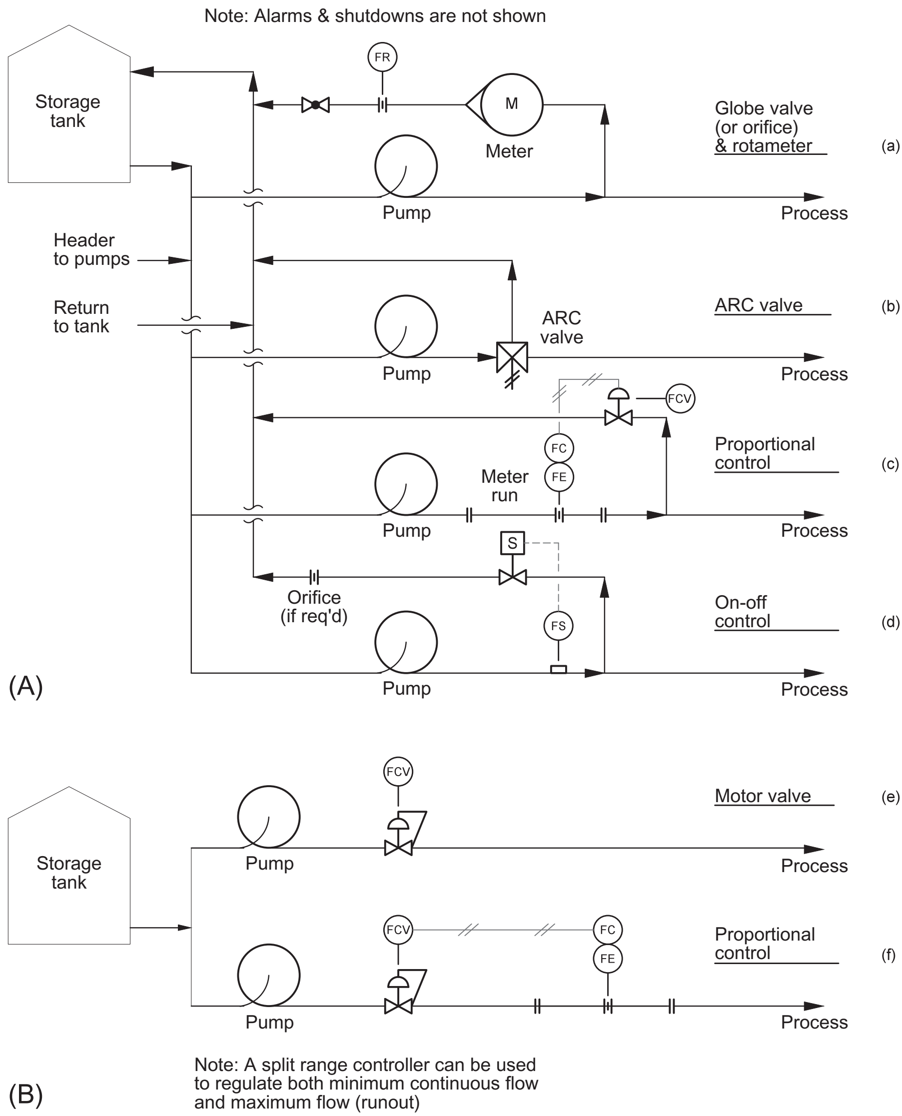


Fig. 3.59 Flow control methods: (I) maintaining minimum continuous flow; (II) preventing run-out.

when pumps are purchased to ensure sufficient capacity for both the process flow rate and the bypass flow rate. If continuous recirculation is added as a retrofit, the pump may tend to operate too far near the right end of its curve. Due to the energy costs, this is only recommended for low-energy services, for example, 10 HP. Note: The globe valve could be completely closed, leaving the pump unprotected.

3.10.2.2 ARC valves (refer to "b" in Fig. 3.59)

ARC valves are often used for flow control, primarily with Sundyne high-speed, integral gear pumps. These valves discharge a sidestream back to suction when flow falls below the recommended minimum. Because ARC valves recirculate only when they have to, they minimize energy loss. However, ARC valves are only available in ASME Classes 150 and 300 and are costly.

ARC valves are most economical with high bypass flow rates and larger differential pressures. Under these conditions, recirculation with either a fixed restriction or globe valve bypass becomes too expensive due to energy losses. ARC valves can also be used when a pump curve will not allow the additional flow required to satisfy the process and fixed-recirculation bypass.

3.10.2.3 Economics

The economics of minimum flow systems depend on the bypass flow rate, differential head, and the equipment already in place:

- Fixed-orifice recirculation is inexpensive and probably the best method at low flow rates and low differential pressures. It uses only a valve or orifice with or without an indicator, pressure gage, or rotor meter. However, it constantly wastes energy across the valve or orifice and can be expensive to operate with high head pumps or with pumps that require a large minimum flow rate.
- Flow control through power monitoring has the same problems inherent in pressure control. Except for applications involving very low flow rates and pressures, this method may be the least expensive. This method is suited for retrofitting existing equipment because no piping changes are required and pump operating conditions will not change.
- On-off bypass control using flow switches and a solenoid valve is simple and relatively inexpensive. It does not waste energy and does not allow the pump to operate at the end of its curve.
- ARC valves and proportional controls are costly. Restrict their use to critical, unattended, nonspared, high head pumps or pumps with large minimum required flow rates.

3.10.3 Variable speed

The majority of motor-driven centrifugal pumps are operated at constant speed. A DC or variable frequency AC motor control can maintain nearly the same pump efficiency over its speed range. This makes it possible to eliminate throttling requirements. To achieve power savings, one must plot the system head curve, calculate the speed required at various capacities, and determine the required motor horsepower over this range (including losses incurred in the variable speed drive). Once the differences in constant speed energy consumption, variable speed consumption, and the number of operating hours at each capacity are determined, a prediction of the potential yearly savings can be estimated.

Fig. 3.60 shows how the concept of the variable speed driver is applied to a pump. Assume the pump is operating at 100% of its capacity. The TDH is represented by Point 1 on the graph. If it becomes desirable to reduce the capacity to 80% of the rated capacity, the constant speed pump operation will move to Point 3. Point 3 requires

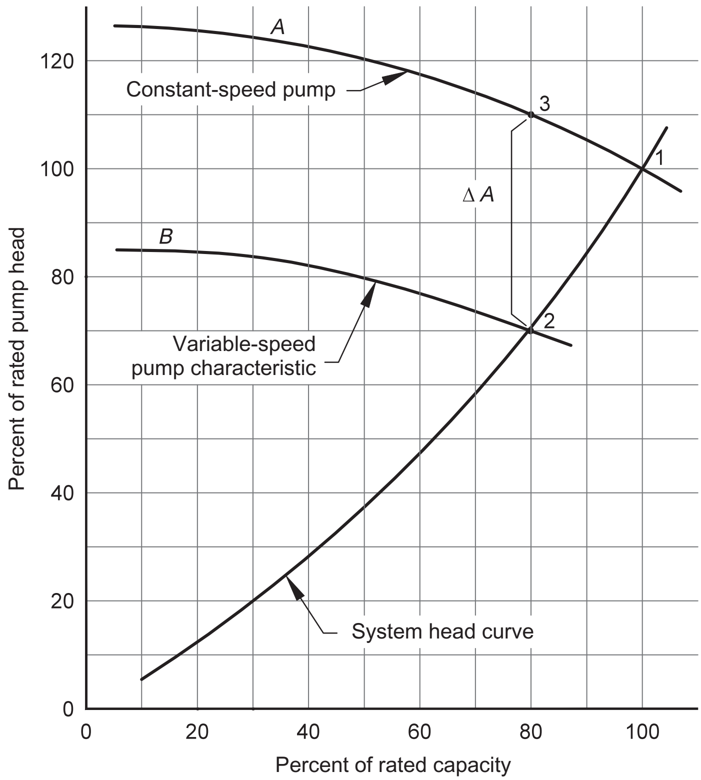


Fig. 3.60 Comparison of constant and variable speed drives.

110% of the head and 92% of the BMP required at Point 1. Using the affinity laws, a variable speed driver could, in effect, find a TDH capacity curve that intersects the system curve at Point 2. Point 2 requires only 70% of the head and 73% of the power required at Point 1. At 80% capacity, the constant speed pump would operate at Point 3 and the variable speed pump at Point 2. The differences in head (100% vs 70%) would have been throttled in a constant speed pump. The potential energy savings represent the difference between 92% and 73% of horsepower, or 19%. Another advantage of variable speed drivers is that throttling and minimum flow valves are not required. The disadvantages are higher cost, lower reliability, and require higher and more specialized maintenance.

3.11 Pump construction details

This section discusses the construction details of the principal parts of centrifugal pumps. The principal parts of centrifugal pumps include:

- Casings and diffusers
- Impellers and wear rings

- Shafts and shaft sleeves
- Shaft sealing
- Balance drums and bearings
- Lubrication system
- Couplings

3.11.1 Casing and diffusers

Centrifugal pumps consist of one or more rotating vanes enclosed within a housing or casing and used to impart energy to a fluid through centrifugal force. This velocity is converted to pressure energy by means of a volute (Fig. 3.61) or by a set of stationary diffusion vanes surrounding the rotating vane(s) (Fig. 3.62). Pumps with volute casings are normally called volute pumps, while those with diffusion vanes are called diffuser pumps or turbine pumps.

3.11.1.1 Casing types

Single-stage centrifugal pumps are usually "volute" type. Multistage pumps are either diffusion-vane or volute. The diffusion-vane or diffuser type incorporates in a cylindrical case a stationary ring of vanes around the periphery of each impeller. Diffusion-vane pumps are widely applied in high-head services. However, volute construction is preferred.

The vane angle for either volute or diffuser, if properly designed, is correct for only the capacity at the BEP. If the pump is operated at some other capacity, the diffuser may act as a hindrance rather than as an aid to efficient operation.

3.11.1.1.1 Single volute

Fig. 3.61 illustrates a typical single-stage end-suction volute pump. End-suction pumps with single-suction impellers have both first-cost and maintenance advantages and not obtainable with double-suction impellers. Because they do not require the extension of a shaft into the impeller suction eye, single-suction impellers are preferred for pumps handling suspended matter. Single-suction impellers are normally used in multistage pumps. Single volute is the most common, because it is simple and efficient. Casing is shaped so that the velocity of the fluid being discharged from the impeller is gradually reduced from near 200 to about 15 fps and converted to pressure as the liquid moves toward the discharge port. The pressure acting on the impeller in a single-volute pump casing design is nearly uniform when the pump is operated at its BEP. Operation above or below the BEP, the pressures around the impeller are not uniform, causing a radial reaction (or radial thrust) that can substantially increase the pump shaft deflection. This unbalanced radial load is placed on the impeller due to the variations of pressure around the periphery. These radial loads occur at flows greater than or less than BEP flow which can lead to:

- Vibration
- Noisy operation
- Premature pump failure

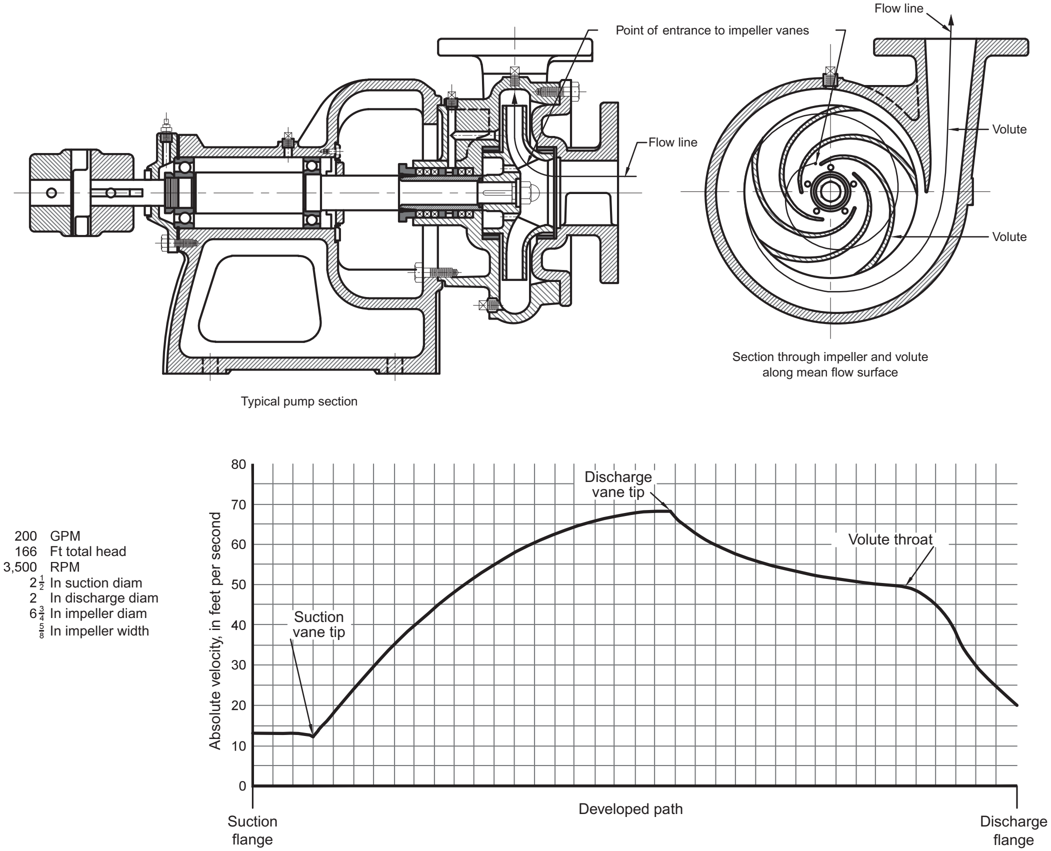


Fig. 3.61 Typical single-stage end-suction volute pump. Courtesy of Worthington Pumps.

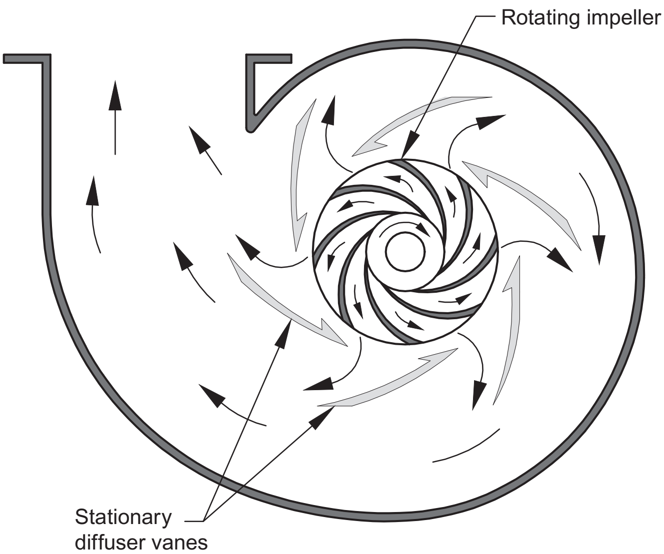


Fig. 3.62 Schematic diagram of a typical diffuser pump.
Courtesy of McGraw Hill.

Radial force is greatest at shutoff and least at maximum efficiency. Radial loads can be reduced by altering the casing type. For example, the radial force can be compensated for by using a stiff shaft or placing a second volute throat on the opposite side of the shaft. This "double-volute" construction is provided on many heavy-duty process type pumps, 3. to 4-inch discharge size or larger.

3.11.1.1.2 Double volute

When it becomes impractical to counteract the radial thrust through the use of a heavier shaft and bearings on a single-volute casing, a double-or-twin volute design can be used. Double-volute pumps consist of two half-capacity volutes, each of which takes the discharge from opposite sides of the impeller (Fig. 3.63). They are superior to the single volute type since the flow of fluid from the impeller produces less shaft deflection and less radial thrust.

3.11.1.1.3 Diffuser

In a diffuser casing the impeller is surrounded by stationary guide vanes that help convert the velocity of the discharge liquid into pressure by changing its flow direction (Fig. 3.62). The diffuser offers peak efficiency at high heads and low flows, thus used in high pressure applications where radial loads are more severe. They are quieter and more efficient at converting liquid velocity to pressure than either volute types, but it is also more costly to purchase and repair. The performance curve is more likely to droop and have severe efficiency drop-off below 50% of BEP flow, thus this casing is only found in large multistage pumps. Pumps with axial flow impellers usually have diffusion vanes following the impeller. Fig. 3.64 summarizes the three types of casings.

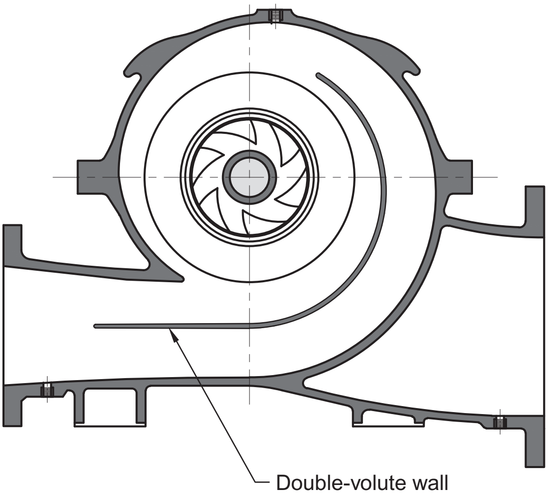


Fig. 3.63 Schematic of a typical double-volute type casing.

It is impractical to use volute casing on pumps where axial flow impellers are used (Fig. 3.65). Instead, the impeller is enclosed in a pipe-like casing. Diffusion vanes are generally used following the impeller. On very low head axial-flow pumps, the diffuser vanes may be omitted. Diffusion vanes are used following the impeller. On very low head axial-flow pumps, the vanes may be omitted.

The efficiency of the volute type is equal to or better than the diffusion-vane type. The diffusion-vane type is more difficult to reassemble after dismantling for maintenance. However, diffusion-vane pumps are preferred because of space considerations. One such use is for pumping deep wells of small diameter. All centrifugal deep-well pumps are "turbine" type pumps with diffusion vanes as an integral part of the case.

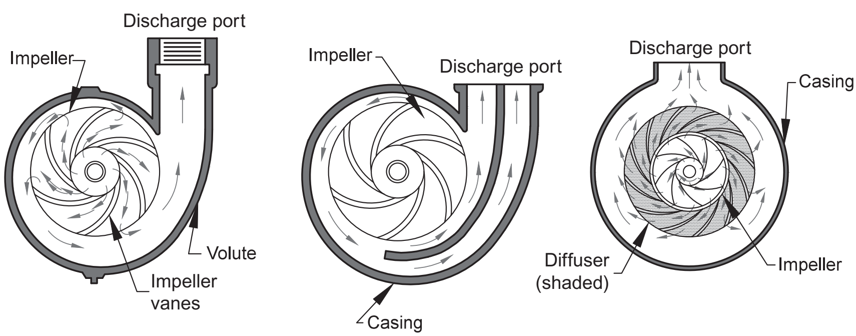


Fig. 3.64 Three types of casings: (Left) single-volute; (middle) double-volute; (right) diffuser.

Fig. 3.65 Typical schematic of a vertical propeller or axial-flow pump.
Courtesy Hydraulic Institute.

3.11.1.2 Casing classification

3.11.1.2.1 Single casing

Single-casing construction consists of a single wall between the fluid and the outside atmosphere. It is the most common design.

3.11.1.2.2 Double casing

Double-casing construction consists of multiple walls between the fluid and the outside atmosphere. They include horizontal pumps, barrel type pumps, and vertical multistage pumps.

Pump casings are classified as either solid or split casings. End-suction single-stage pumps are often made in one-piece solid casings. At least one side of the casing must have an opening with a cover so the impeller can be assembled in the pump. Since the casing is not a true solid casing but split on its vertical axis for access, it is sometimes called a radially split casing. In inexpensive open-impeller pumps, the impeller rotates within close clearance of the pump casing. If the intended service is more severe, a side plate is mounted within the casing to provide a renewable close clearance guide to the liquid flowing through the open impeller.

3.11.1.2.3 Split casing

A split casing is made of two or more parts fastened together. Pump casing is either axially (horizontally) split or radially (vertically) split. Axially split is used principally in high-flow and multistage designs. Pumps with the most common head-capacity ranges are radially split. The term horizontally or "axial" split has been used to describe pumps with casings divided by a horizontal plane through the shaft centerline or axis. In a horizontally split casing the halves of the casings are joined in the same plane (parallel) as the shaft (Fig. 3.66). They are used in pumping services that meet the following criteria:

- Specific gravity of the fluid is less than 0.7
- Temperature of the fluid does not exceed 400°F
- Discharge pressure does not exceed 1.000 psig

They are typically used with large, high capacity pumps. The major advantage is that bearings are used to support the shaft on both sides of the impeller and thus can handle large radial forces.

In multistage pumps with more than one impeller arranged in series, the term barrel pump is sometimes used to describe a radially or vertically split casing. Radially split or barrel pumps tend to be used for higher pressure service, for large temperature changes, or for hazardous liquids because the gasket provides a better seal. Radial or split multistage pumps are more expensive than axially split multistage pumps. In the radial split pumps the casings are joined in a plane that is perpendicular to the plane of the shaft (Fig. 3.67). The axial split pump is known as "back pullout" model since the motor and drive components can be pulled out from the backside of the pump without removing the housing from the piping. They are typically used in services that require a better seal, such as:

- High pressure service
- Large temperature changes
- Hazardous (toxic) liquids

Multistage pump with more than one impeller arranged in series in a radially split casing is called a "barrel pump."

Multistage pumps have adjoining chambers subjected to differential pressure. Means must be made available to isolate these chambers from one another so that leakage from high to low pressure will be kept at a minimum and will take place only at the clearance joints formed between the stationary and rotating elements of the pump. The isolation wall used to separate two adjacent chambers of a multistage pump is called a stage piece, diaphragm, or interstage diaphragm. The stage piece may be formed of a single piece, or it may be fitted with a renewable stage piece bushing at the clearance joint between the stationary stage piece and the part of the rotor immediately inside the former.

Horizontal shaft pumps may be further classified according to the location of the suction nozzle as follows:

- End suction (Fig. 3.61)
- Side suction (Fig. 3.66)
- Bottom suction (Fig. 3.68)
- Top suction

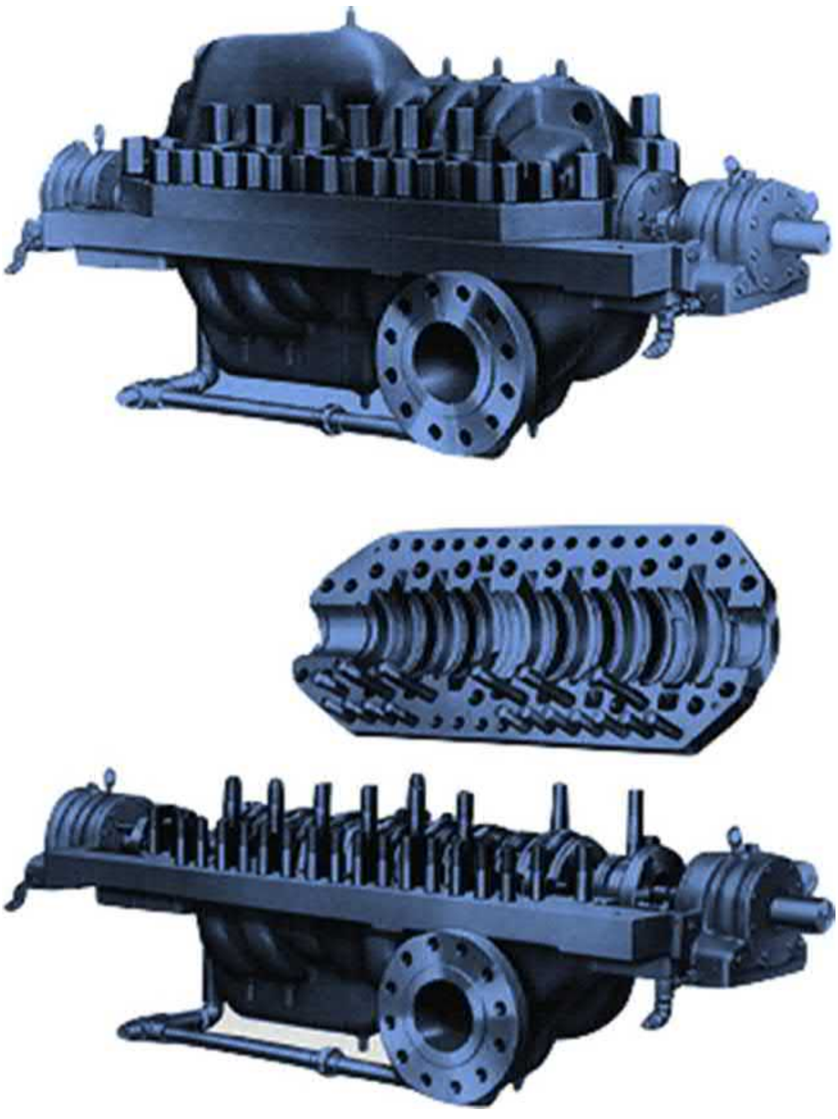


Fig. 3.66 Example of a multistage, horizontally split pump.
Courtesy of Gould Pumps.

3.11.1.3 Pump supports

It is important that pumps and their drivers be removable from their mountings. They are usually bolted and doweled to machined surfaces which in turn are firmly connected to the foundations. These machined surfaces are usually part of a common bedplate on which either the pump or the pump and its driver has been prealigned.

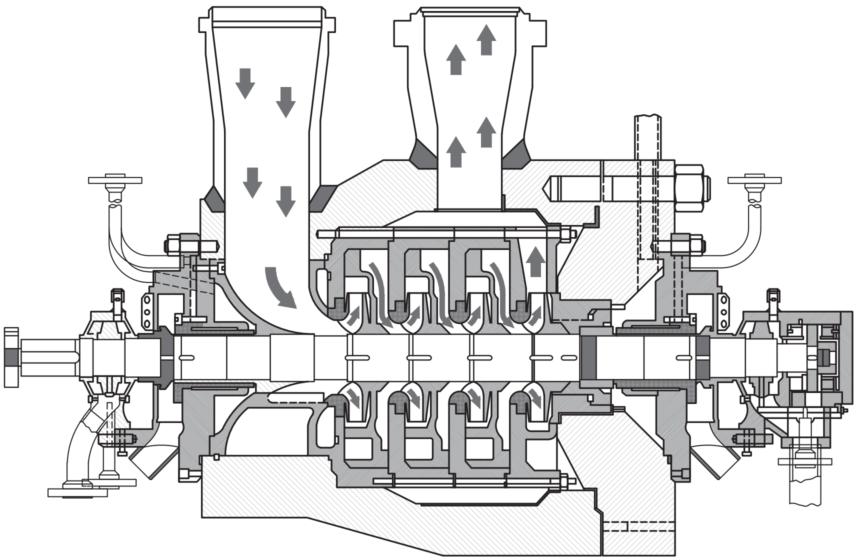


Fig. 3.67 Schematic diagram of a double-casing multistage pump with a radially split inner casing.
Courtesy Worthington Pumps.

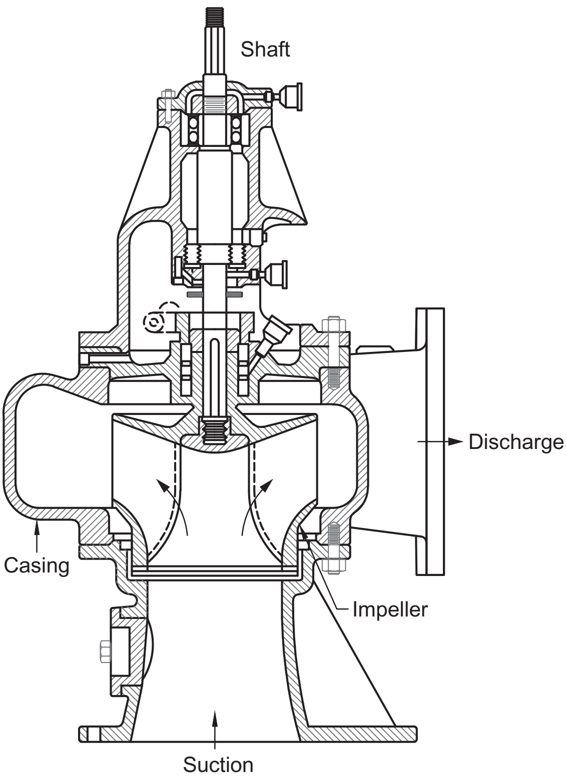


Fig. 3.68 Schematic diagram of a bottom suction pump with a mixed flow impeller.

3.11.1.3.1 Baseplates

The function of a pump baseplate is to furnish mounting surfaces for the pump feet that are capable of being rigidly attached to the foundation (Fig. 3.69). Although baseplates are designed to be quite rigid, they deflect if improperly supported. Therefore it is necessary to support them on foundations that can supply the required rigidity. Since the base can spring out of shape by improper handling it is imperative that the alignment be carefully rechecked during erection and prior to starting the unit.

As the unit size increases so does the size, weight, and cost of the base required. The cost of a prealigned base for most large units could exceed the cost of the field work necessary to align individual bedplates and to mount the component parts, Therefore these bases are therefore used if their function as a drip collector justifies the additional cost. A more rigid installation can be obtained by using individual bases or soleplates and building up the foundation to various heights under the separate portions of the equipment.

The rigidity of the baseplate used to mount the pump and its drive chain will have a direct effect on the life of the pump and its mean time between failures (MTBF) or repair. The base must support the mounted equipment and dynamic loads transmitted by the pump and drive system. The least costly base is nothing more than a piece of channel steel or rolled steel plate. These bases must have several features:

- The mounting surfaces are flat and parallel.
- There is provision to properly level the base prior to grouting.
- There is a way to grout the base.
- There is provision to collect leakage.

Fig. 3.69 Centrifugal pump on a structural steel bedplate.

An example of an excellent baseplate is one fabricated of heavy sections of structural steel plate. It would incorporate the following features:

- *Continuous welds*: This eliminates any chance of moisture collecting in the void pockets of skip welds with subsequent rusting.
- Each section of this type base will have 5 or 6 in. diameter grout holes in each section, ½-in. diameter vent holes in each corner or each section on the base. The grout holes should also have a 1-in. raised lip. If any grout hole is located under a drip source, it should be provided with a tight-fitting cover.
- The *mounting surfaces* for the equipment should be machined flat and parallel with 0.002 in. overall run end to end and side to side in a free and unrestrained condition. Some manufacturers may object to this tight tolerance—it can be done by any good pump company.
- *A 2-in. diameter drain connection*: This will be able to pass any debris that might accumulate on the base.
- A sloped drain pan with about a 1:12 pitch.
- *Rounded corners*: Rounding the corners will eliminate stress concentrations in the grout and foundation. The corner radius should be 2 in.
- *Horizontal positioning screws*: Two at each driver foot location to simplify alignment. No positioning screws are needed at the pump feet, as the driver is aligned to the pump.
- *Leveling screws*: One at each foundation bolt hole. This will simplify leveling of the baseplate prior to grouting.
- *Free surface area at equipment mounting pads*: Sufficient free surface area should be provided at each pad, both axially and transversely, to place a machinist's level during the baseplate leveling procedure.
- *Undersurface preparation*: The underside of the baseplate should be grit blasted to structural steel paint council (SSPC) level 10-near white- with a 2–3 mil profile and coated with unfilled 2-part epoxy paint if epoxy grout is used to grout the baseplate to the foundation. If metal base grout is used, then the underside of the baseplate is to be primed and coated with machinery enamel.

In horizontal single-stage centrifugal pumps the bracket- and centerline-mounted case arrangements are available.

3.11.1.3.2 Foot mounted

Foot-mounted pumps (Fig. 3.4) are used for horizontal pumps that have their casings supported by the baseplate because they are less expensive, suitable for ambient and moderate temperature (<250°F), and light duty service. Foot-mounted support simplifies maintenance since the pump internals can be serviced without disturbing the nozzle flanges.

3.11.1.3.3 Centerline mounting

Centerline-mounted (Fig. 3.70) pumps are used for horizontal pumps single-stage, centrifugal pumps because they provide superior support for heavy-duty service and high temperatures (>250°F). Advantages include:

- Piping stresses are transmitted more directly to the foundations and are less likely to cause misalignment and distortion of the pump.
- Centerline-mounted pumps in accordance with API 610 generally have heavier construction with greater case thickness, heavier shafts, heavier bolting, and high design pressures.

Fig. 3.70 Centerline-mounted centrifugal pump.
Courtesy of Flowserve Corporation.

- Piping and driver can be left in place while the complete rotating element, including the bearing housing and stuffing box, is removed for repairs. This is called the "back pullout" feature.
- Some pumps are designed with larger impeller eye areas which need less $NPSH_R$ than bracket-mounted pumps for the same operating conditions.
- Movement of the centerline is minimized as the temperature rises and coupling alignment is preserved.

3.11.1.3.4 Vertical in-line

Single-stage single-suction centrifugal pumps are made in a vertical in-line design as shown in Fig. 3.71. The in-line pump is used in a variety of services in production facilities. The pump casing is flanged directly to the body and a vertical motor is supported by the pump. The in-line pump offers the following advantages:

- Lower initial cost since there are fewer parts, no fabricated baseplate, no pump bearing housing on some designs (bearings are in the motor), and no flexible couplings or coupling guards.
- Lower installation costs since the foundation is smaller or not needed at all and the piping is simplified.
- Lower maintenance cost because the pump has fewer parts and is permanently aligned with its driver.
- Has a small footprint.

In-line pumps are applicable for temperatures up to 250°F, flows up to 3000 gpm, and heads up to 600 ft. They usually have mechanical seals to seal the shaft but can be furnished with packing. The three basic types of shaft coupling designs used for in-line pumps are as follows:

- Flexible spacer coupling—allows changing the mechanical seal without removing the motor.
- Integral or close-coupled-built-in alignment and a short stiff shaft but the motor and impeller assembly must be lifted and removed to change the seal.

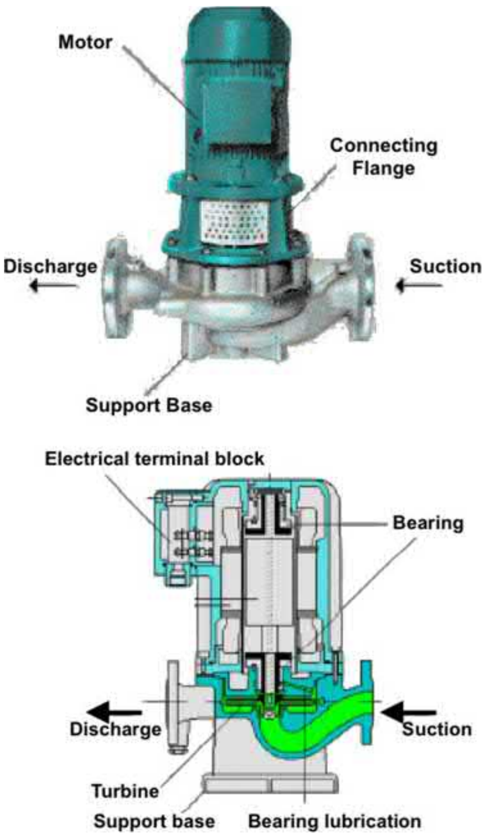


Fig. 3.71 Vertical in-line pump: (Top) external view; (bottom) cross-section view).

- Axially split rigid coupling—least reliable due to the inability of getting and maintaining proper alignment between the pump and motor shaft. Flexible coupling allows changing the mechanical seal without removing the motor. Alignment of motor and pump shafts are easier to maintain

3.11.2 Impellers and wearing rings

3.11.2.1 Background

Centrifugal pumps are designed with respect to:

- Number of suctions (single or double)
- Number of impellers (single, double, or multistage)
- Output
  - Large volume and low TDH
  - Medium volume and medium TDH
  - Small volume and high TDH
- *Impellers* (type and number of vanes, etc.)

The impeller is a casting of specially designed vanes. It is mounted on the shaft and transforms the rotary motion of the shaft into kinetic energy of the fluid being pumped.

The fluid enters from the suction side of the pump, through the eye, along the vanes, and out of the impeller outside diameter (OD) to the volute. Most impellers are arranged from one side only and are called single-suction design (refer to Fig. 3.72). High flow models use impellers that accept suction from both sides and are called double-suction design (refer to Fig. 3.73). Impellers are made of materials such as cast iron, cast bronze, or stainless alloys. Special plastic materials and exotic metals such as titanium can also be used.

3.11.2.2 Impeller mechanical types

The efficiency of a centrifugal pump is determined by the impeller. Vanes are designed to meet a given range of flow conditions. Fig. 3.74 shows the three basic types of impellers. Fig. 3.75 shows impeller nomenclature.

3.11.2.2.1 Open impellers

Strictly speaking, an open impeller (Fig. 3.76) consists of nothing but vanes. Vanes are attached to the central hub, without any form or sidewall or shroud. The disadvantage of this impeller is structural weakness. If the vanes are long, they are strengthened by ribs or a partial shroud. Open impellers are usually used in small, inexpensive pumps or pumps handling abrasive liquids or suspended solids (in which the impeller rotates between two side plates, between the casing walls of the volute, or between the

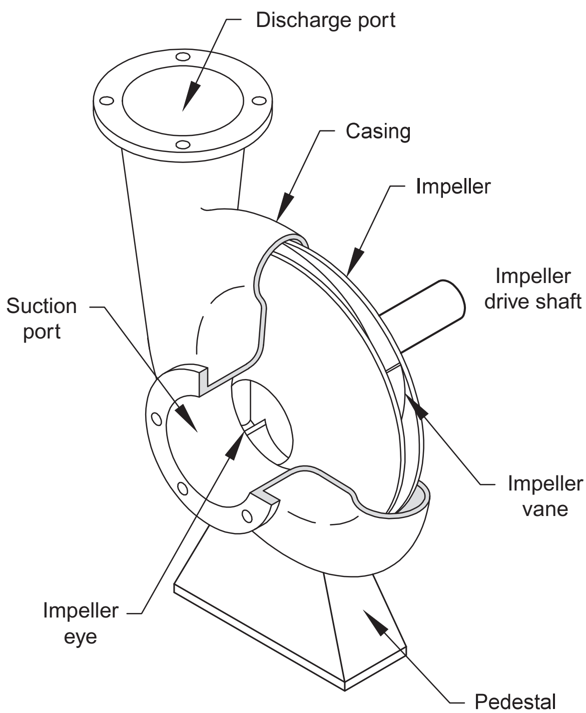
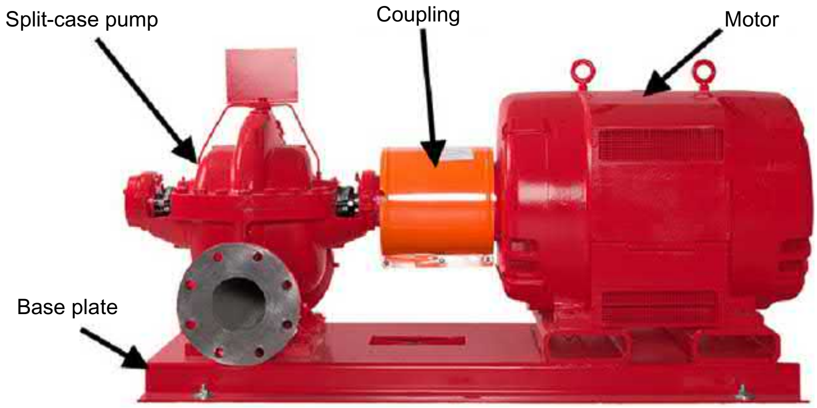


Fig. 3.72 Single-suction centrifugal pump.



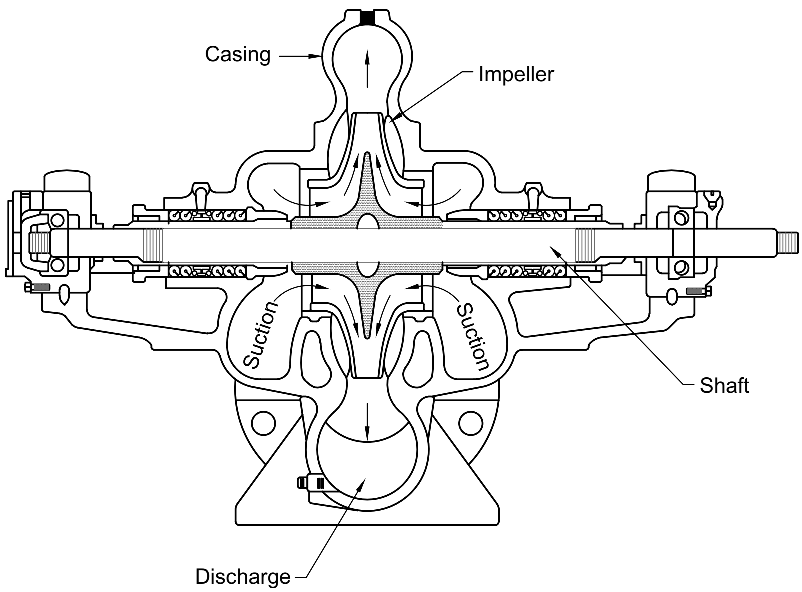


Fig. 3.73 Double-suction centrifugal pump. (Top) External view; (bottom) cutaway of double-suction.

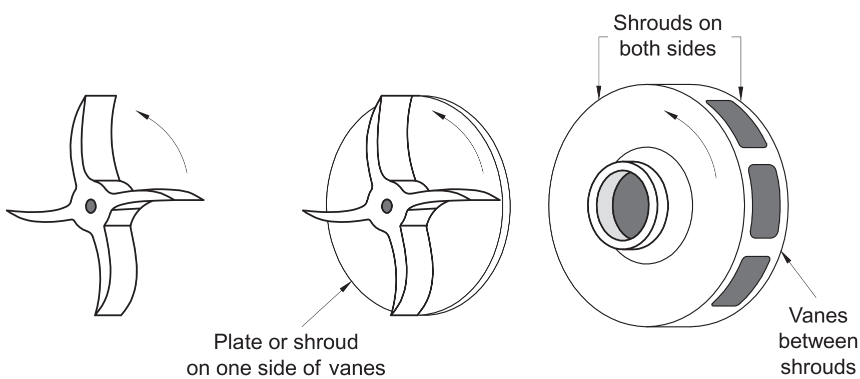


Fig. 3.74 Basic types of centrifugal pump impellers: open, semiopen, and closed.

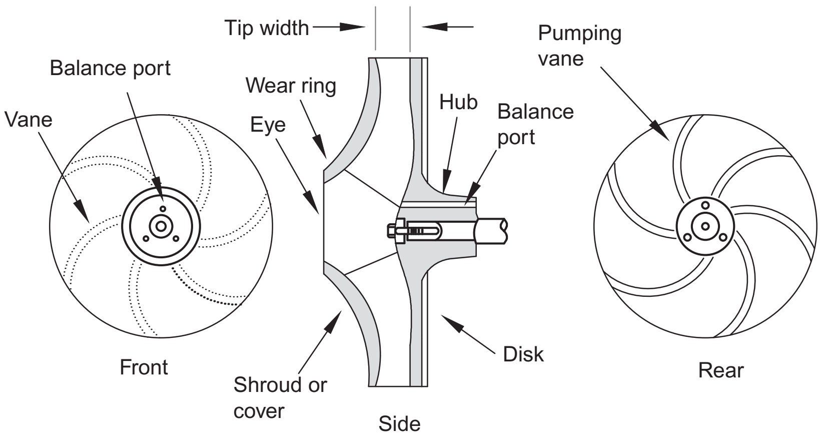


Fig. 3.75 Impeller nomenclature.

Fig. 3.76 Open impeller with partial shroud.

stuffing box head and the suction head). The clearance between the impeller vanes and the sidewalls allows for some water slippage (similar to slip in a reciprocating pump). This slippage increases as wear increases. To restore the original efficiency, both the impeller and side plate must be replaced. This involves a much larger expense than would be experienced if a closed impeller was used where simple rings form the leakage joint.

3.11.2.2.2 Semiopen (partially open)

The semiopen impeller (Fig. 3.77) incorporates a back wall (shroud) that serves to stiffen the vanes and adds mechanical strength. Pump-out vanes may or may not be included and these are located on the back of the impeller shroud (Fig. 3.78). Their function is to reduce the pressure at the back hub of the impeller and to prevent pumped foreign matter from becoming lodged behind the impeller and interfering with the proper operation of the pump and stuffing box. The semiopen impeller is used in medium diameter pumps and with liquids containing small amounts of suspended

(A)

(B)

Fig. 3.77 (A) Semiopen impeller. (B) Cutaway of a centrifugal pump showing shaft, shaft sleeve, and impeller.

Fig. 3.78 Open impeller with partial shroud.

Fig. 3.79 Example of straight-vane single-suction closed impeller.
Courtesy of Worthington Pumps.

solids. They offer higher efficiencies and lower $NPSH_R$ than the open impeller. They can be adjusted for wear by axial movement of the shaft. It is important that a small clearance or gap exists between the impeller vanes and the housing. If the clearance is too large, slippage and recirculation will occur which in turn results in reduced efficiency and positive heat build-up.

3.11.2.2.3 Closed impellers

The closed impeller (Figs. 3.79 and 3.80), which is nearly always used for pumps handling clear liquids, incorporates shrouds or enclosing sidewalls that totally enclose the impeller passageways from the suction eye to the periphery. Although this design prevents the liquid slippage that occurs between an open or semiopen impeller and its side plates, a running joint is necessary between the impeller and casing to separate the discharge and suction chambers of the pump. This running joint is usually formed by a relatively short cylindrical surface on the impeller shroud that rotates within a slightly larger stationary cylindrical surface. If one or both surfaces are made renewable, the leakage joint can be repaired when wear causes excessive leakage.

Fig. 3.80 Double-suction closed impeller.

Closed impellers are used in large pumps where high efficiencies and low $NPSH_R$ are critical. They can operate in suspended solid service without clogging but will exhibit high wear rates. They are the most widely used type of impeller for centrifugal pumps handling clear liquids. They rely upon close clearance wear rings located on the impeller and on the pump housing which separates the inlet pressure from the pressure within the pump, reduces axial loads and maintains efficiency.

If the pump shaft terminates at the impeller so that the impeller is supported by bearings on only one side, the impeller is called an *overhung impeller*. This type of construction is the best one for end-suction pumps with single-suction impellers.

3.11.2.3 Impeller nomenclature

The inlet of the impeller just ahead of the vanes is called the "suction eye" (Fig. 3.75). In a closed-impeller pump, the suction eye diameter is taken as the smallest inside diameter of the shroud. In determining the area of the suction eye, the area occupied by the impeller shaft hub is deducted.

By definition, a hub is the central part, usually cylindrical, of a wheel. The term applies only to that central part of an impeller which receives the pump shaft.

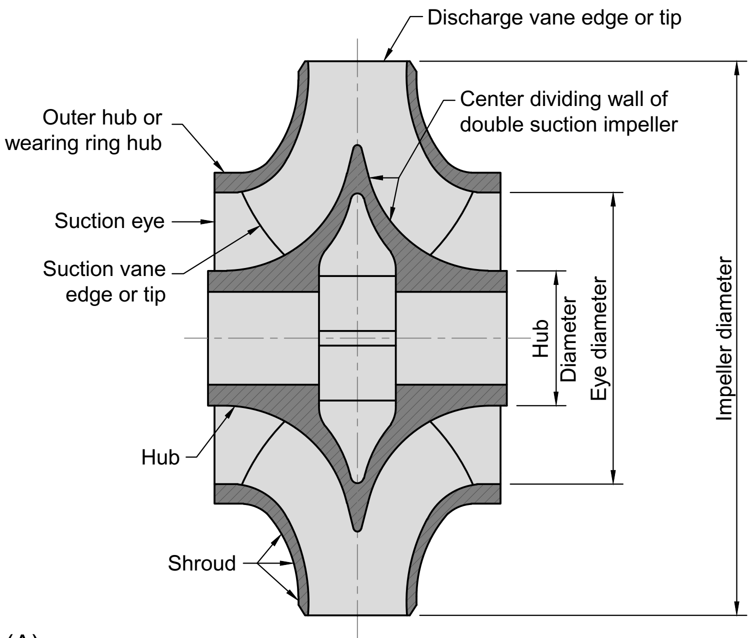
3.11.2.4 Impeller flow types

The impeller is the heart of the centrifugal pump. It rotates the liquid mass with the peripheral speed of its vane tips, thereby determining the head produced or the pump working pressure. Impellers are classified as (1) single-suction impellers (Fig. 3.75) or (2) double-suction (Fig. 3.81) impellers.

In a single-suction impeller, the liquid enters the suction eye on one side only. As a double-suction impeller is, in effect, two single-suction impellers arranged back to back in a single casting, the liquid enters the impeller simultaneously from both sides. In double-suction impellers, the two casing suction passageways are normally connected to a common suction passage and a single suction nozzle. Double-suction impellers are mainly used in a horizontal split-case pump. It provides the option of using a single pump to handle a higher flow for large capacity performance. Some pumps will require motors of several hundred horsepower. There are some designs where the double-suction impeller is mounted on the end of the shaft in a special volute and mated to piping for easy installation.

For small units, the single-suction impeller is more practical to manufacture than the double suction because the passageways are not divided into two very narrow passages. In addition, single-suction impellers are sometimes preferred for structural reasons. End-suction pumps with single-suction overhung impellers have both initial cost and maintenance advantages not obtainable with a double-suction impeller. Because an overhung impeller does not require shaft extension into the impeller eye, it is preferred for pumps handling suspended matter.

For general service, single-stage axially split casing design, a double-suction impeller is favored for the following two reasons:



(A)

(B)

Fig. 3.81 (A) Example diagram showing the parts of a double-suction impeller. Courtesy of McGraw Hill. (B) Horizontal split-case pump with a double-suction impeller.

(1) It is theoretically in axial hydraulic balance, thereby eliminating the need for an oversized thrust bearing under constant full load.
(2) The greater suction area in a double-suction impeller compared to a single-suction design permits the pump to operate with less NPSH for a given capacity.

In multistage pumps, single-suction impellers are almost universally used because of the design, first cost, and maintenance complexity that double-suction staging introduces.

3.11.2.5 Number of impellers

3.11.2.5.1 Single-stage pumps

The single-stage, single-suction pump is widely accepted, highly reliable, and by far the most widely used pump in production operations. It consists of one impeller. It is used in pumping services of low to moderate TDHs. The TDH is a function of impeller's speed, normally not higher than 700 ft/min. The impeller can be either single suction or double suction. However, single-suction impellers have higher unbalanced thrust and radial forces at off-design flows than double suction and multistage.

3.11.2.5.2 Multistage pumps

The multistage pump has two or more impellers. They are used in pumping services of moderate to high TDHs. Each stage is essentially a separate pump. Each stage is located within the same housing and installed on the same shaft. Eight or more stages can be installed on a single horizontal shaft. There is no limit of stages installed on a vertical shaft. The pumped fluid flows from the first stage to successive stages. Each stage increases the head by about the same amount. The first impeller may be either single suction or double suction.

3.11.2.6 Impeller sizing

The diameter of an impeller can be estimated by the following equation:

$$D = \frac{1839\sqrt{TDH'}}{N} \tag{3.22}$$

Where:

D = impeller diameter (in.)
TDH' = head per stage at best efficiency (ft)
N = pump rotative speed (rpm)

For a constant head, the impeller diameter can be reduced by selecting a higher speed. For a constant speed, the impeller diameter can be reduced only by lowering the required head.

3.11.2.7 Wear rings

3.11.2.7.1 Background

Wear rings provide an easy and economical renewable leakage joint between the impeller and casing. Wear rings should be included on all centrifugal pumps, especially those in hydrocarbon or waterflood service. They are not normally required on small horsepower. Sandy service requires a flushing fluid to be injected.

Wear rings are usually placed in the volute. Some pumps will have rotating wear rings that are mounted on the hub of the impeller so that any wear occurring will take place on less costly replacement components rather than the more costly volute or impeller. Some applications will have wear rings at both the impeller eye and volute.

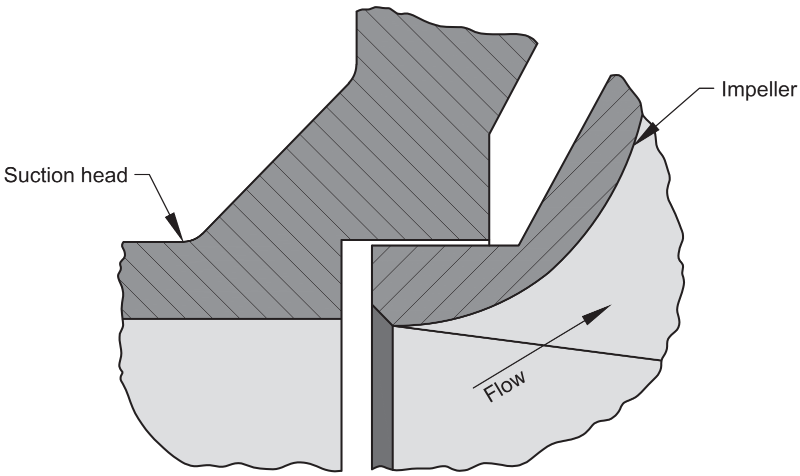


Fig. 3.82 Schematic diagram of a plain flat leakage joint (no rings).
Courtesy of McGraw Hill.

A leakage joint without renewable parts is used only in very small, inexpensive pumps. (Fig. 3.82). Wear rings are usually in pairs, one stationary, one rotating. The rotating ring is attached to the impeller, the stationary ring is concentric with the impeller wear ring but seated in the casing. The stationary ring is called a casing ring if mounted in the casing, a suction-cover ring or suction-head ring if mounted in a suction cover or head, or a stuffing-box cover ring if mounted in the stuffing-box cover. The primary purpose of these rings is to minimize internal leakage from the discharge back to suction. In well-designed pumps of moderate size, this leakage is about 5% of the total liquid passed through the impeller. The less the wear ring clearance, the less the internal leakage and the higher the pump efficiency. However, wear ring and pump seizure can result from too close clearances. Wear ring trouble may be due to any of the following listed causes. Extra wear ring clearance may prevent these problems:

- Distortion of pump case from pipe stresses or from improper warm-up
- Deflection of the shaft, causing contact between the wear rings
- Unbalance in the rotating element
- Lodging of hard foreign bodies between wear rings
- Thermal transients which cause loosening of the fit and eventual wear ring movement
- Galling due to improper wear ring material combination
- Eccentric fit due to improper machining and/or assembly

Shaft deflection is usually due to unbalance of the rotating element caused by hydraulic side thrust in the volute, unbalanced impellers, or both.

Wear rings may be provided in bronze, cast iron, stainless steel, and other alloys. Wear rings can be faced with hard materials such as ceramics, Colmonoy, tungsten carbide or silicon carbide, or in the case of 400 series stainless steels (steels with a chrome content in excess of 12%) can be hardened. In any case, the surface hardness

Wearing ring diameter (in)	Diametral clearance (in)
<2	0.010
2.000–2.499	0.011
2.500–2.999	0.012
3.000–3.499	0.014
3.500–3.999	0.016
4.000–4.999	0.016
5.000–5.999	0.017
6.000–6.999	0.018
7.000–7.999	0.019
8.000–8.999	0.020
9.000–9.999	0.021
10.000–10.999	0.022
11.000–11.999	0.023
Note: For nongalling materials and pumps operating below 350°F	

Fig. 3.83 API 610 minimum wear ring clearance for metallic rings. Courtesy of American Petroleum Institute (API).

of the mating parts must be at least 50 Brinell points different from another to prevent seizing or galling should an upset occur and the parts make rubbing contact.

Fig. 3.83 lists the minimum wear ring clearances recommended by API Standard 610. These clearances are for processes operating at temperatures below 350°F. For pumps operating above 350°F with metallic wear rings, an additional 0.002-in. clearance for each 100°F above 350°F should be added to the values shown in Fig. 3.83.

Use of nonmetallic wear ring materials offers opportunities for improved reliability in services where frequent start-up occurs, dry running occasionally occurs, or radial deflection is high. Many nonmetallic materials are limited to temperatures below 350°F. One should check with the manufacturer for specific wear ring limitations.

To restore original clearances after wear, operations personnel must either:

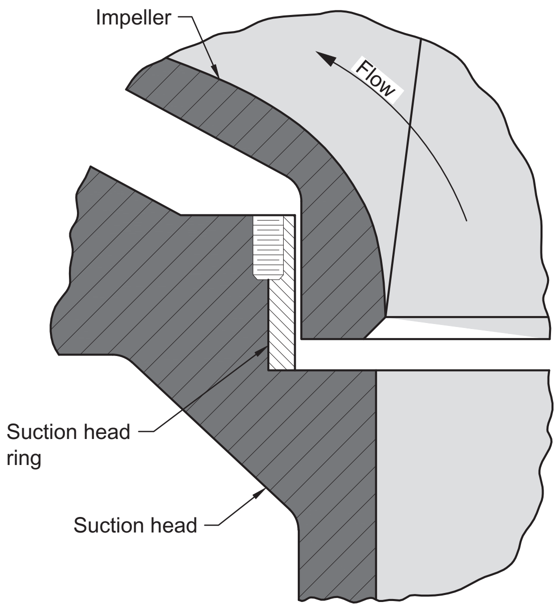
(1) Build up the worn surfaces by welding, metal spraying, or other means, and then true up the part; or
(2) Buy new parts.

The new parts are not very costly in small pumps, especially if the stationary casing element is a simple suction cover. As a matter of fact, the cost of a renewable stationary ring would differ very little from that of a totally new suction cover for these units. This is not the case for larger pumps, nor if the stationary element is part of a complicated casting. If the initial cost of a pump is important, the designer should provide a means for both stationary parts and the impeller to be remachined. Renewable casting and impeller rings can then be installed.

Nomenclature for the casing or stationary part that forms the leakage joint surface varies as follows:

(1) "Casing ring"—If mounted in the casing.
(2) "Suction-cover ring" or "Suction-head ring"—If mounted in a suction cover or head (Fig. 3.84)
(3) "Stuffing box-cover ring" or "head ring"—If mounted in the stuffing box cover or head

Fig. 3.84 Single flat "casing ring" construction.



A renewable part for the impeller wearing surface is called the "impeller ring." Pumps with both stationary and rotating rings are said to have "double-ring" (Fig. 3.85) construction.

3.11.2.7.2 Wear ring types

There are various types of wear ring designs. The selection of the most desirable type depends on the

- Liquid being handled
- Pressure differential across the leakage joint
- Running speed
- Pump design

The most common ring constructions are the flat type (Figs. 3.84 and 3.85) and the "L" type (Fig. 3.86). The leakage joint in the flat type is a straight annular clearance. In the L-type ring (Fig. 3.86), the axial clearance between the impeller and casing ring is large so that the velocity of the liquid flowing into the stream entering the suction eye of the impeller is low. The L-type casing rings shown in Figs. 3.86 and 3.87 have the additional function of guiding the liquid into the impeller eye; they are called "nozzle rings."

Another popular wear ring configuration is the "Labyrinth" type ring (Figs. 3.88 and 3.89), which have two or more annular leakage joints connected by relief chambers. In leakage joints involving a single unbroken path, the flow is a function of the area and length of the joint and the pressure differential across the joint. If the path is broken by relief chambers (Figs. 3.88–3.90), the velocity energy in the

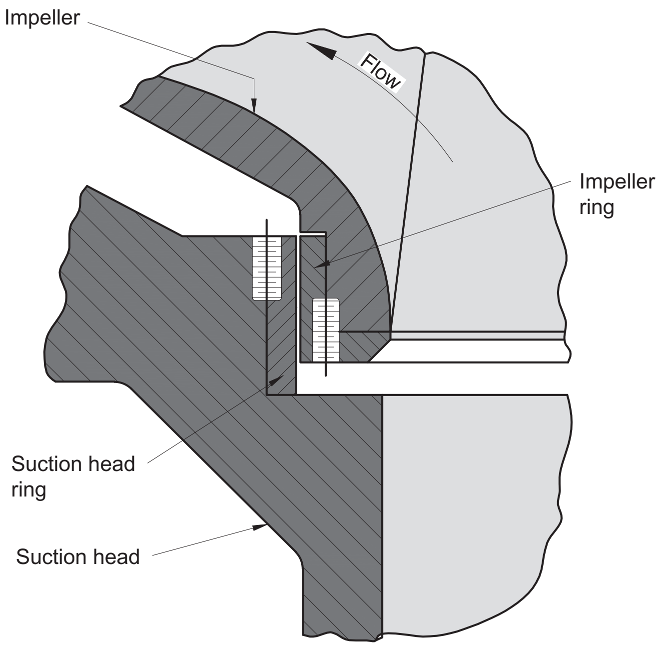


Fig. 3.85 "Double flat ring" construction.

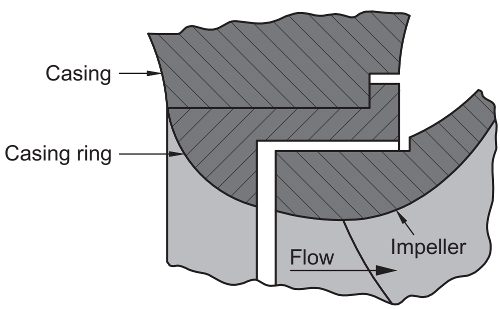


Fig. 3.86 Schematic diagram of an L-type nozzle casing ring. Courtesy of McGraw Hill.

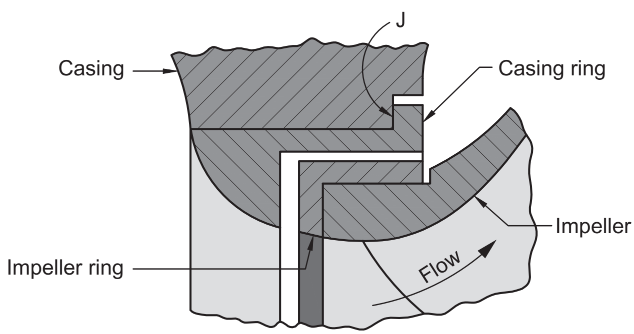


Fig. 3.87 Schematic diagram of an L-type nozzle with double casing ring. Courtesy of McGraw Hill.

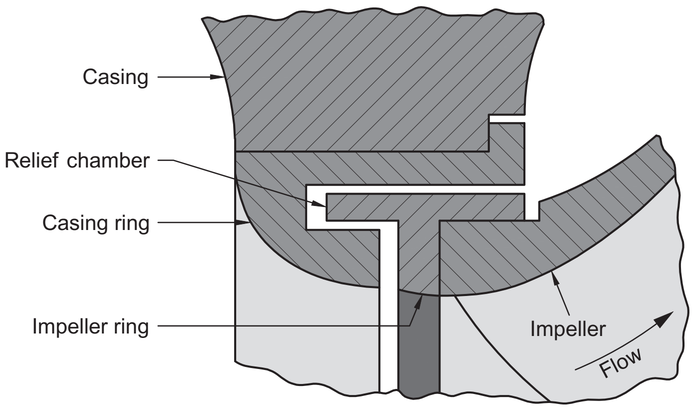


Fig. 3.88 "Single labyrinth" of intermeshing type (double ring construction with nozzle type ring).

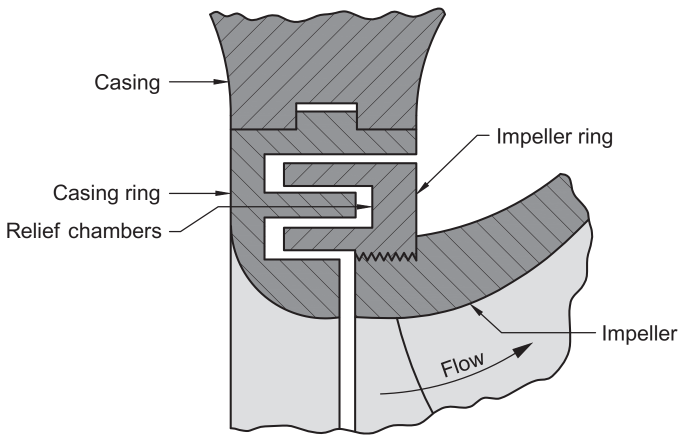


Fig. 3.89 "Double labyrinth" ring construction.

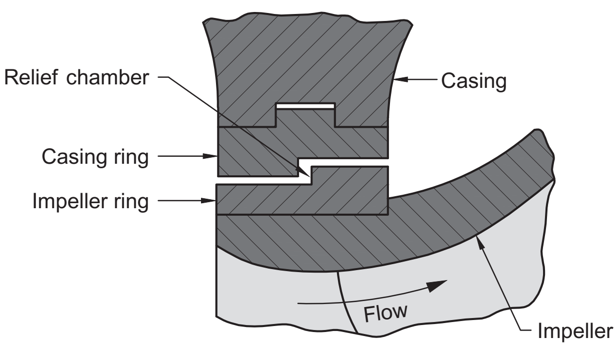


Fig. 3.90 Step type leakage joint (double rings).

jet dissipates in each relief chamber, increasing resistance. As a result, with several relief chambers and several leakage joints for the same actual leakage flow (Fig. 3.89), the area and hence the clearance can be greater than for an unbroken, shorter leakage joint.

The single labyrinth ring with only one relief chamber (Fig. 3.88) is often called an intermeshing ring. The step-ring type (Fig. 3.90) utilizes two flat ring elements of slightly different diameters over the total leakage joint width with a relief chamber between them. Other ring designs also use some form of relief chamber.

3.11.3 Shaft and shaft sleeves

3.11.3.1 Shaft

The shaft is designed to transfer energy from the driver (motor, turbine, etc.) to the pump impeller. The shaft transmits the torques encountered in starting and operating the pump while supporting the impeller and other rotating parts. The shaft is supported by the bearing frame assembly and designed to allow minimum deflection at the impeller. The shaft must be large enough to transmit the necessary energy to the liquid being pumped. It must also have sufficient strength to resist deflection from hydraulic thrust of the liquid in the pump case. Horizontal shafts must also be capable of carrying the weight of the rotating parts. These factors require a shaft diameter consistent with the:

- Strength of the material
- Distance between the bearings
- Distance from the bearings to the impeller

Increasing the shaft diameter may increase the

- entrance velocity
- $NPSH_R$
- Larger bearings
- Larger stuffing-box area
- Other parts

When a pump shaft rotates at any speed corresponding to its natural frequency, minor unbalances will be magnified. These speeds are called the "critical" speeds. A "rigid" shaft is one where the operating speed is lower than its critical speed. Whereas a flexible shaft is one with an operating speed higher than its critical speed. Shafts are designed by the pump manufacturer. Common practice is to use shafts of such rigidity that the critical speed is at least 20% above the operating speed. Experience has proved equally satisfactory results can be obtained with lighter shafts with a critical speed of about 60%–75% of the operating speed. This is sufficient margin to avoid any danger caused by operation close to the critical speed.

Each shaft usually has a sleeve affixed to it at the mechanical seal or packing area.

3.11.3.2 Shaft sleeves

Pump shafts are usually protected from erosion, corrosion, and wear at stuffing boxes; leakage joints; internal bearings; and in waterways by renewable sleeves. The shaft sleeve is a replaceable wear part that is generally less costly than the shaft itself. The shaft sleeve protects the shaft from wear at a stuffing box. Since the packing must be kept in close contact with the shaft and sides of the stuffing box, there is rubbing pressure on the shaft which in time will cause wear. Since the shaft is an expensive part of the pump, it is common practice to fit removable sleeves over the shaft in the packing area. These sleeves are easy to replace when worn. The sleeve can be made from higher grade materials than the shaft. It can also be hardened and/or coated to increase its wear resistance. However, the sleeve does not increase the shaft stiffness.

The shaft sleeve should have the following characteristics:

- Hard, smooth, and corrosion resistant
- Prevent leakage between the shaft and the sleeve
- Sleeve should extend beyond the packing gland so operator can distinguish if leakage is from sleeve or packing

Shaft sleeves' primary function is that of protecting the shaft from wear at the stuffing box. Shaft sleeves serving other functions are giving specific names to indicate their function. For example, in large high head pumps that put a high axial load on the sleeve would have a sleeve configuration as shown in Fig. 3.91. In pumps with overhung impellers, various types of sleeves are used. For example, the sleeve protects the impeller hub from wear in pumps where the stuffing box is close to the impeller (Fig. 3.92).

Lantern rings are used in packed stuffing boxes to distribute the sealing and lubricating liquid around the shaft. They are also used for "leak-off" in conjunction with a throat bushing to reduce the pressure on the packing itself. Both types of lantern rings are shown in Fig. 3.93. As shown in Fig. 3.93, the pump packing gland compresses the packing rings in the stuffing box. Packing glands can be made in two pieces so they can be removed entirely from the shaft to provide adequate clearance for working on the packing.

3.11.4 Shaft sealing

Shaft sealing prevents leakage at the point where the shaft enters the pump casing. Lack of a good shaft seal may result in down time, process, or environmental contamination and jeopardize the safety of personnel. The two basic types of seals are as follows:

- Stuffing box (packing)
- Mechanical seal

3.11.4.1 Stuffing box (packing)

Pump packing, contained in stuffing boxes, may be used in some applications to prevent extreme loss of fluid from the low-pressure side of the pump where the shaft penetrates the volute. If a pump handles suction lift and the pressure at the stuffing box is below

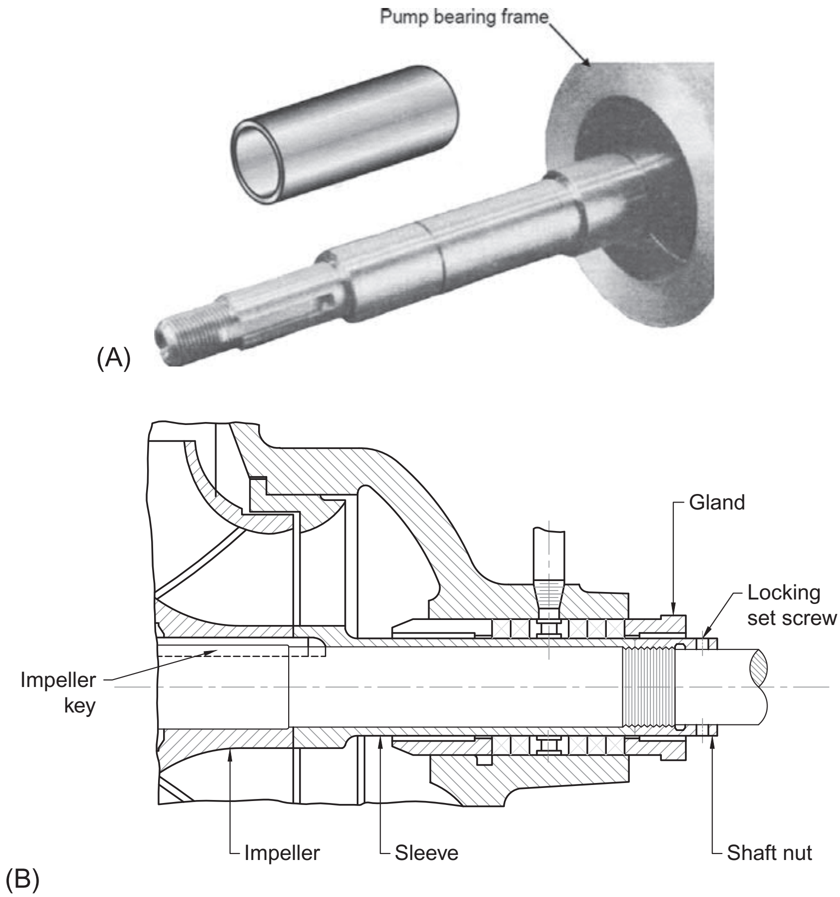


Fig. 3.91 (A) Shaft/shaft sleeve components. (B) Sleeve with external lock nut and impeller key.

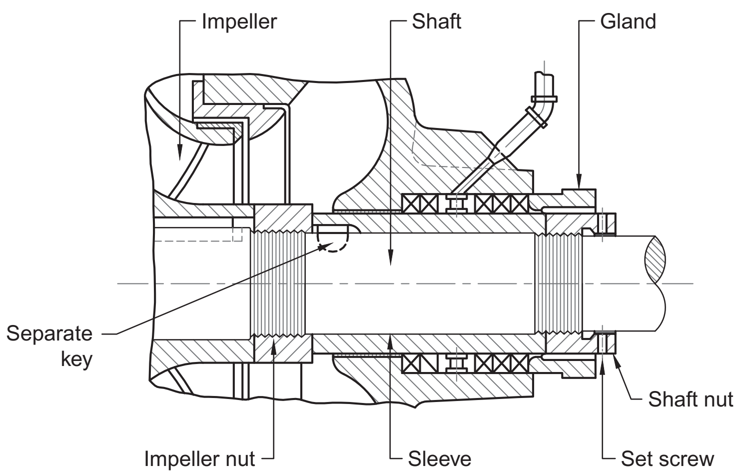


Fig. 3.92 Shaft sleeve for pumps with overhung-impeller hubs extending into the stuffing box.

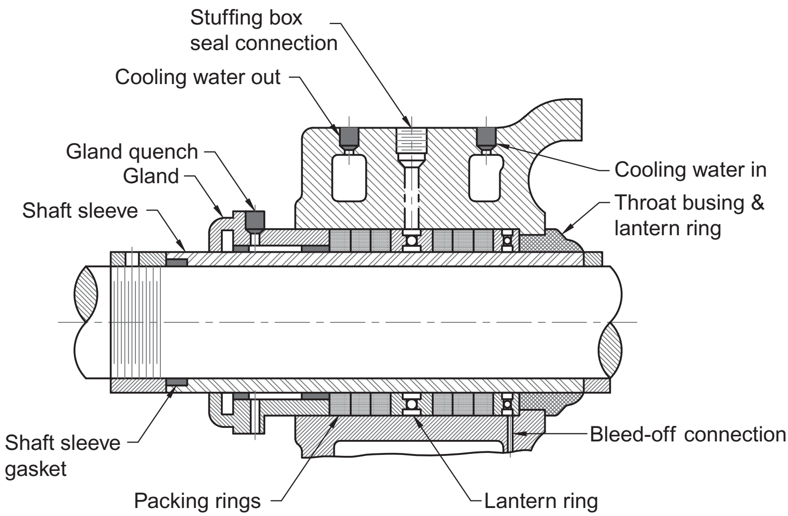


Fig. 3.93 Conventional stuffing box illustrating a "throat" lantern ring on the top and a "leak-off" type lantern ring at the bottom of the box.

atmospheric, the stuffing box function is to prevent air leakage into the pump. If the pressure is above atmospheric, the function is to prevent liquid leakage out of the pump.

Most applications allow for a small amount of leakage between the shaft and packing, in order to lubricate and cool the packing material.

The stuffing box accommodates a number of rings of packing around the shaft or shaft sleeve (Fig. 3.94). If sealing the box is desired, a lantern ring or seal cage (Fig. 3.95) is used that separates the rings of packing into approximately equal sections. Packing is compressed to give the desired fit on the shaft or shaft sleeve by a gland that can be adjusted in an axial direction.

When a pump operates with negative suction head, the inner end of the stuffing box is under vacuum, and air tends to leak into the pump. For this type of service, packing is usually separated into two sections by a lantern ring or seal cage (Fig. 3.96). Water or some other sealing fluid is introduced under pressure into the space, causing flow of sealing fluid in both axial directions. This construction is also used for pumps handling flammable or chemically active and dangerous liquids since it prevents outflow of the pumped liquid.

High temperatures or pressures complicate the problem of maintaining stuffing box packing. Pumps in these services are usually provided with jacketed, water-cooled stuffing boxes. The cooling water removes heat from the liquid leaking through the stuffing box and heat generated by friction in the box.

3.11.4.2 Stuffing box packing

Packing is a pressure breakdown device. Packing must be flexible and capable of being compressed for proper operation. It fills the space between the rotating shaft and the inside of the stuffing box (Figs. 3.97 and 3.98). Packing must also absorb

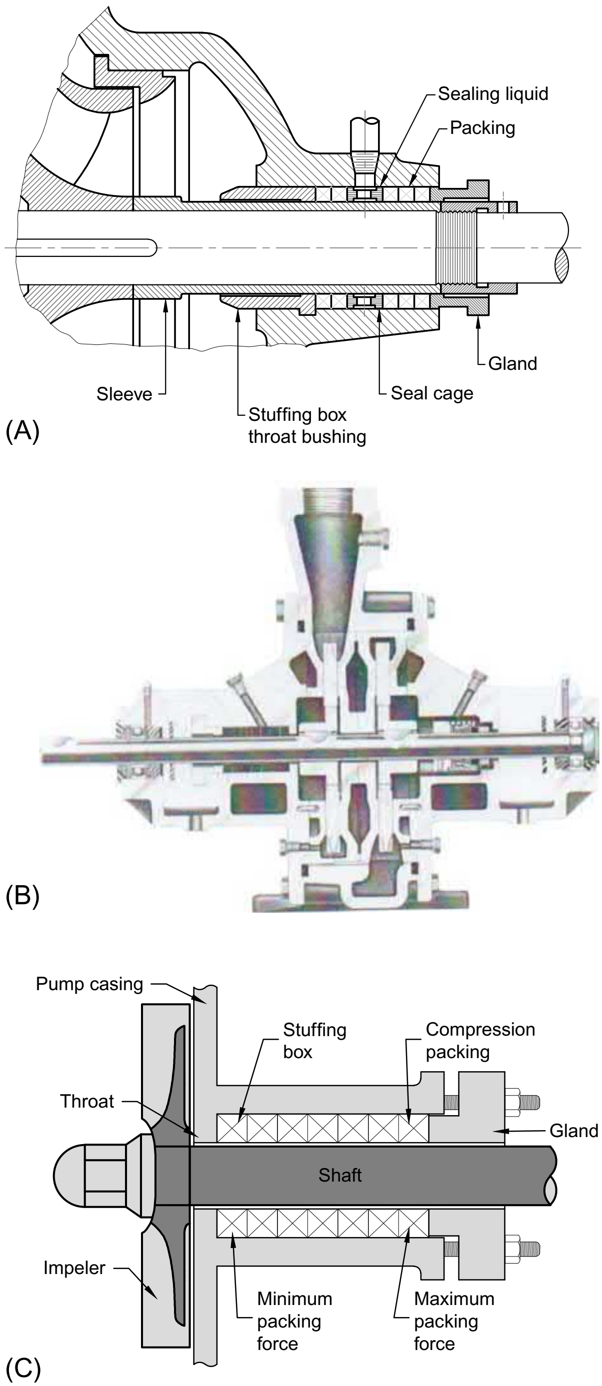


Fig. 3.94 (A) Conventional stuffing box with bottoming ring. (B) Cutaway of a centrifugal pump illustrating the stuffing box and packing. (C) Stuffing box assembly.

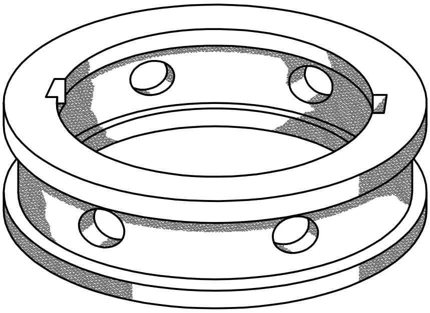


Fig. 3.95 (Top) Lantern ring-Lantern ring; (bottom) lantern ring installed in stuffing ox.

energy without failing or damaging the rotating shaft or shaft sleeve. There are numerous packing materials, each adapted to some particular service. Two of the most commonly used types are as follows:

- *Asbestos*—soft and suitable for hot and cold water applications. It is the most common packing used in general service under normal pressures (under 200 psi).
- *Metallic*—composed of flexible metallic strands or foil with graphite or oil lubricant impregnation and with a plastic core. Packing self-lubricating for its start-up period. Impregnation makes. Babbitt foil is used on water and oil service for low and medium temperatures (up to 450°F) and medium to high pressures.

All packing requires a small amount of leakage which ensures proper lubrication and carries off frictional heat. Besides low cost, packing has the advantage of being easy to replace. However, it has the disadvantages of requiring both a small leakage rate for

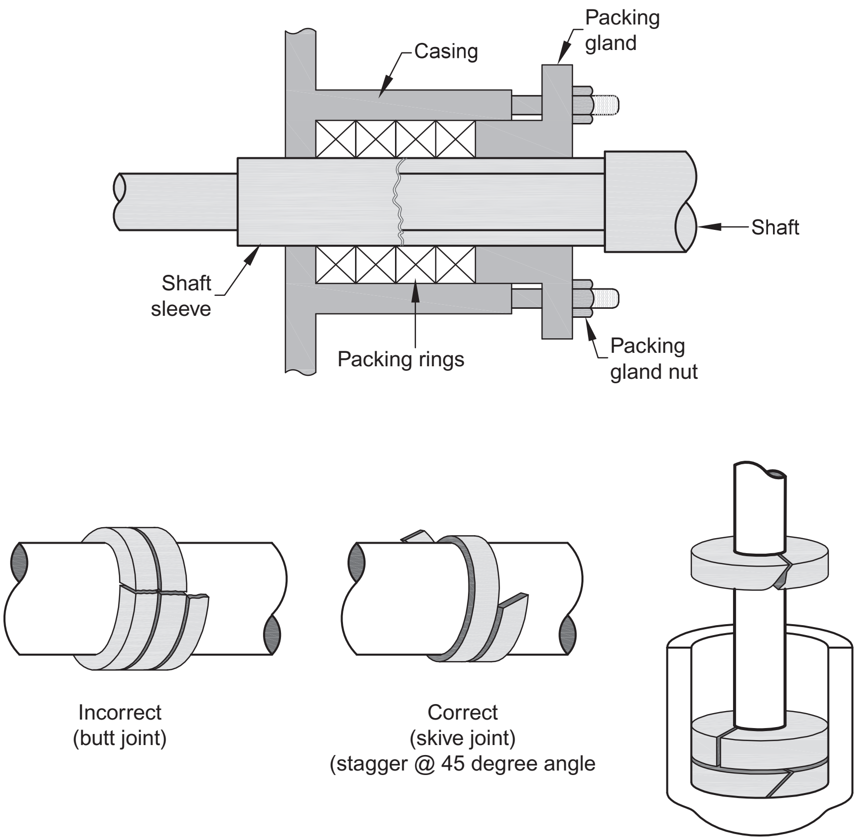


Fig. 3.96 Packing: (top) soft packing controls shaft leakage; (bottom) correct and incorrect packing installation.

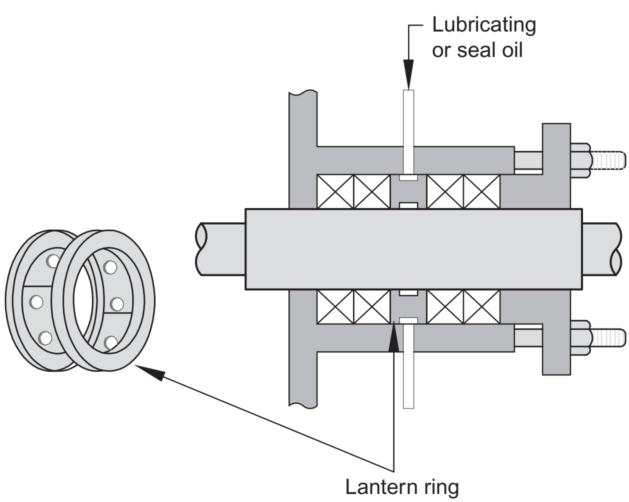


Fig. 3.97 Lantern ring used to allow external lubrication.

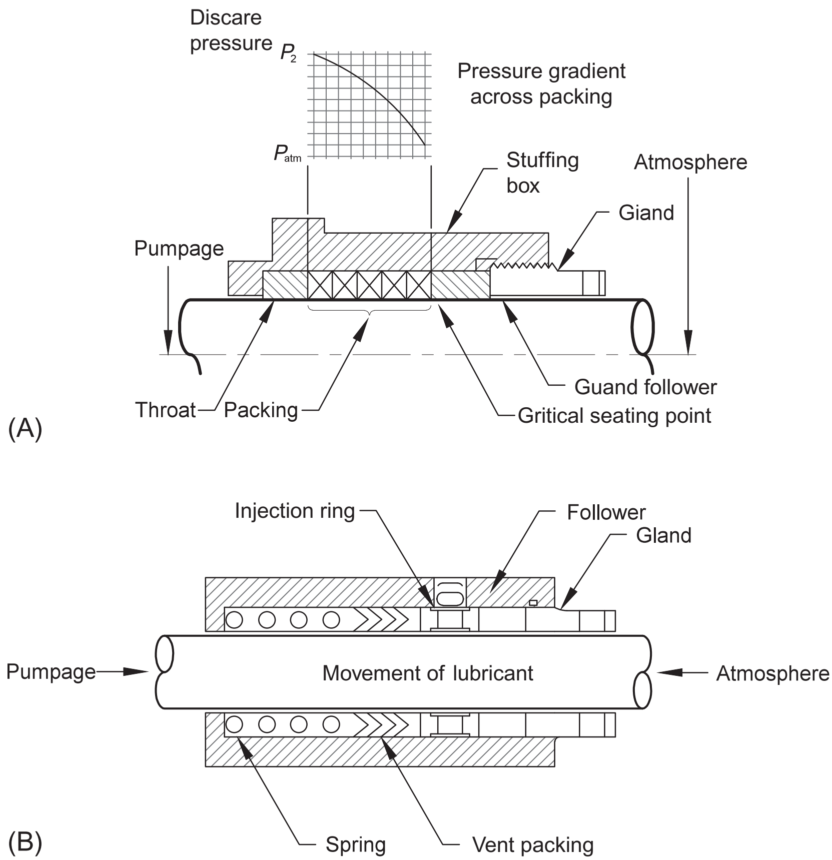


Fig. 3.98 Schematic of general packing arrangement. (A) Square cut packing; (B) Chevron packing.

proper operation and continuous adjustment as it wears. Packing cannot tolerate contaminants (sand, dirt) without scoring the shaft and requires an external source of clean lubricant to inject when pump is used in vacuum service.

Packing is generally limited to water service where leakage can be tolerated. Packing is required by the National Fire Protection Agency (NFPA) for firewater pumps since mechanical seals leak far more fluid when they fail than soft packing.

3.11.4.3 Mechanical seals

Mechanical seals are provided to prevent leakage from the volute where the shaft passes and rotates in contact with the fluid being handled. The sealing between the moving shaft or shaft sleeve and the stationary portion of the conventional stuffing box with composition packing is accomplished by means of rings of composition packing forced between the two surfaces and held tightly in place by a stuffing box gland.

The leakage around the shaft is controlled by adjusting the packing gland studs. The actual sealing surfaces consist of the axial rotating surfaces of the shaft, or shaft sleeve, and the stationary packing. After a certain point, leakage continues regardless of how tight the packing gland studs are tightened. The leakage results in a rapid increase in horsepower that generates heat that cannot be properly dissipated resulting in stuffing box failure.

The earlier undesirable characteristics prohibit the use of packing as the sealing medium between rotating surfaces if the leakage is to be held to an absolute minimum under high pressure. Another factor that makes stuffing boxes unsatisfactory for certain applications is the relatively small lubricating value of many liquids commonly handled by centrifugal pumps in upstream production operations. Therefore seal oil must be introduced into the lantern ring in the stuffing box.

As a result of the aforementioned limitations, mechanical seals were developed. The mechanical seal is used to ensure a pressure tight seal between the rotating shaft, or shaft sleeve, and the pump casing. The seal is performed by the continuous contact between two highly polished flat sealing surfaces, located in a plane perpendicular to the shaft centerline (Fig. 3.99), one surface being connected to the shaft and the other to the stationary portion of the pump. The polished surfaces, made of dissimilar materials, are held in continual contact by a spring, forming a fluid-tight seal between the rotating and stationary members with very small frictional forces.

Mating faces of mechanical seals can be made from a wide variety of materials. Generally, carbon is used for the rotating face. Carbons are available with a number of binders or compositions. The stationary face which is usually the harder material is also available in many materials and compositions. The third sealing member in the mechanical seal is the stationary elastomer which is also available in a wide range of materials with a wide range in cost. A common Buna-N-O-ring might cost $2/dozen whereas the same size O-ring made of Kalrez could cost $1000.

The mechanical seal is preferred over conventional packing except for firewater pump applications. Mechanical seals have the following advantages:

- Simple to install
- Do not require leakage to function properly
- Do not require the constant attention (maintenance) of a skilled mechanic
- Do not require "run-in" time
- Operation does not cause shaft wear
- Offers a lower overall lifecycle cost

Mechanical seals have the following disadvantages:

- Higher first cost then packing
- Minimal failure warning (failure tends to be swift with large leakage)
- Cannot accommodate axial shaft movement

Mechanical surfaces are highly polished so as to:

- Keep leakage to a minimum
- Provide a low friction surface
- Generate the least amount of heat

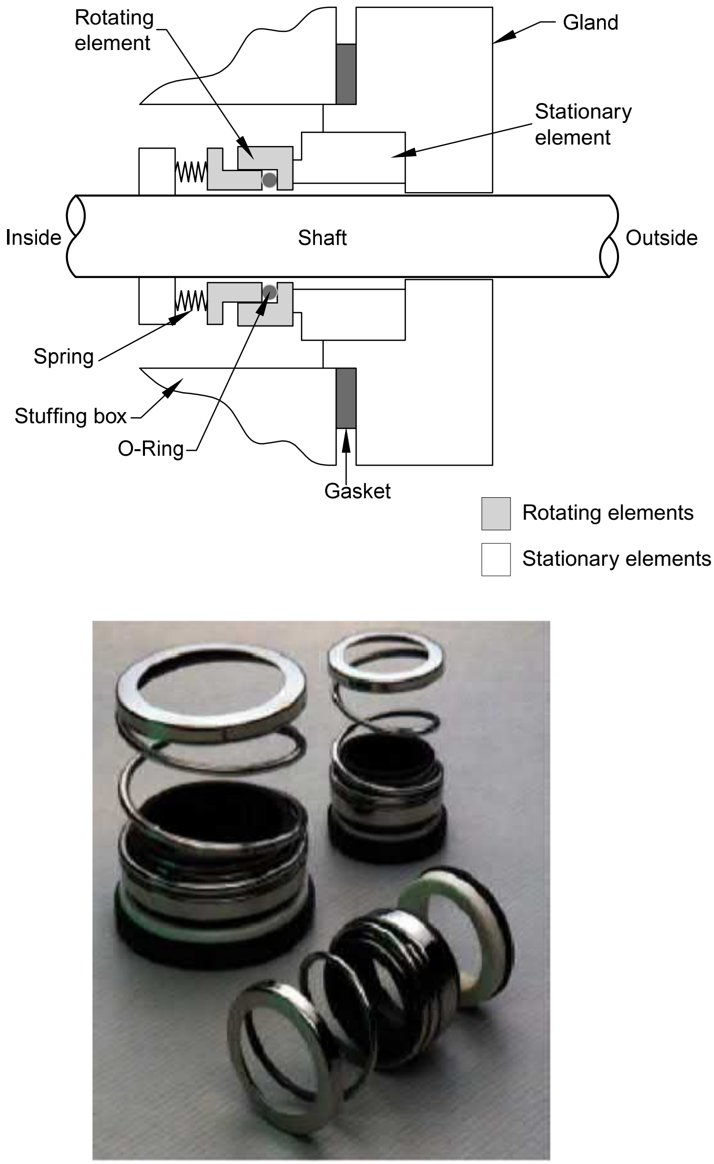


Fig. 3.99 (A) Mechanical seal components.

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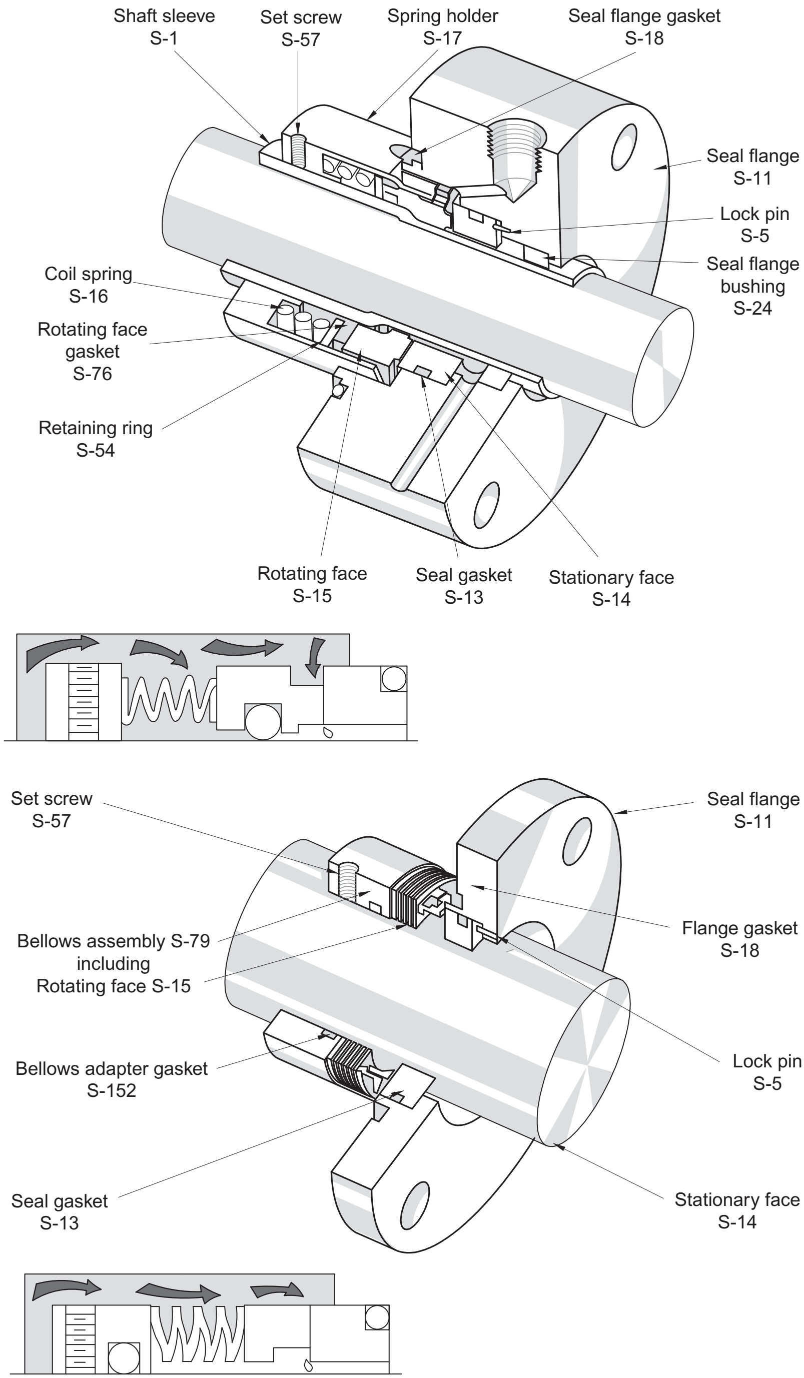


Fig. 3.99— Cont'd (B) (1 of 2): "Spring Pusher" type mechanical seal assembly used for low to moderate temperatures; (2 of 2) "Bellows" type used for high temperatures.

3.11.4.3.1 Types of mechanical seals

Figs. 3.100–3.106 are schematic diagrams showing the general mechanical seal arrangements for the following seal arrangements:

- Internal seal (Fig. 3.100)
- External seal (Fig. 3.101)
- Balanced seal (Fig. 3.102)
  - used for pressures greater than 150 psi (1030 kPa)
- Unbalanced (Fig. 3.103)
  - used for pressures less than 150 psi (1030 kPa)

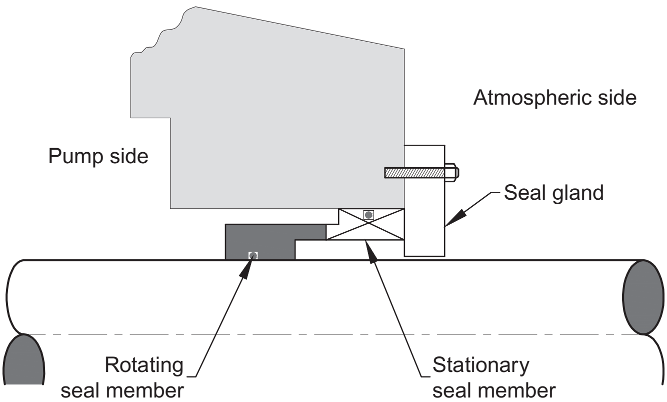


Fig. 3.100 Schematic diagram of internal mechanical seal assembly.

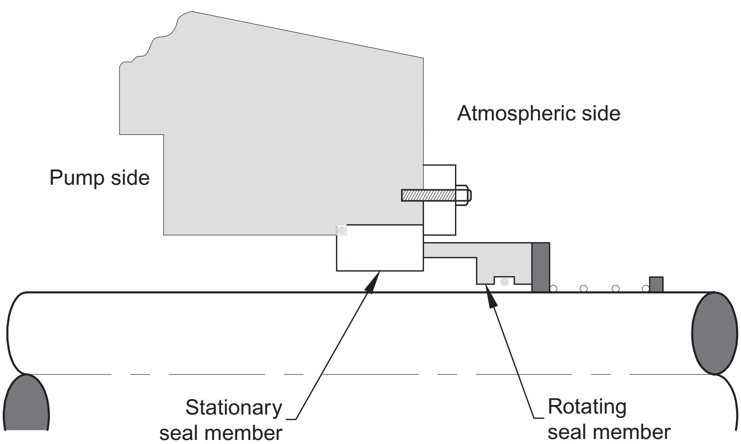


Fig. 3.101 Schematic diagram of external mechanical seal assembly.

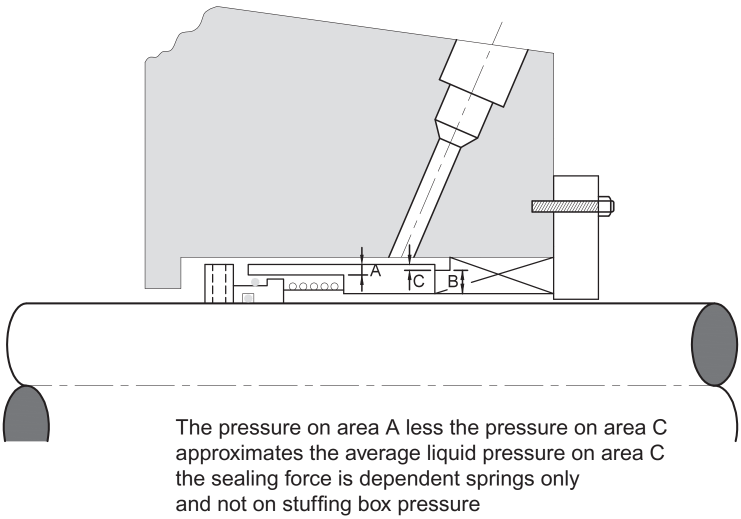


Fig. 3.102 Schematic diagram of balanced mechanical seal assembly.

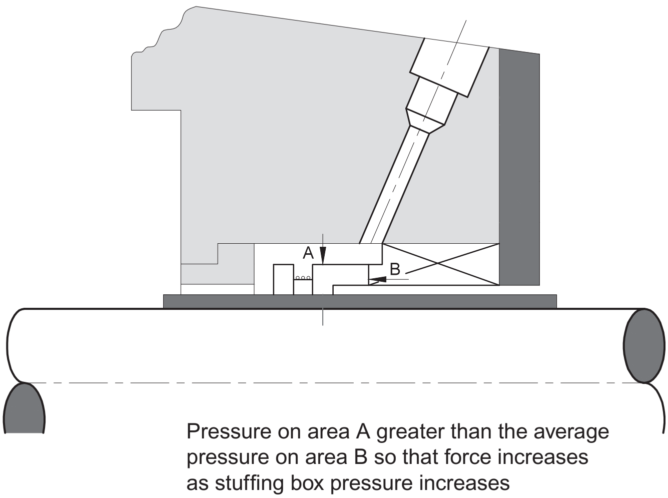


Fig. 3.103 Schematic diagram of unbalanced mechanical seal assembly.

- Double seal (Fig. 3.104)
  - Consists of two mechanical seals placed back to back
  - Used when handling toxic or highly flammable fluids that cannot be permitted to the atmosphere
  - Used with corrosive or abrasive fluids, or fluids at extreme temperatures
  - A throttle bushing and alarm downstream of the seal between the clean fluid and the atmosphere may be installed to warn of failure of this part of the seal

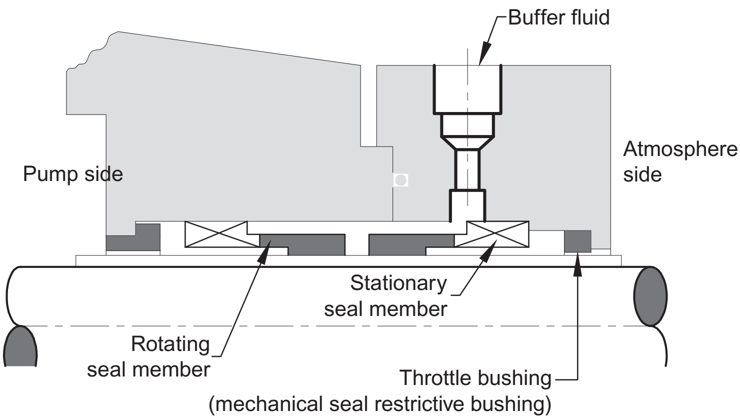


Fig. 3.104 Schematic diagram of double seal assembly.

- Tandem seal (Figs. 3.105 and 3.106)
  - Specialized type of double seal
  - Consists of an inside and outside seal, separated by a space that is sealed by a light lubricating oil, called seal oil
  - Pressure switch is usually placed in the seal oil section which sends a signal to alarm if the inner seal should fail
  - Used in critical service where leakage due to seal failure must be prohibited
- Cartridge seal
  - Preassembled onto a sleeve and seated
  - More expensive but does not require much skill to be properly installed
  - Primarily used in multistage vertical and axially split horizontal pumps
  - Not applied to overhung pumps due to space and general pump design constraints

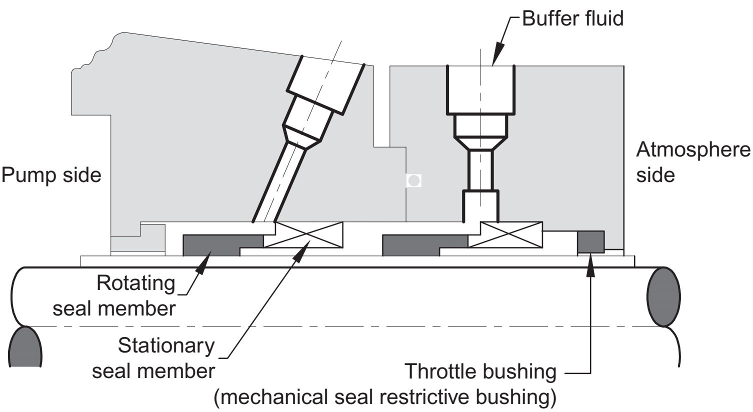


Fig. 3.105 Schematic diagram of tandem seal assembly.

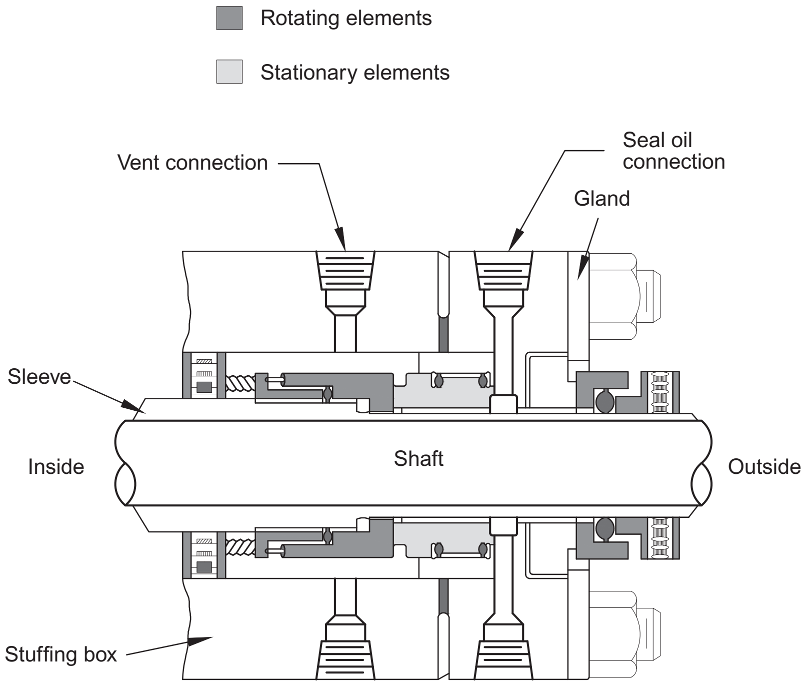


Fig. 3.106 Schematic diagram of tandem seal assembly.

Hydrocarbon leakage from a mechanical seal can form deposits or coke on contact with the atmosphere. This coking can damage the seal. To minimize coking, a stream quench is often employed to wash away the coke as it forms.

The stuffing box and mechanical seal gland must be of the four-bolt design. This will eliminate cocking of the gland which can occur if a two-bolt design gland arrangement is used. As three points determine a plane, the gland which holds the stationary seal component would be tight against the box face, yet it could be cocked, and the rotating seal face, which is generally the softer material, would wear unevenly if the stationary seal member was cocked. With four bolts, all four gland nuts will have to bear on the gland.

With a double seal, the inner seal prevents the pumped liquid from entering the seal cavity, while the out seal prevents the buffer liquid from leaking to atmosphere. This system is pressurized and requires either a pressure maintenance pump or a pressurizing gas blanket on the buffer liquid to accomplish the sealing. A pressurized external fluid reservoir with forced circulation is typically used with a double seal arrangement.

Double seals are used in toxic services where a pressurized clean seal fluid is designed to leak the lower pressure process if there is a failure in the primary seal. A throttle bushing and alarm downstream of the seal between the clean fluid and the atmosphere may be installed to warn of failure of this part of the seal. Emission.

To accomplish this, tandem or double mechanical seals must be used. These dual seals use a buffer liquid in the box cavity. As a portion of the buffer liquid passes through the inner seal to the pumped product, and a portion leaks through the outer seal to the atmosphere, the buffer liquid must be compatible with the pumped liquid and safe to be discharged to the atmosphere.

With a tandem seal, both seals work in the same direction, restricting flow outward from the box throat. This permits the seal cavity to operate at atmospheric pressure. A simple reservoir vented to atmosphere or to a flare system is sufficient. A nonpressurized external fluid reservoir with forced circulation is typically used with a tandem seal arrangement. The reservoir can be fitted with any number of bells and whistles, such as level switches or gauges, pressure switches, and cooling or hearing coils.

Volatile organic hydrocarbons (VOCs) are limited by the EPA. Most regulatory agencies have adapted policies of no increased emissions should a plant expansion be undertaken. In this case, existing VOC sources, such as pumps, must be retrofitted to upgrade their shaft sealing to reduce VOC.

A throttle bushing is a restrictive bushing or sealing device designed to limit flow out of the seal in the event of failure. Since a leaking seal will increase pressure between the seal and throttle bushing, throttle bushings are used whenever a seal failure alarm is required. Figs. 3.107–3.109 show schematic diagrams of several different types of seal designs. Tandem seals are used in critical service where leakage due to seal failure must be prohibited. They are constructed of two seal assemblies acting in series and are separated by a buffer fluid at a pressure lower than the sealing pressure. If the primary seal fails, the pressure or reservoir level in the buffer fluid system will increase, triggering an alarm. There may be a throttle bushing and alarm downstream of the secondary seal to provide warning of secondary seal failure as well.

3.11.4.3.2 Factors affecting seal performance

Factors affecting seal performance include:

- Abrasives
- High temperatures
- Fluctuating seal pressures
- Misalignment

The earlier factors should not be ignored because a large majority of centrifugal pump failures are due to mechanical seal failure.

Abrasives can be kept away from the close clearance seal faces by injecting a flushing fluid into the seal chamber. Solid contents up to about 200 ppm can be handled with standard flushing arrangements. Above 200 ppm and particle sizes greater than 10 μm, a cyclone separator should be installed in the standard flushing line. Above 2% by weight, refer to API 610 for guidance.

High temperature can cause premature elastomer failure, reduce the fluid's viscosity and lubricity, and increase the fluid's vapor pressure. Close tolerances of the seal faces generate a lot of heat that must be removed from the seal faces. To ensure long life, narrow seal faces are used to minimize the surface areas in contact

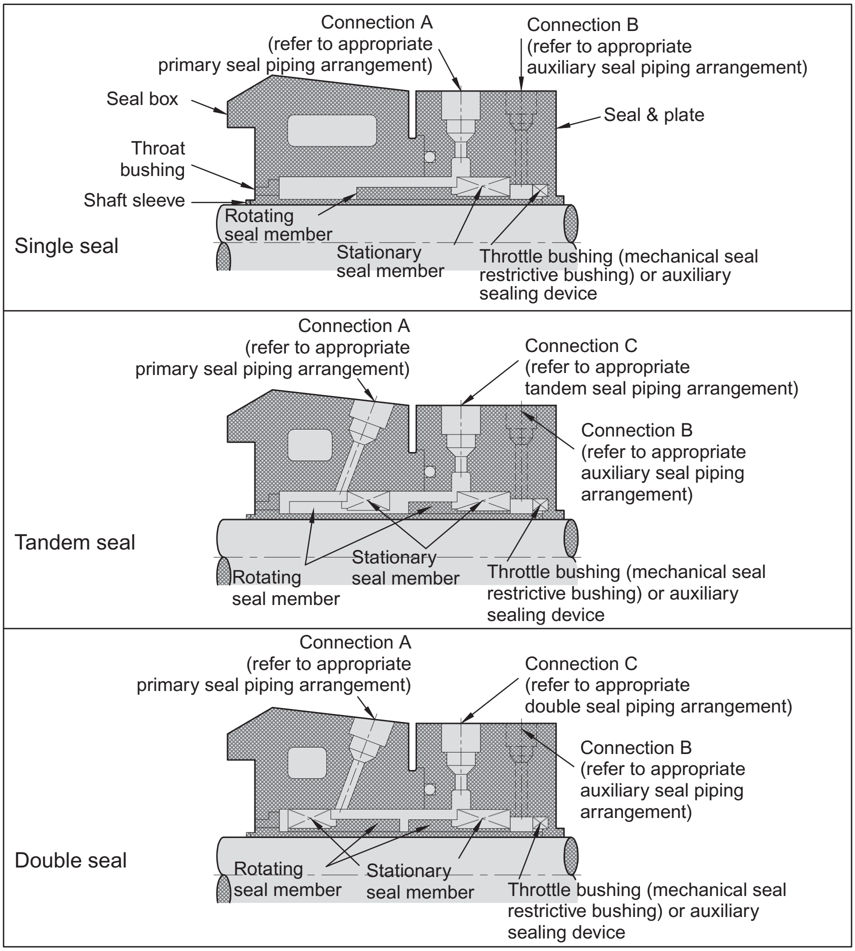


Fig. 3.107 Schematic diagram showing the general mechanical seal arrangements. Courtesy Durametallic Corporation.

with each other. Heat generated at the seal interface is removed by using seals with high heat conductance properties and a good, cool flashing system. Seal fluid should be 25°F below its vaporization point at seal chamber pressures. This is usually obtained by using external cooling, such as finned tubes, or small, externally driven coolers.

Fluctuating seal pressures can cause alterations in seal face wear patterns and reduce seal life expectancy. Maintaining a near-constant seal pressure is possible only by maintaining near constant discharge pressures. System changes should be relatively slow. When starting pumps, the discharge valve should be nearly closed to create higher discharge and seal pressures.

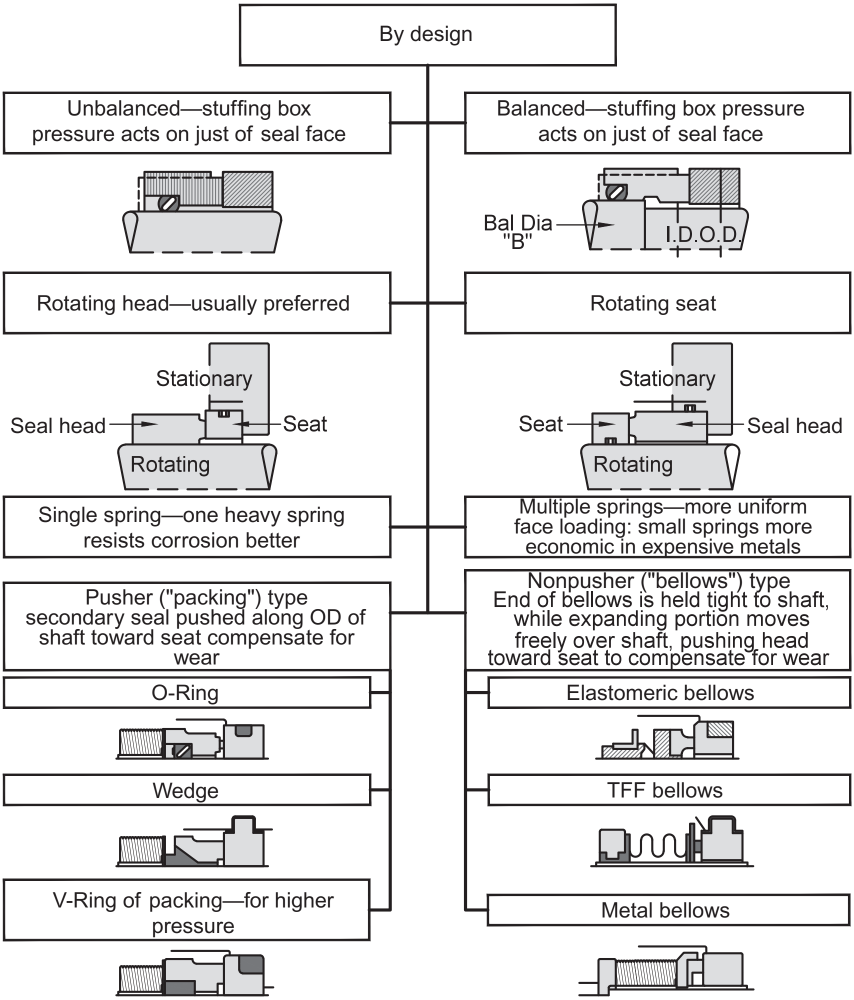


Fig. 3.108 Tree diagram showing several different seal designs.

To avoid *misalignment* it is important to ensure the pump is free of all piping loads. This is done by aligning the pump and motor without the piping attached and then connecting the piping without disturbing the alignment indicators. Excessive piping loads on the pump case will result in seal deflection and seal failures.

3.11.4.3.3 API seal classification code

In order to clarify seal type descriptions with a concise and brief technique, an API Seal Classification Code is used. It is a five-letter code described by:

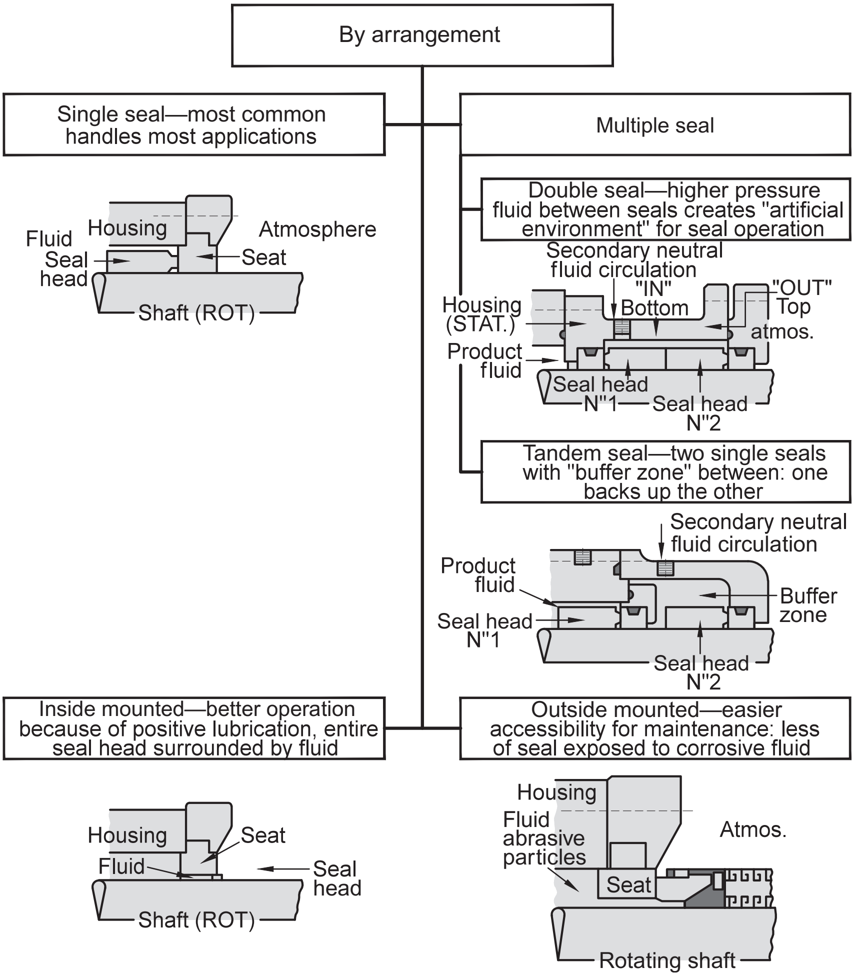


Fig. 3.109 Tree diagram showing different seal arrangements. Courtesy of Durametallic Corporation.

First letter	B—Balanced
	U—Unbalanced
Second letter	S—Single
	D—Double
	T—Tandem
Third letter	P—Plain end type
	T—Throttle bushing
Fourth letter	Gasket material (refer to Table 3.8)

Table 3.8 Gasket material

Fourth letter	**Stationary seal ring gasket**	**Seal ring to sleeve gasket**[a]
E	Fluoroelastomer	TFE
F	Fluoroelastomer	Fluoroelastomer
G	TFE	TFE
H	Nitrile	Nitrile
I	FFKM elastomer	FFKM elastomer
R	Graphite foil	Graphite foil
X	As specified	As specified

[a]Refer to Table 3.7 for temperature limitations of above materials.

- Temperature limits should be in accordance with Table 3.9 (API 610)
- Fluoroelastomers should not be used when aromatics are present in pumped fluids
- Teflon (TFE) and Kalrez (FFKM) are the most commonly used gasket and "O-ring" materials
- Nitrile (Buna N) is not suitable in sour or high temperature service
- Graphite is used in high temperature, corrosive service but it is not suitable in applications where a lot of flexing is required

Fifth letter	Face material (refer to Table 3.10)

- Stellite (J) or Ni-Rest (K) is acceptable for standard oilfield applications
- L through N should be specified for applications where abrasive materials are present

For example, a *BTPEL* seal is a balanced, tandem seal with plain end plate, fluoroelastomer O-rings, a carbon steel ring, and a tungsten carbide-1 mating seal ring.

Table 3.9 Temperature limitation for mechanical seal gaskets

	Ambient or pumping temperature (F)	
Gasket material	**Minimum**	**Maximum**
TFE	−150	500
Nitrile (Buna-N)	−40	250
Neoprene	0	200
Fluoroelastomer	0	400
Metal bellows	[a]	[a]
FFKM elastomer	10	500
Graphite foil	−400	+750[b]

[a]Consult manufacturer.
[b]Maximum temperature is +1600 F for nonoxidizing atmospheres: consult manufacturer.

Table 3.10 Face material

Fifth letter	Seal ring	Mating seal ring
J	Carbon	Stellite
K	Carbon	N-resist
L	Carbon	Tungsten carbide-1
M	Carbon	Tungsten carbide-2
N	Carbon	Silicon carbide
X	As specified	As specified

3.11.4.3.4 API 610 seal piping plans

The references for mechanical seal systems are found in API 610—Centrifugal Pumps and API 682—Shaft Sealing Systems. Piping systems are required for mechanical seals to provide a flushing fluid across seal faces, establish flow paths for various seal configurations, and establish location of components (e.g., coolers, reservoirs, pressure switches). The API has established standard seal piping systems that are designed by the plan numbers shown in Figs. 3.110 and 3.111). Table 3.11 is a general use guide to provide a starting point for making a cost/benefit decision for any installation.

3.11.5 Balancing drums and bearings

3.11.5.1 Balancing drums

In multistage pumps where the impellers all face in one direction, the axial thrust is cumulative and can reach very high values. In this configuration, excessively large thrust bearings would be required. Another alternative is to use a hydraulic balancing device that balances the axial thrust and reduces the pressure on the stuffing box adjacent to the last stage of the impeller. The hydraulic balancing device may be a balancing drum, a balancing disk, or a combination of the two.

A typical balancing drum is shown in Fig. 3.112. The balancing drum has the discharge pressure on one side and a lower pressure, usually suction pressure, on the other side. The cross-sectional face area of the drum is determined to approximately counterbalance the hydraulic thrust from the impellers. Balance drums reduce the pressure in the stuffing box on the discharge end.

The forces acting on the balancing drum in Fig. 3.112 are the following:

- *Forces toward the discharge end*: The discharge pressure multiplied by the front balancing area (*Area B*) of the drum
- *Forces toward the suction end*: The backpressure in the balancing chamber multiplied by the back balancing area (*Area C*)

The forces toward the discharge end are greater than the forces toward the second end, thereby counterbalancing the axial thrust exerted upon the single-suction impellers. In practice, 100% balance is unattainable and the slight but predictable unbalance can be carried on a thrust bearing, the balancing drum is often designed to balance only 90%–95% of total impeller thrust.

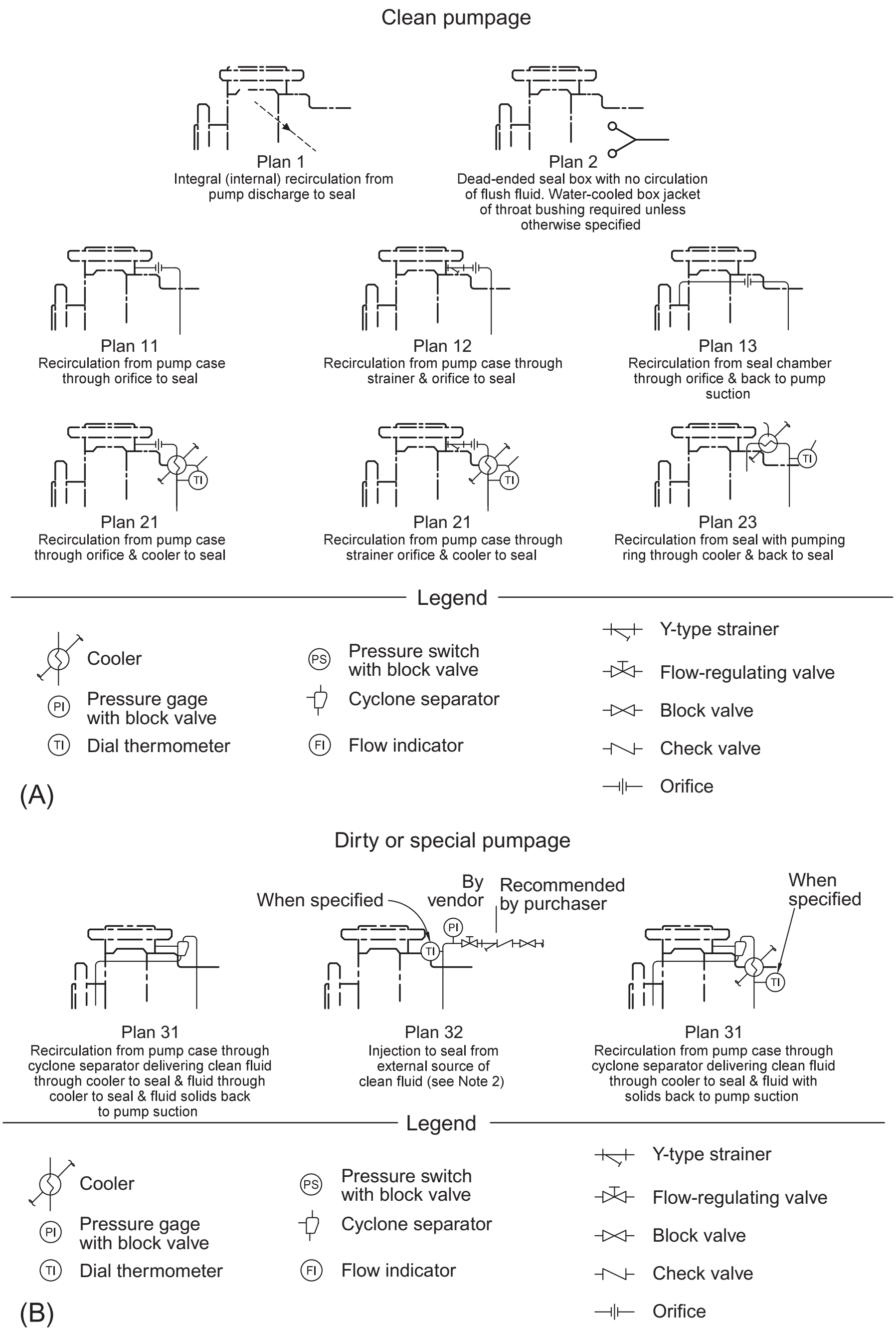


Fig. 3.110 (A) Schematic diagram showing API standard seal piping systems: clean pumpage; bottom— dirty or special pumpage. (B) Schematic diagram showing API standard seal piping systems: dirty or special pumpage.

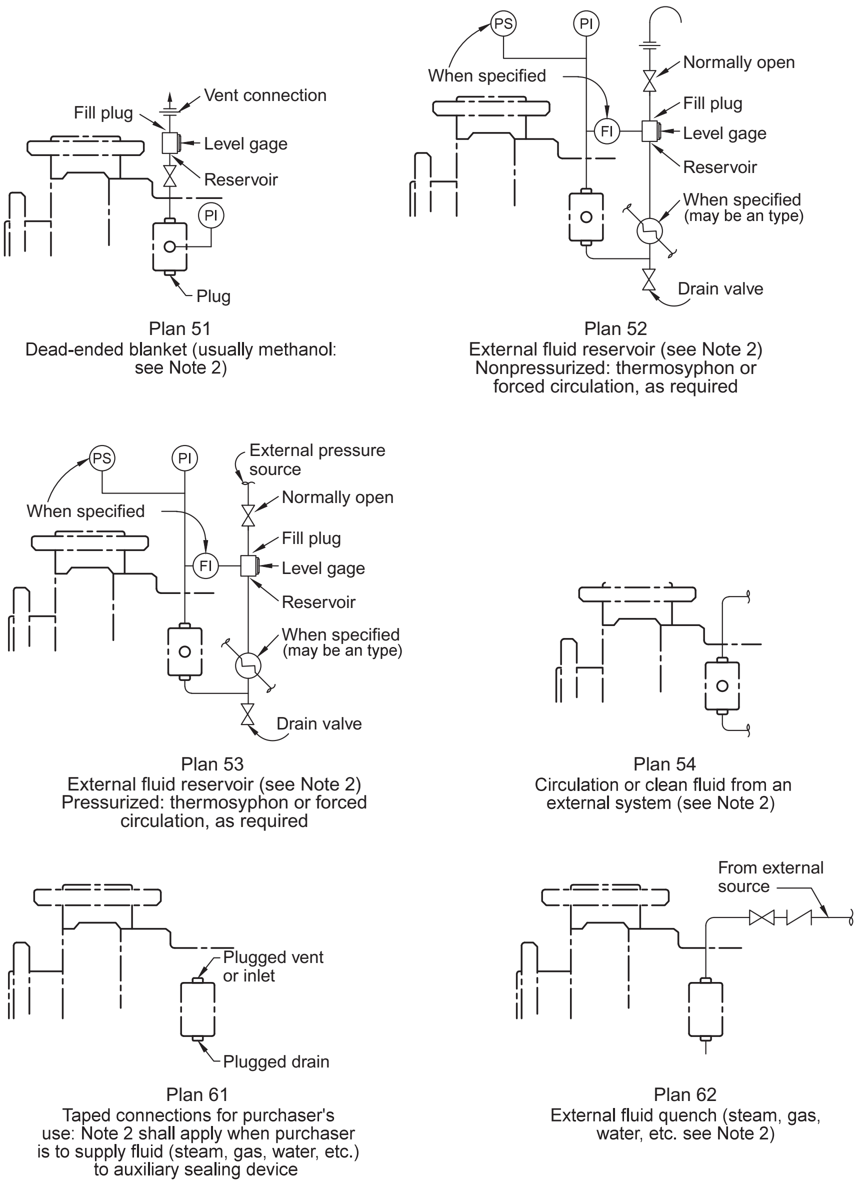


Fig. 3.111 Schematic diagram showing API Standard seal piping systems.

3.11.5.2 Bearings

As shown in Fig. 3.113, besides rotating, the shaft may tend to move in two other directions: radially due to loads acting perpendicular to the shaft centerline, and axially due to thrust loads parallel to the shaft centerline.

Table 3.11 Seal usage guide

Service	Pressure		Seal system	Seal piping plans
	kPa	psi		
Water	<690	<100	Packing	Plan 12 or 13
	>690	>100	Balanced mechanical seal	Plan 12 or 13
Hydrocarbons—lease facilities	<690	<100	Mechanical seal	Plan 12 or 13
	690–3450	100–500	Balanced mechanical seal	Plan 12 or 13
	>3450	>500	Balanced tandem seal	Plan 12 or 13 with 52
Hydrocarbons—major installations and plants	<690	<100	Mechanical seal with throttle bushing	Plan 12 or 13
	>690	>100	Balanced tandem seals with throttle bushing	Plan 12 or 13
Toxic fluids	All	All	Double balanced seal with throttle bushing	Plan 12 or 13 with 52

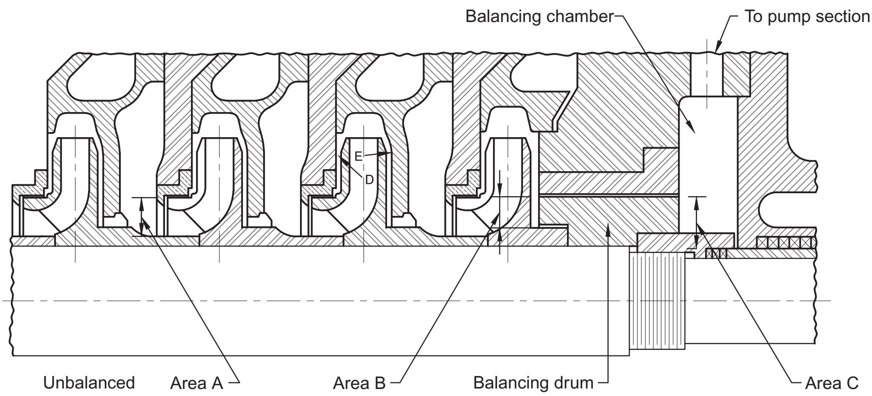


Fig. 3.112 Schematic diagram illustrating a balancing drum.

The function of bearings in centrifugal pumps is to keep the shaft and rotor in correct alignment with the stationary parts under the action of radial and transverse loads. Bearings that give radial positioning to the rotor are known as line bearings or radial bearings, and those that locate the rotor axially are called thrust bearings. In most applications the thrust bearings actually serve as both thrust and radial bearings.

Pumps of the same basic design are often made with two or more different types of bearings to meet requirements of either service conditions or purchaser preference. In most pumps, bearings are classified as either hydrodynamically lubricated (oil film sleeve type) or rolling contact (antifriction).

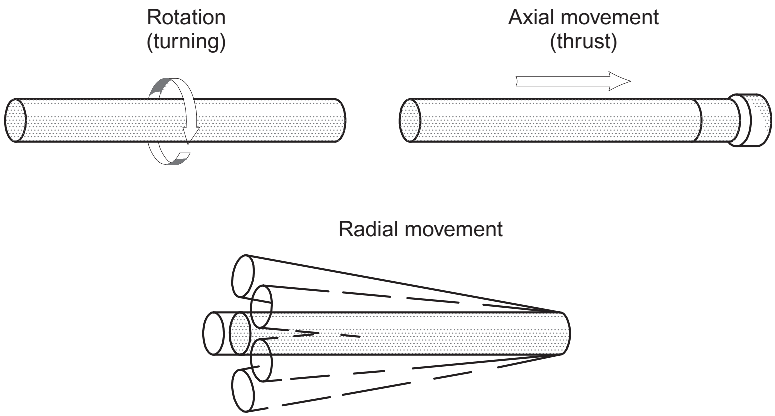


Fig. 3.113 Shaft movements.

3.11.5.2.1 Hydrodynamic lubricated bearings

The primary motion of the hydrodynamically lubricated type is a sliding of one surface over another. The surfaces of this bearing are separated by a relatively thin layer of lubricant that prevents any metallic contact, except when starting or stopping under load. With the roller type the primary motion is a rolling of one surface over another.

The three most commonly used hydrodynamic lubricated bearings are as follows:

- Journal radial bearing
- Flat plate thrust bearing
- Kingsbury

Journal radial bearing: As shown in Fig. 3.114, oil enters the bearing through supply holes strategically placed along the bearing circumference. The oil then flows axially and circumferentially along the bearing and out the ends. As the oil flows through the bearing, radial loads compress the oil film, which generates high pressure within the bearing. This type of bearing is suitable for radial loads only.

For most applications, the yield strength of the bearing material in compression will dictate the maximum load a journal bearing can handle. Since the fluid is all that contacts the bearing material, the maximum loading capability is a function of fluid pressure generated within the bearing. Maximum allowable oil pressures for common bearing materials are listed in Table 3.12.

Flat plate thrust bearing: As shown in Fig. 3.115, flat plate thrust bearings consist of a stationary flat plate and a rotating flat plate that are mounted on the shaft. The stationary flat plate has radial grooves in which oil is injected. Oil then flows circumferentially and radially out of the bearing. This type of bearing is designed for axial loads only. The maximum loading for this type of bearing should not exceed 50 psi (1030 kPa).

Kingsbury Thrust Bearing: As shown in Fig. 3.116, the bearing consists of several radially mounted pads that tilt circumferentially. Oil is fed into the bearing near the

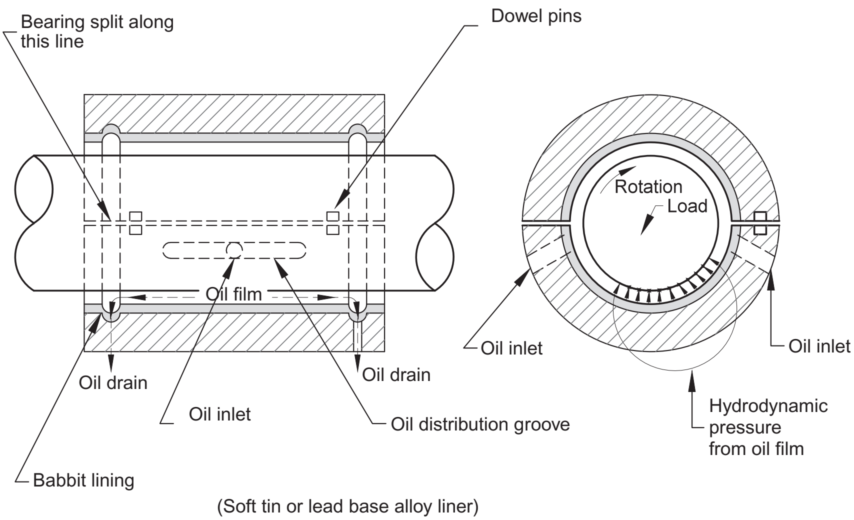


Fig. 3.114 Schematic diagram of a journal radial bearing.

Table 3.12 Maximum allowable oil pressures

Bearing material	Maximum oil pressure (kPa)	Maximum oil pressure (psi)
Lead-base	1400–5500	200–800
Tin-base	5500–7000	800–1000
Cadmium-base	8300–10,300	1200–1500
Copper-lead	13,800–27,600	2000–4000
Silver-lead-indium	34,500	5000
Bronzes	68,900	10,000

center and it flows radially outward. When the shaft is stationary, the pads lie with their surfaces parallel to the shaft face. As the bearing is started, an oil film is created between the pad and shaft. Each pad tilts to an angle that assures the proper distribution of film pressures within the bearing. This type of bearing is designed for axial loads only.

The maximum allowable loading for this type of bearing should not exceed 500 psi (3450 kPa). In multistage pumps, a Kingsbury bearing may be required above approximately 4000 rpm.

The advantages of hydrodynamically lubricated bearings are as follows:

- Bearing life is not a function of speed or load. Very little wear occurs.
- Oil forces tend to hold the shaft in the center of the bearing which provides some degree of dampening.
- Journal and flat plate thrust bearings have a very low relative cost.

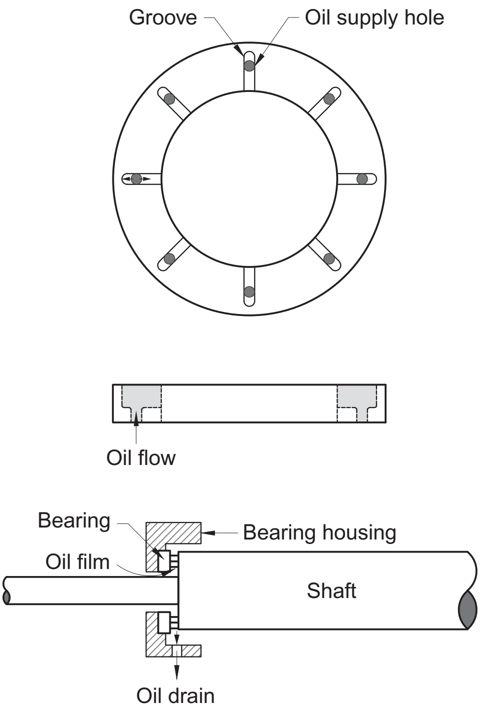


Fig. 3.115 Schematic diagram of a flat plate thrust bearing.

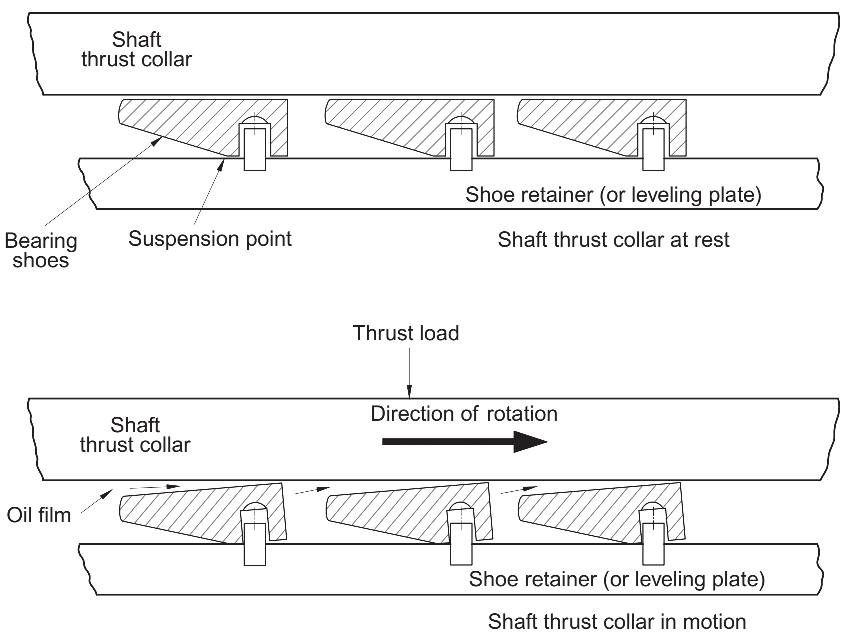


Fig. 3.116 Schematic diagram of a Kingsbury thrust bearing.

The disadvantages of hydrodynamically lubricated bearing are as follows:

- Require a continuous supply of cool uncontaminated oil at a constant fixed pressure.
- Journal bearings can tolerate no axial load, while flat plate and Kingsbury bearings can tolerate no radial load.
- They have a relatively high friction drag as opposed to rolling contact bearings and thus have higher horsepower losses.

3.11.5.2.2 Rolling contact bearings

Rolling contact or antifriction bearings are used to support the applied load by actual metal-to-metal contact over a relatively small area. Rolling contact bearings are based on the following principles of operation:

- A continuous lubrication system is generally not required. Occasional greasing of the bearing is all that is required.
- They offer very little resistance to movement, therefore, horsepower losses are smaller than with hydrodynamically lubricated bearings.
- They can be designed for both thrust and radial loads simultaneously.
- They can be designed to be self-aligning.

The disadvantages of rolling contact bearings are as follows:

- They have a specific life for the design load.
- They have limited load/speed capability.

The most commonly used rolling contact bearings are as follows:

- Ball bearing (single/double row)
- Spherical bearing (single/double row)
- Cylindrical bearing

Ball bearing: Fig. 3.117 is a schematic diagram of a single-row ball bearing. The principal action of a "single-row" ball bearing is the rolling of the bearing element (ball) between inner and outer races. The inner race is fixed to the shaft and it rotates with the shaft. The outer race is rigidly attached to the bearing housing. There is point contact between balls and races. Fig. 3.118 is a schematic diagram of a double-row ball bearing. The double-row bearing is identical in operation to the single-row ball bearing. This bearing has a higher radial and axial load capacity than single-row ball bearings.

Spherical bearing: Fig. 3.119 is a schematic diagram of a double-row spherical bearing. The spherical bearing is similar to the ball bearing. This bearing consists of a rolling element between a fixed and a rotating race. The elements are barrel shaped with a radius of curvature in the axial direction equal to the radius of the inner surface of the outer ring. There is line contact between the elements and the races. The spherical bearing is noteworthy for its ability to carry heavy radial loads and moderate thrust loads combined with its ability to run properly under conditions of misalignment.

Cylindrical bearing: Fig. 3.120 is a schematic diagram of a cylindrical bearing. The cylindrical bearing is identical in operation to other rolling contact bearings. The rolling element of this bearing is a cylinder whose axis is parallel to the shaft. There is

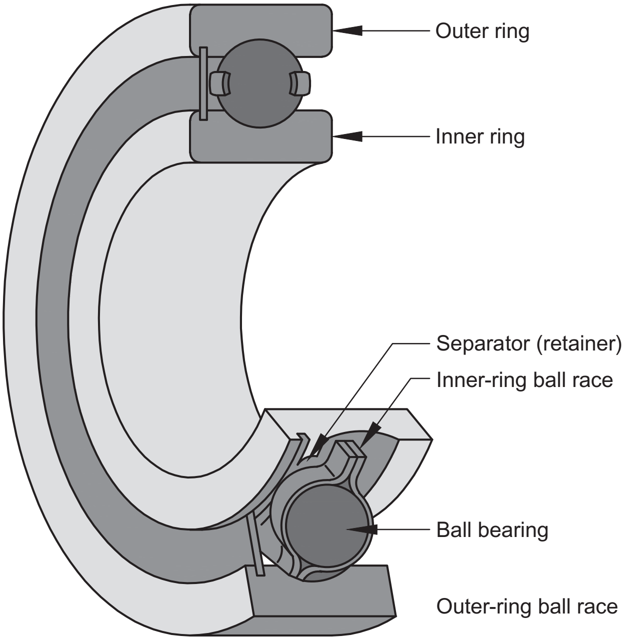


Fig. 3.117 Schematic diagram of a single-row ball bearing.

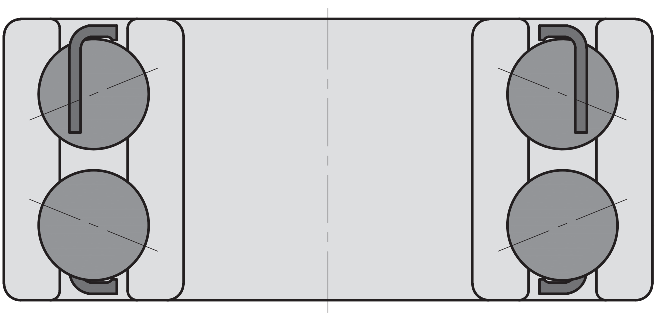


Fig. 3.118 Schematic diagram of a double-row ball bearing.

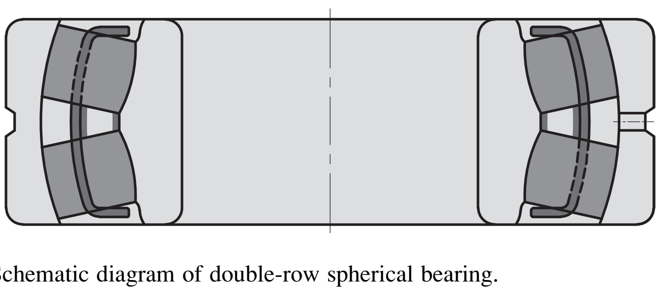


Fig. 3.119 Schematic diagram of double-row spherical bearing.

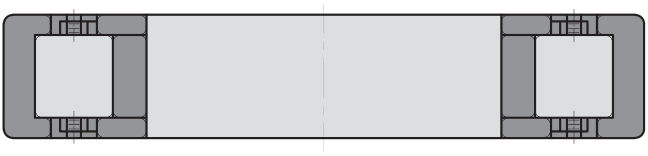


Fig. 3.120 Schematic diagram of a cylindrical bearing.

line contact between the races and element. This bearing can carry a heavier load than a ball bearing and can accommodate some axial movement.

Fig. 3.121 illustrates a pump shaft supported by both radial and thrust ball bearings.

The load-carrying capability of rolling contact bearings is a function of the number of revolutions the bearing will accrue in its life. This relationship can be expressed as:

$$L_n = \left(\frac{C}{W}\right)^3 \tag{3.23}$$

Where:

$L_{n=}$ Life of bearing in millions of revolutions. Per API 610, this life is the number of revolutions accrued in 3 years continuous operation
$W =$ Bearing load, Ib
$C =$ Bearing constant. Represents the load that the bearing can carry for 1 million revolutions, with only 10% failure rate

This equation should be used to verify manufacturer's bearing selection for any given application.

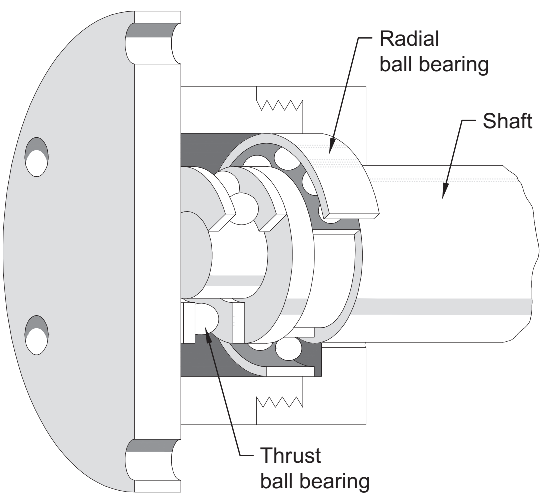


Fig. 3.121 Pump shaft supported by both radial and thrust ball bearings.

3.11.5.2.3 Bearing selection guidelines

Bearing design and selection are the manufacturer's responsibility. However, the following criteria should be considered when a bearing is to be selected for any given application:

- The continuous lubrication system required by hydrodynamic lubricated bearings is a high-maintenance item. Therefore rolling contact bearings should be specified whenever possible.
- The bearings selected for most pumps encountered in oil and gas applications should be capable of axial and radial loads. The designer may wish to specify this capability on the pump data sheet.
- Whenever rolling contact bearings are specified, the life should be specified. Per API Standard 610 this life is three years continuous operation.
- API Standard 610 requires hydrodynamic bearing for the following applications:
  – All barrel pumps
  – When the product of bearing diameter in (mm) and pump rpm is greater than or equal to 300,000
  – When standard rolling contact bearings fail to meet the 25,000 h of operation criteria
  – When the product of pump rated horsepower and rated rpm is 2.7 million or greater

3.11.6 Lubrication systems

Bearings require just enough oil to keep them evenly coated. Too much oil can cause bearings to overheat and fail. The choice of lubrication for pump bearings is dictated by the type of bearings, application requirements, cost considerations, and purchaser preference. The types of lubrication systems are as follows:

- Grease systems
- Splash "oil ring" systems
- Constant level oiler
- Force feed (pressurized)

There is no industry practice or recommended practice that dictates a specific lubrication for a given application. The lubrication system is normally determined by the pump manufacturer unless experience dictates otherwise. Hydrodynamically lubricated bearings commonly require a force-feed (pressurized) oil system. Lubrication for rolling contact bearings (antifriction) is dependent on the type of bearing used and the operating speed. Ball bearings used in centrifugal pumps are usually grease lubricated for pump operating speeds below 5000 rpm and oil lubricated for operating speeds above 5000 rpm. However, many oil field pumps with operating speeds below 5000 rpm are supplied with an oil-lubricated system due to purchaser preference.

In pumps with oil-lubricated ball bearings, a suitable oil level must be maintained in the housing.

3.11.6.1 Grease systems

Grease is injected into the bearing housing through an external fitting. Grease provides cooling and lubrication for the contact surfaces. Grease used only with ball or roller bearings. The advantages of grease systems are as follows:

- Bearing housing design is simple
- Easy to maintain
- Low installed and low operating relative costs

Grease systems are not suitable in applications requiring lots of heat removal or highly loaded bearings.

3.11.6.2 Splash "oil ring" systems

As illustrated in Figs. 3.122 and 3.123, oil rings ride on the shaft and picks up lubricant from the reservoir and carries the oil to top of the drive shaft, where the oil then flows into the bearings. Splash systems are self-pumping and have low installed and operating relative costs. They are not suitable for highly loaded shafts.

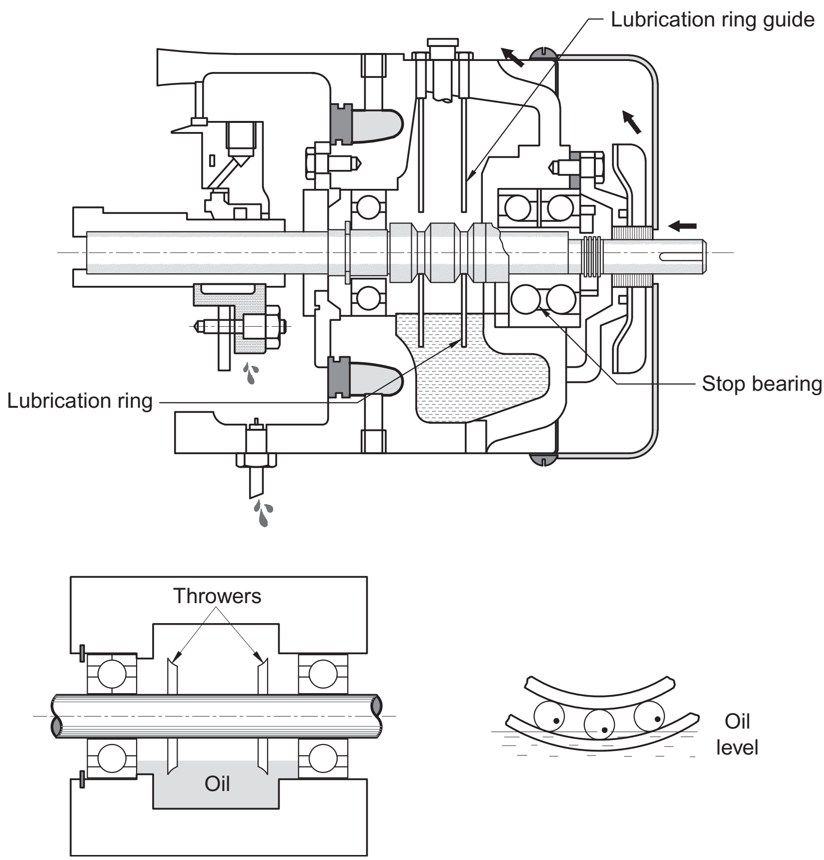


Fig. 3.122 (Top) Schematic of roller ball bearings fitted with splash "oil ring" system; (bottom) cutaway of splash oil ring lubrication.

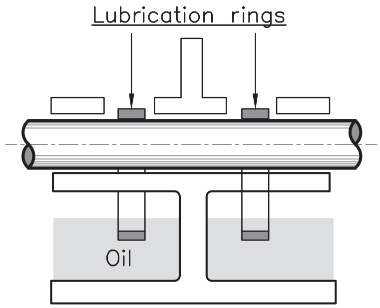


Fig. 3.123 Schematic diagram showing splash oil rings lubricating a bearing.

3.11.6.3 Constant level oiler

In pumps with oil-lubricated ball bearings, a suitable oil level must be maintained in the housing. As shown in Figs. 3.124 and 3.125, this level should be at roughly the center of the lowermost ball of a stationary bearing. A higher oil level can result in damaged bearings since the heat will not be able to dissipate quickly enough to keep the bearings from overheating. Due to the advantages of interchangeability, some pump lines are built with bearing housings that can be adapted either to oil or grease lubrication with a minimum of modification (Fig. 3.126).

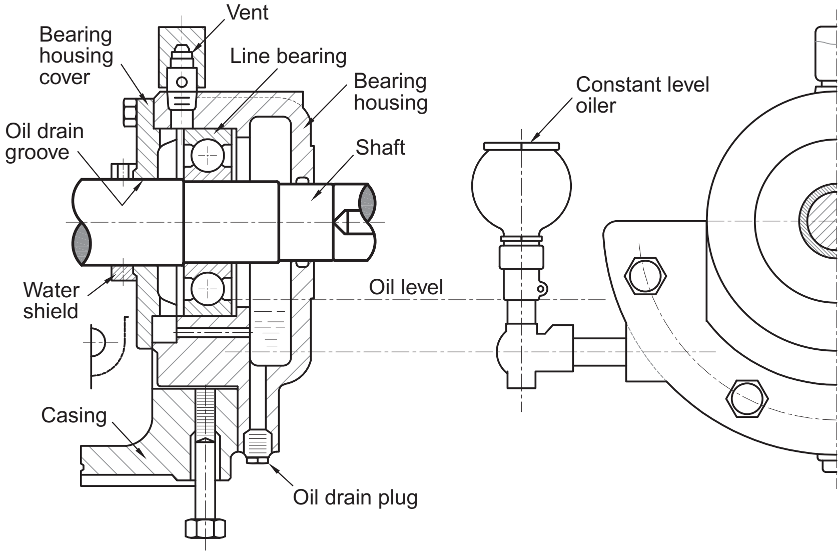


Fig. 3.124 Schematic of a constant level oiler system.

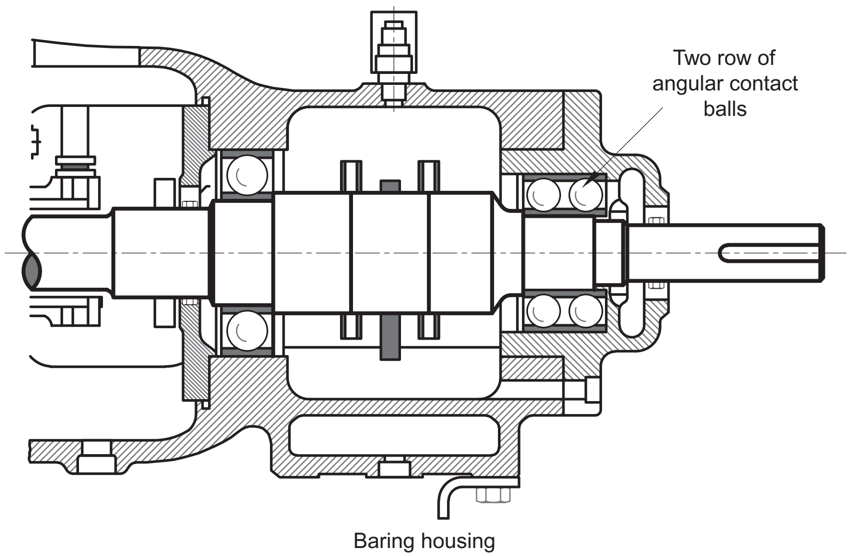


Fig. 3.125 Schematic diagram illustrating an inappropriate oil bath lubrication system.

3.11.6.4 Force-feed (pressurized) systems

As shown in Fig. 3.127, force-feed systems use a small rotary pump to move liquid from the bearing housing reservoir, through a cooler and then back to the bearings. Hydrodynamic lubricated bearings (sleeve bearings) may require a pressurized lube oil system. The pressurized lube oil system is normally designed to supply oil at a suitable pressure to the pump bearings, driver, and any other driven equipment, including gears and continuously lubricated couplings. Detailed requirements for a pump pressurized lube oil system are outlined in API Standard 610 and API Standard 614.

Steps in selection of a pressurized lube oil system are as follows:

1. Determine bearing heat rejection and required oil flow for bearing size and speed.
2. Select the shaft-driven oil pump and/or auxiliary oil pump. Select an oil pump of adequate capacity or rating for the appropriate rpm, including driver oil requirements, if specified.
3. Select the heat exchanger. The heat exchanger must meet all heat transfer requirements, including driver requirements when specified.
4. Select the filter to meet micron particle size and flow requirements for the pump and driver bearings.
5. Select the reservoir (it should have a minimum of three minutes retention time). Divide the required oil flow, including driver requirements, into the reservoir capacity to obtain retention time.

3.11.7 Cooling water systems

The control of temperature generated in the stuffing box and bearings housings is an important factor in extending the life of packing, seals, and bearings. Figs. 3.128 and 3.129 from API Standard show typical cooling water piping for overhung-impeller pumps and between-bearing pumps.

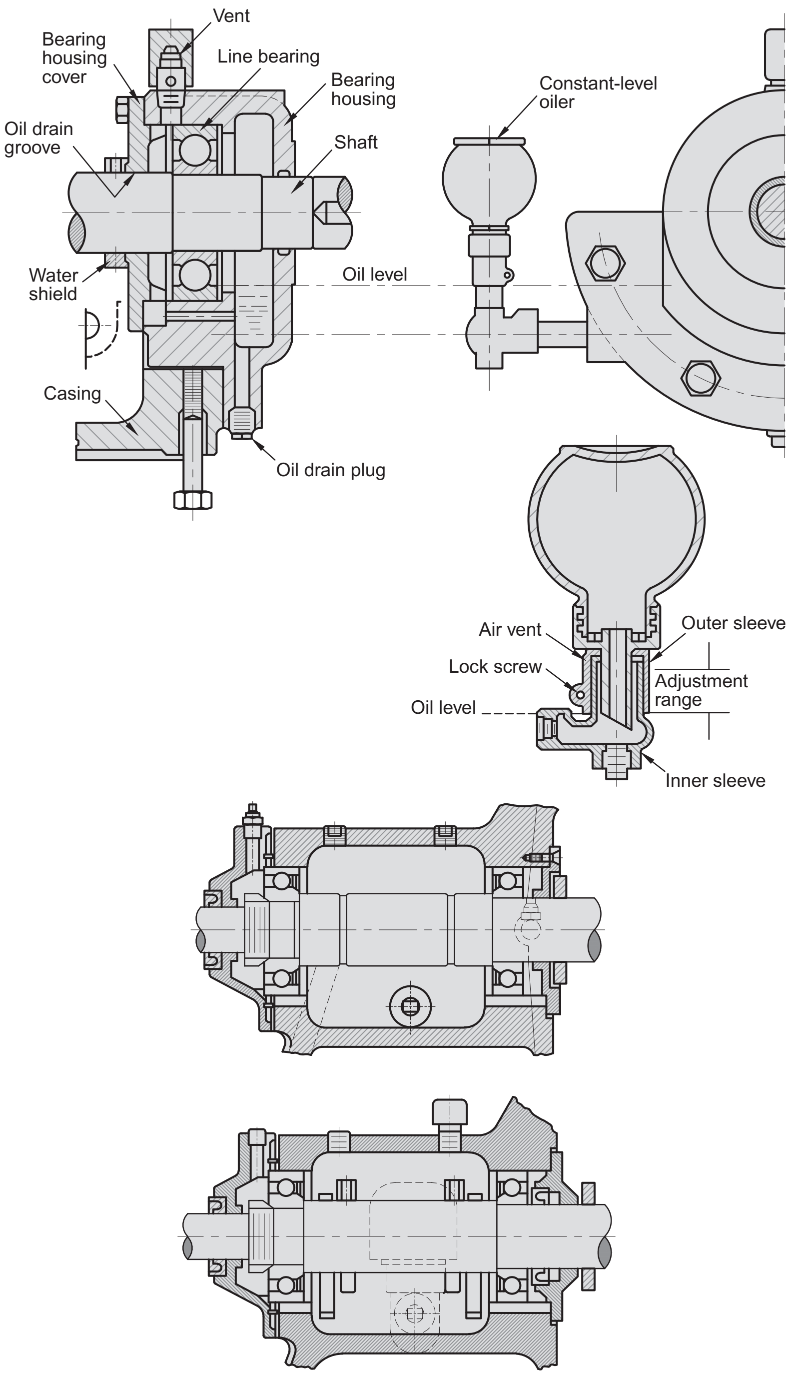


Fig. 3.126 Schematic diagram showing ball bearings arranged with oil rings in the housing (top) and for grease lubrication (bottom).
Courtesy McGraw Hill.

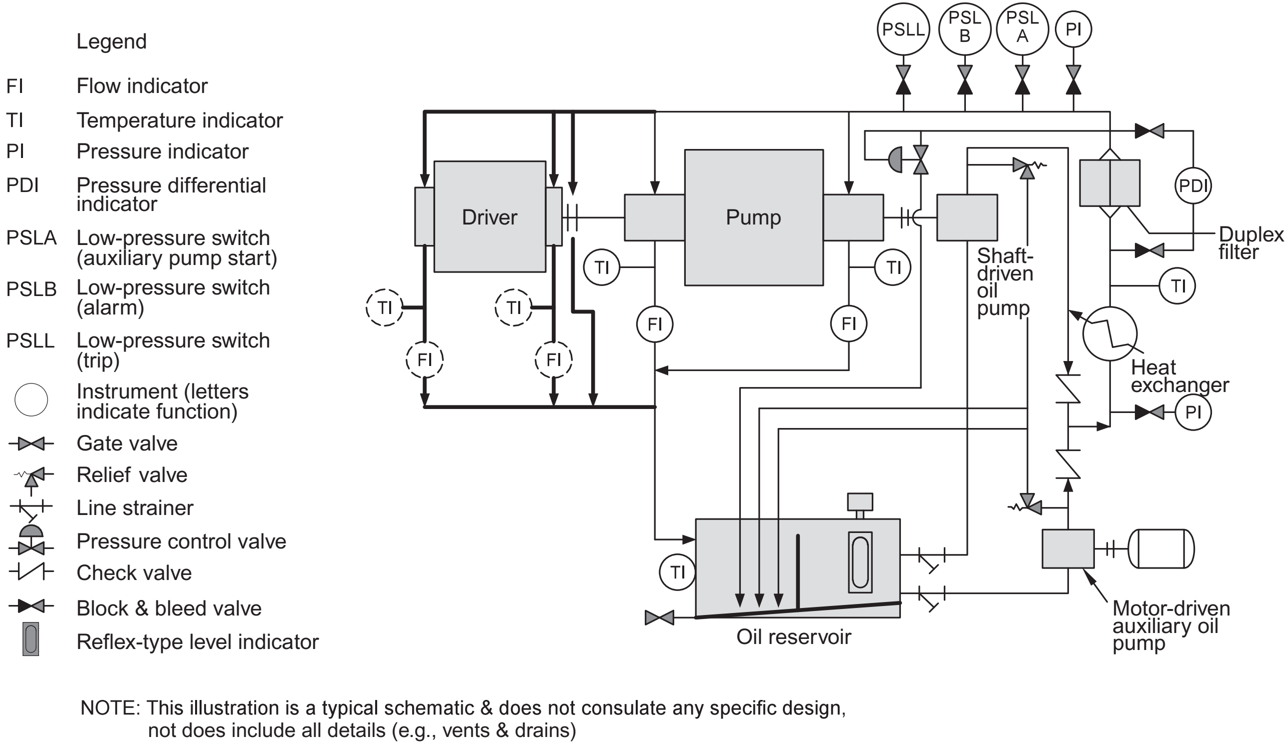


Fig. 3.127 Schematic flow diagram of a typical pressurized lube oil system.

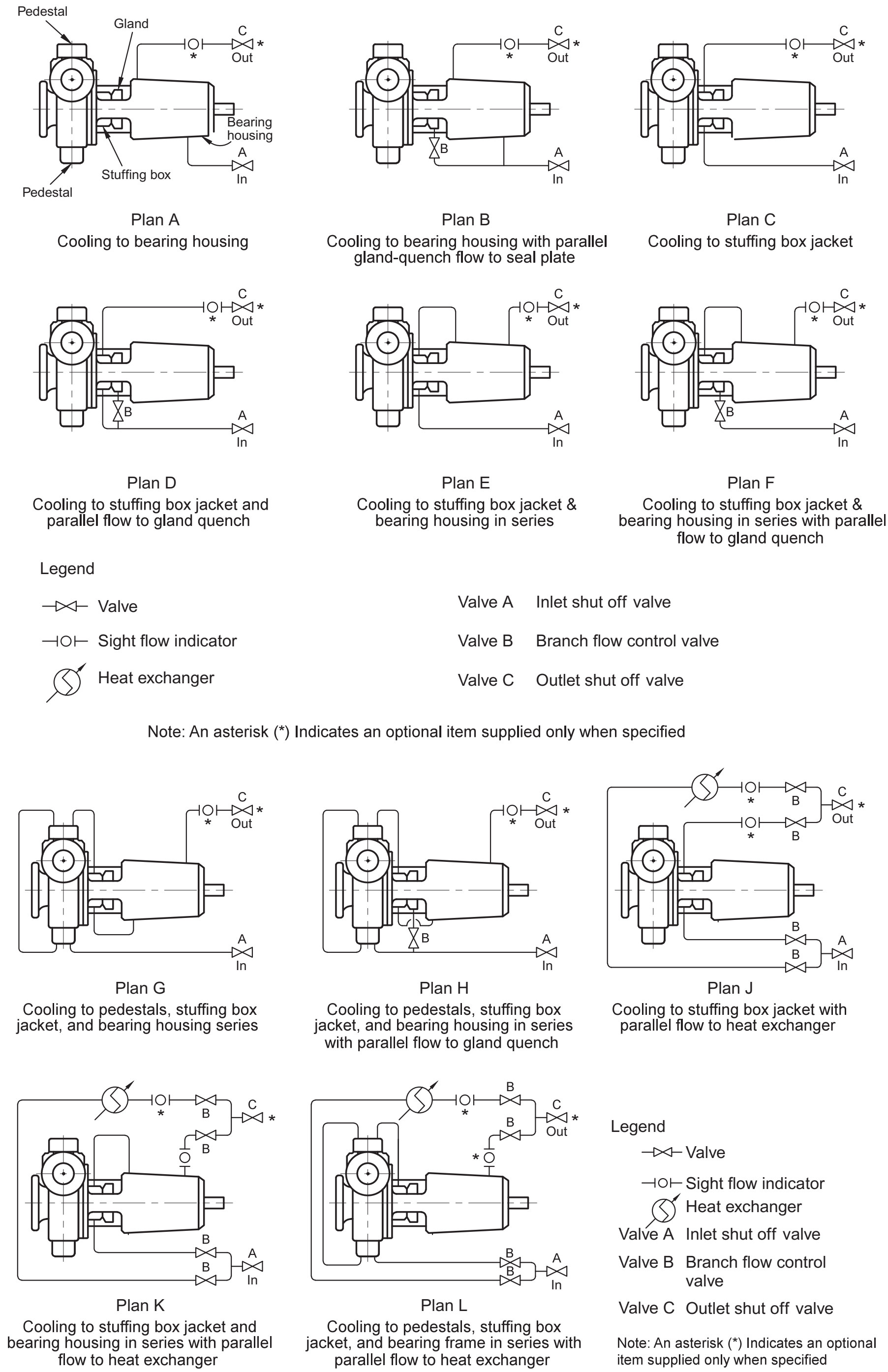


Fig. 3.128 (1 of 2) Schematic flow diagrams of typical cooling-water piping for overhung-impeller pumps. Courtesy of API. (2 of 2) Schematic flow diagrams of typical cooling-water piping for overhung-impeller pumps.
Courtesy of API.

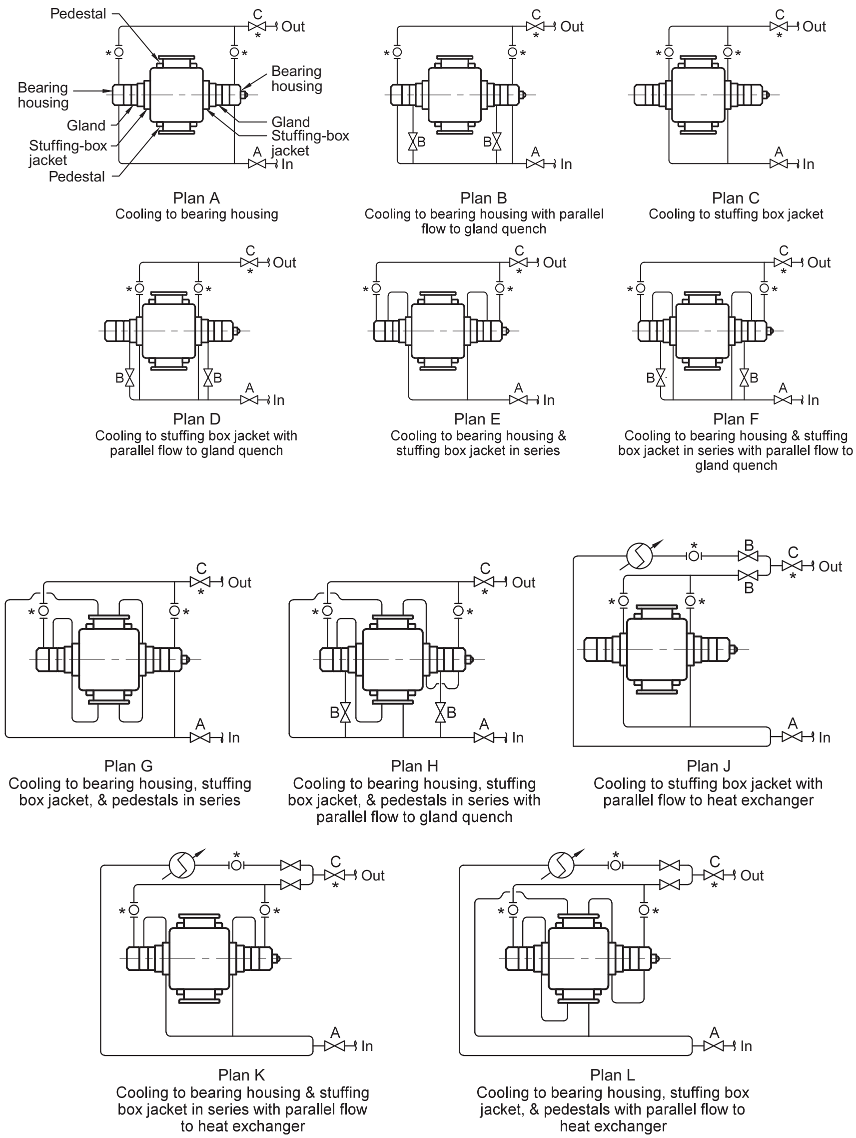


Fig. 3.129 (1 of 2) Schematic flow diagrams of typical cooling-water piping for between-bearings pumps. Courtesy of API. (2 of 2) Schematic flow diagrams of typical cooling-water piping for between-bearings pumps.
Courtesy of API.

Some “rules of thumb” when installing cooling water systems for stuffing box and bearing housings are as follows:

- Pressurized oil system—maintain temperature below 160°F (71°C)
- Nonpressurized oil system—maintain temperature below 180°F (82°C)
- Stuffing box with packing—maintain temperature below 200°F (93°C)
- Stuffing box with mechanical seals—maintain temperature below 180°F (82°C) unless seal materials indicate higher operating temperatures are allowed

3.11.8 Couplings

Centrifugal pumps are connected to their drivers through couplings, except close-coupled units, in which case the impeller is mounted on an extension of the shaft of the driver. The coupling connects the driver to the pump shaft and is also somewhat forgiving as to misalignment. The purpose of a coupling is to:

- transmit power (horsepower) from the driver shaft to the pump shaft
- compensate for small amounts of misalignment
- dampen axial thrust and torsional vibration
- provide enough clearance between the two shaft ends to facilitate maintenance

Couplings can be either rigid or flexible.

3.11.8.1 Rigid couplings

A rigid coupling is one that does not allow axial or radial motion between the driving and driven shafts. As shown in Fig. 3.130, a rigid coupling consists of two flanges, one mounted on each shaft. Rigid couplings require exact alignment between shafts

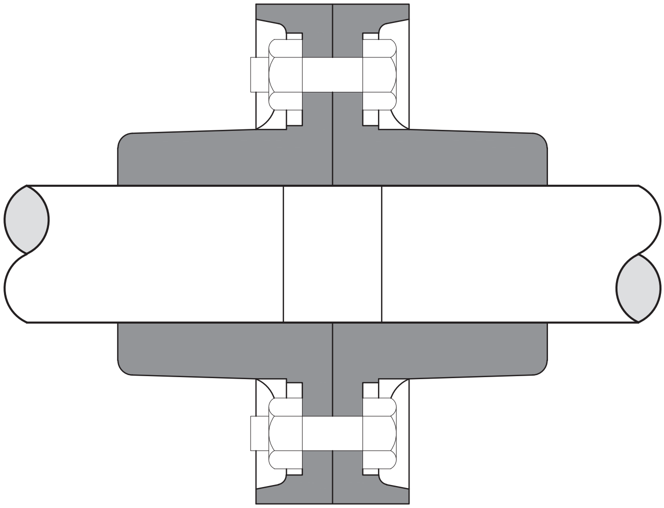


Fig. 3.130 Schematic of a typical rigid coupling.

otherwise considerable wear on the bearings and seals will occur. Rigid couplings are primarily used in vertical pumps.

3.11.8.2 Flexible couplings

Couplings are highly stressed moving parts. Proper alignment is essential as misalignment adds to the cylindrical stresses. Excessive misalignment will cause fatigue or wear-related failures dependent upon the degree of misalignment. The flexible coupling connects the two shafts but is capable of transmitting torque from the driving to the driven shaft while allowing for misalignment.

Couplings are rated for a maximum amount of misalignment, measured in degrees. The greater the distance between the hubs, the greater the measured misalignment can be without exceeding the maximum limit. This feature allows alignment easier and reduces the chance of vibration or other alignment-related pump problems.

Most couplings connecting the pump shaft and driver shaft are usually flexible. Flexible couplings consist of a flexible element, which:

- is capable of handling misalignment and absorbing torque fluctuations
- require no lubrication
- tolerate more misalignment
- usually cost less
- have a lower failure rate
- should not be applied in high horsepower or corrosive applications

Most couplings for horizontal pumps are flexible disc type or gear type.

3.11.8.2.1 Flexible disc type

Industry experience indicates the primary cause of coupling failures is lack of lubrication. The flexible disc-type (Fig. 3.131) coupling is preferred because it does not require lubrication. Flexible disc-type couplings should have stainless steel discs to resist corrosion. Spacers are recommended to allow less stringent alignment tolerances and to facilitate maintenance. One disadvantage of older flexible disc-type couplings is the danger posed from flying debris when a failure occurs. Newer designs prevent the danger of flying debris in the event of a failure. Fig. 3.132 shows a sectional view of the Thomas 71 flexible disc-type coupling.

3.11.8.2.2 Spring grid type

The spring-grid-type coupling (Fig. 3.133) consists of two steel flanged steel hubs, a special tempered steel spring forming a complete cylindrical grid, and a steel shell cover. The steel spring is interwoven between groves machined on the hub mounted on each shaft. The peripheries of the hubs are slotted to receive the spring member. This type of coupling is much more limited in their allowable misalignment than are the disc-type couplings and thus is not preferred. The coupling must be lubricated with grease to prevent excessive wear of the grid. They are available for horsepower applications up to 200 hp.

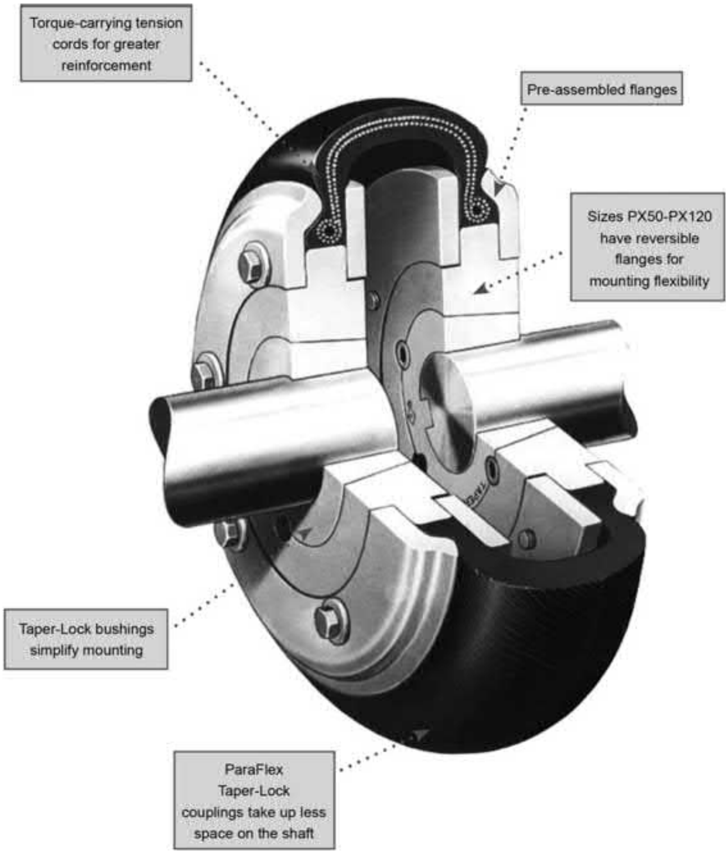


Fig. 3.131 ParaFlex's flexible disk-type "Taper-Lock" coupling.
Courtesy of ParaFlex.

3.11.8.2.3 Gear type

Figs. 3.134 and 3.135 show a gear-type flexible coupling. In the outer shell of the coupling, inside each end, a ring of internally cut gear teeth meshes with the gears on the driving and driven halves of the coupling. The torque is transmitted through the gear teeth, whereas the necessary sliding action and ability for slight adjustments in position comes from a certain freedom of action provided between the two sets of teeth. To prevent any tendency to bind because of friction, the gears work in a constant bath of oil that is retained within the outer shell. Some high-speed applications use light grease.

3.11.8.2.4 Limited-end-float type

Limited-end-float-type (LEF) (Fig. 3.136) couplings are used where axial movement must be limited. It is predominately used where end float must be restricted and maintained. For example, in sleeve bearings and rotor systems where thrust is not

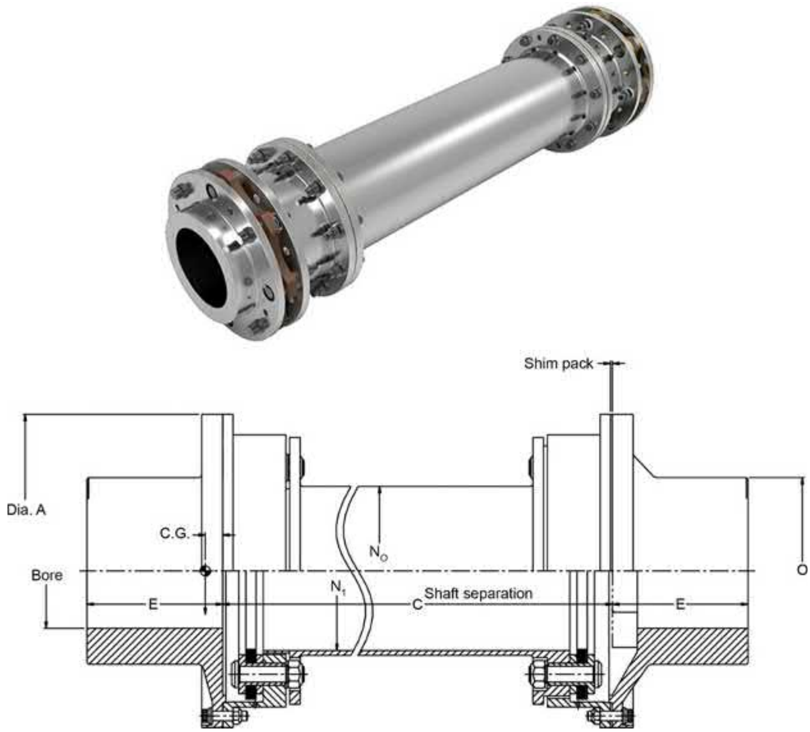
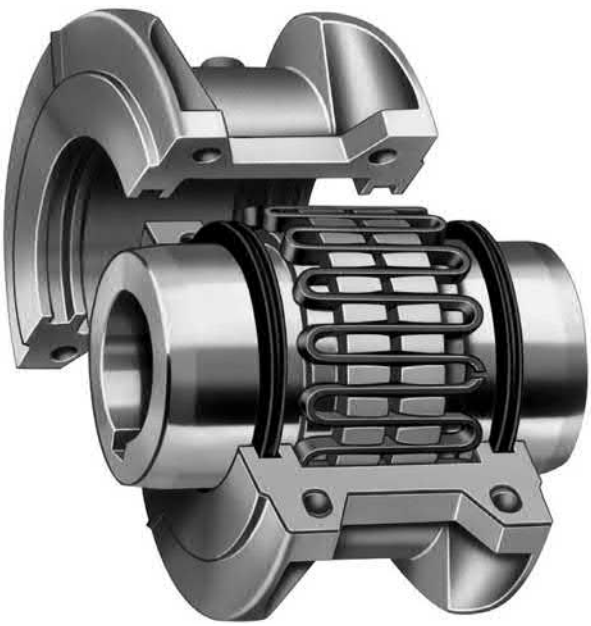


Fig. 3.132 Kop-flex high performance disk-type coupling: (top): assembled coupling; (bottom): sectional view.
Courtesy of Kop-Flex.

Fig. 3.133 Kop-Flex flexible spring-grid-type coupling.
Courtesy of Kop-Flex Couplings.



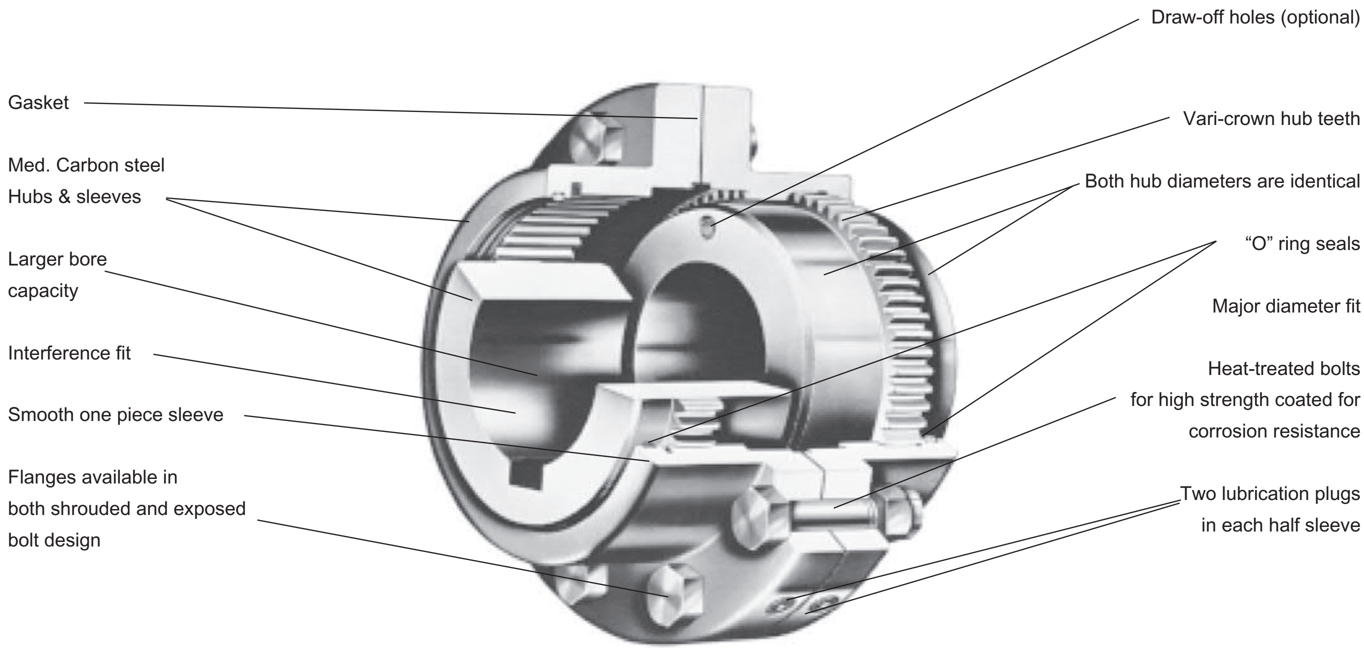


Fig. 3.134 Cutaway diagram of an Emerson Grease-Packed Tooth coupling. Courtesy of Emerson Transmission Company.

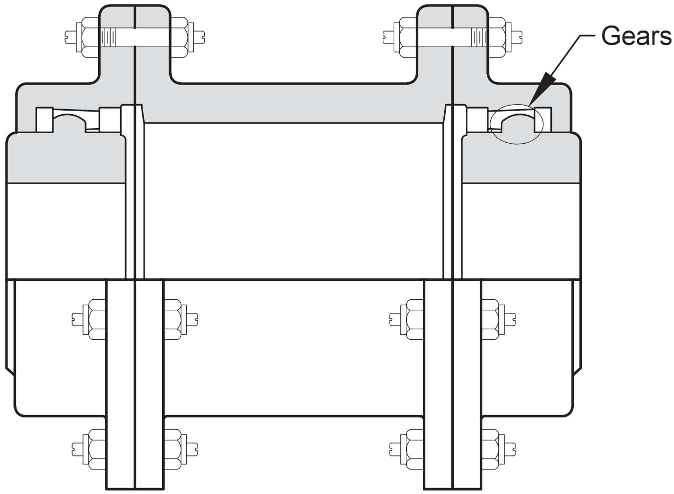


Fig. 3.135 Cutaway of a Kop-Flex Continuously Lubricating gear-type coupling (top): assembled; (bottom): sectional view.
Courtesy of Kop-Flex.

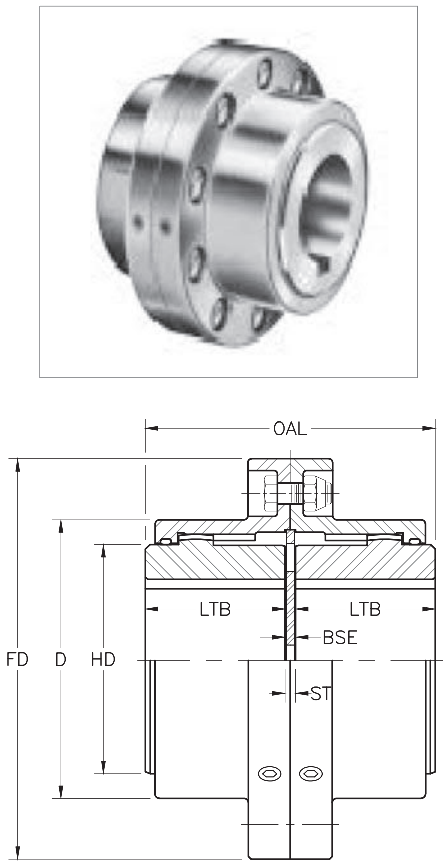


Fig. 3.136 Schematic of Lovejoy's Limited-End-Float Type Coupling.
Courtesy of Lovejoy Company.

permitted, the coupling must maintain the position of driving units and driven units. The LEF couplings are required with large motor (>250 HP at 3600 rpm) in which the sleeve bearings are designed to permit the rotor to move axially ¼–½-in. The LEF coupling keeps the motor sleeve bearings within their axial limits and lets any electrical thrust from the motor transmit to the pump thrust bearing.

3.11.8.2.5 Coupling spacers

Spacer-type couplings consist of an additional length of shaft that is installed between the coupling and pump drive shaft and/or coupling and driver so that the bearings and seal can be removed without moving the driver or pump from its foundation. The coupling spacer should be at least 2 in. longer than the bearing assembly or seal assembly that is to be removed. The coupling spacer should normally be specified as it allows the removal of a mechanical seal or the disassembly of a pump without disturbing the alignment of the pump and driver. Industry practice is to specify a coupling spacer and let the pump manufacturer be responsible for the correct length. The spacer-type coupling has the advantage that it allows greater actual misalignment between the pump and driver shafts. Coupling spacers are recommended for all flexible coupling applications.

3.11.8.3 Coupling guards

Couplings should be protected by a coupling guard. Guard covers should cover the rotating parts to within ½ in. of the pump and driver housings.

3.12 Axial thrust

3.12.1 Single-stage pumps

A single-suction, closed or semiopen impeller is inherently unbalanced and subject to continual end thrust. Thrust is directed axially toward the suction because of the low pressures that exist in the impeller eye during pump operation. Axial loading is the net resultant of the following three forces (Fig. 3.137):

- *Force 1*: Caused by the discharge pressure that is located in the clearance spaces behind the impeller which tends to push the shaft toward the suction end
- *Force 2*: Is a result of the discharge pressure on the other side of the impeller that tends to push the shaft away from the suction end
- *Force 3*: Is a result of the low suction pressure acting over the impeller eye area that tends to push the impeller away from the suction end

The earlier pressures are not uniform; are higher at the periphery and lower at the center; and vary with specific gravity, clearance, and impeller design. The larger the TDH and the larger the impeller eye diameter, the larger the thrust. Excessive thrust may result in bearing and seal damage.

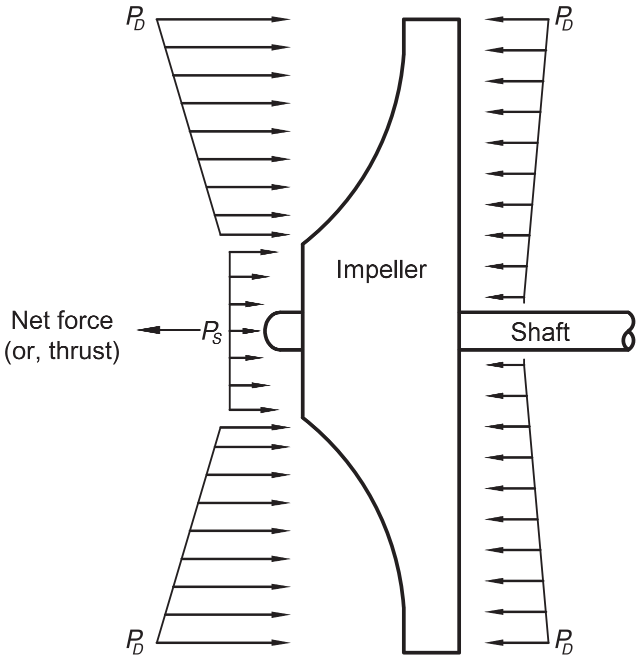


Fig. 3.137 Axial loading for closed and semiopen single-suction impellers.

Theoretically, a double-suction impeller is in hydraulic axial balance with the pressures on one side equal to and counterbalancing the pressures on the other (Fig. 3.138). In practice, this balance may not be achieved 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 being too close to the pump suction nozzle, may cause unequal flows to the suction eyes.

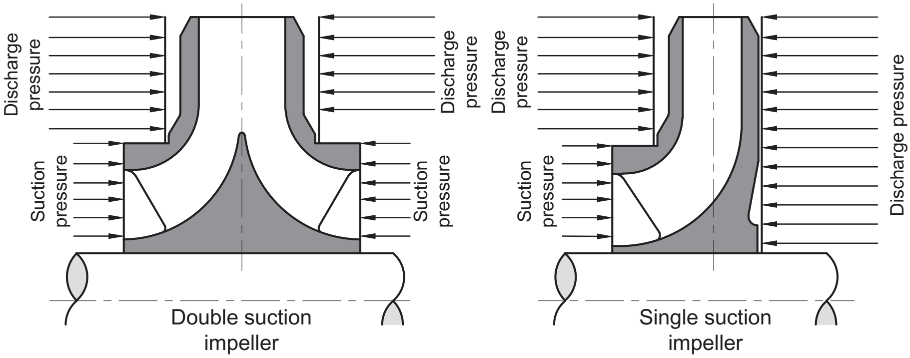


Fig. 3.138 Orgin of pressure acting on impeller shrouds to produce axial thrust.

(3) The two sides of the discharge casing may not be symmetrical or the impeller may be located off-center. These conditions will alter the flow characteristics between the impeller shrouds and casing, causing unequal pressures on the shrouds.
(4) Unequal leakage through the two leakage joints will tend to upset the balance.

Combined, the earlier factors create definite axial unbalance. To compensate for this, all centrifugal pumps, even those with double-suction impellers, incorporate thrust bearings.

The single-suction radial flow impeller with the shaft passing through the impeller eye (Fig. 3.138) is subject to axial thrust because a portion of the front wall is exposed to suction pressure, thus exposing relatively more backwall surface to discharge pressure. If the discharge chamber pressure were uniform over the entire impeller surface, the axial force acting toward the suction would be equal to the product of the net pressure generated by the impeller and the unbalanced annular area.

Pressure on the two single-suction impeller walls is not uniform. The liquid trapped between the impeller shrouds and casing walls is in rotation, and the pressure at the impeller peripheral is higher than at the impeller hub. Although one need not be concerned with the theoretical calculations for this pressure variation. Fig. 3.139 describes it qualitatively.

To eliminate the axial thrust of a single-suction impeller, a pump can be provided with both front and back wear rings. To equalize thrust areas, the inner diameter of both rings is made the same (Fig. 3.140). Pressure approximately equal to the suction pressure that is maintained in a chamber located on the impeller side of the back wear ring is returned into the suction area through these holes.

In large single-stage single-suction pumps, balancing holes are considered undesirable because leakage back to the impeller suction opposes the main flow, creating disturbances. In these pumps, a piped connection to the suction replaces the balancing holes.

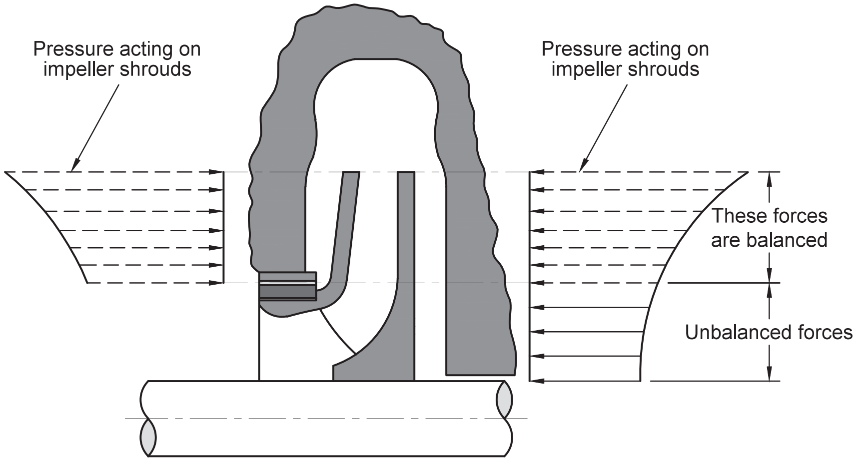


Fig. 3.139 Actual pressure distribution on front and back shrouds of single-suction impeller with shaft through impeller eye.

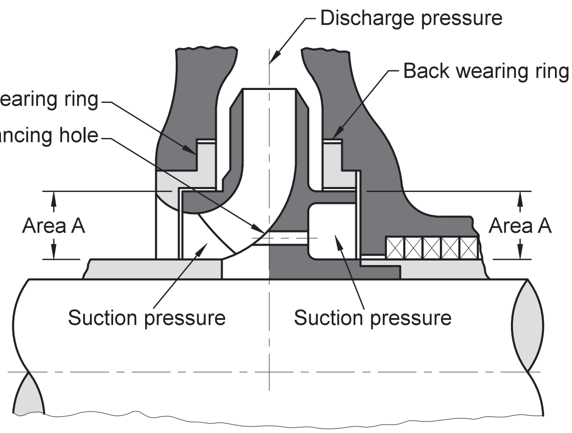


Fig. 3.140 Balancing axial thrust of a single-suction impeller with wear ring on the back and balancing holes.

3.12.2 Multistage pumps

Multistage pumps are relatively low capacity when compared to the entire range covered by centrifugal pumps. Therefore it is seldom necessary to use double-suction impellers just to reduce the NPSH required for a given capacity. Even if a double-suction impeller is desirable for the first stage of a large capacity multistage pump, it is not necessary for the remaining stages. As to the advantage of the axial balance it provides, it must be considered that a certain amount of axial thrust is actually present in all centrifugal pumps and the necessity of a thrust bearing is therefore not eliminated.

Most multistage pumps are built with single-suction impellers. Two single-suction impeller arrangements used in multistage pump are as follows:

(1) Several single-suction impellers may be mounted on one shaft, each having its suction inlet facing in the same direction and its stages following one another in ascending order of pressure (Fig. 3.141). Axial thrust is often balanced by a hydraulic balancing drum (see later in this chapter).
(2) An even number of single-suction impellers can be mounted on one shaft, one half of these facing in an opposite direction to the second half. With this arrangement, axial thrust on the one half is compensated by the thrust in the opposite direction on the other half (Fig. 3.142). This mounting of single-suction impellers back to back is called "opposed impellers."

The "opposed impeller" arrangement balances axial thrust only if the following conditions are maintained:

(1) The pump must be provided with two stuffing boxes.
(2) The shaft must have a constant diameter.
(3) The impeller hubs must not extend through the interstage portion of the casing separating adjacent stages.

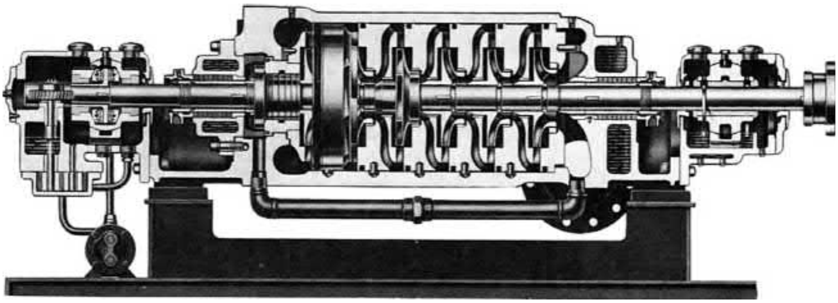


Fig. 3.141 Multistage pump with single-suction impellers facing in one direction and hydraulic balancing device.

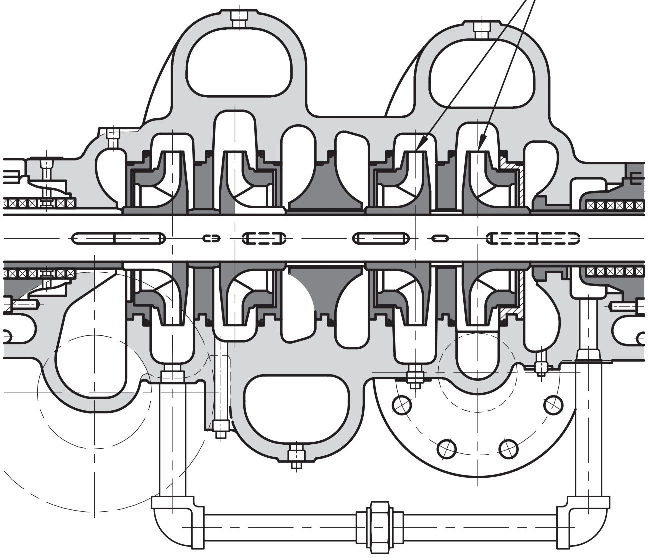


Fig. 3.142 Four-stage pump with opposed impellers.

Most multistage pumps fulfill the first condition. Due to structural requirements, the last two conditions are not practical. A slight residual thrust is usually present in multistage opposed-impeller pumps, unless impeller hubs or wear rings are located on different diameters for various stages. Since such construction would eliminate axial thrust only at the expense of reduced interchangeability and increased manufacturing cost, this residual thrust, being relatively small, is usually carried on the thrust bearing.

If less than design flows are expected for extended periods, it may be necessary to specify heavy-duty duplex 40° contact angle thrust bearings, mounted back to back, instead of the standard double-row thrust bearings (refer to Fig. 3.143).

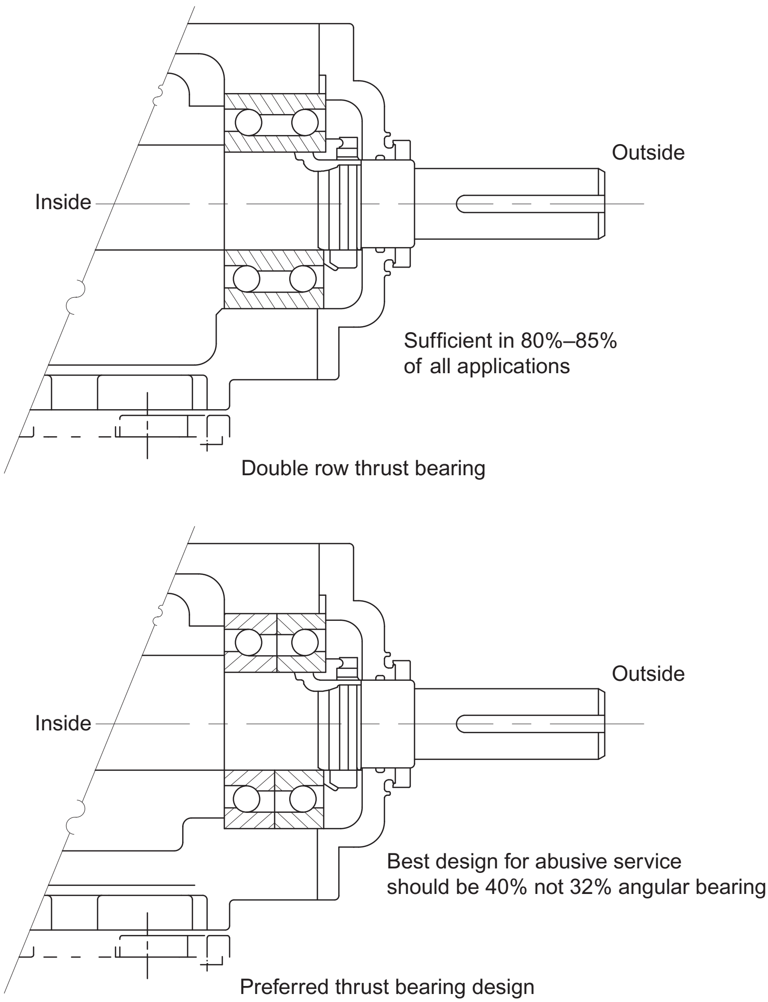


Fig. 3.143 Thrust bearing design: (top) double row thrust bearing; (bottom) preferred thrust bearing design.

3.13 Radial thrust

3.13.1 Single-volute pumps

In a single-volute pump casing design (Fig. 3.144), uniform or near-uniform pressures act on the impeller when the pump is operated at the best efficiency point (BEP). At flow rates other than the BEP (Fig. 3.145), the pressures around the impeller are not uniform, and there is a resultant radial reaction (F). A graphical representation of the typical change in this force with pump capacity is shown in Fig. 3.146; note that the

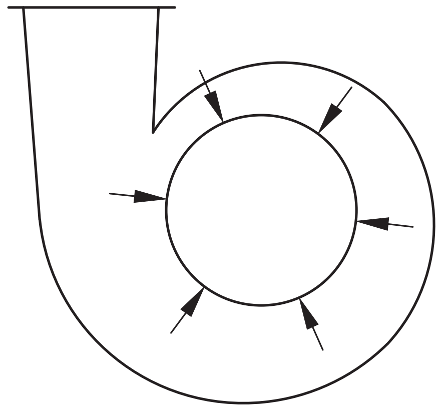


Fig. 3.144 Zero radial reaction in a single-volute casing (uniform pressures exist at design capacity).

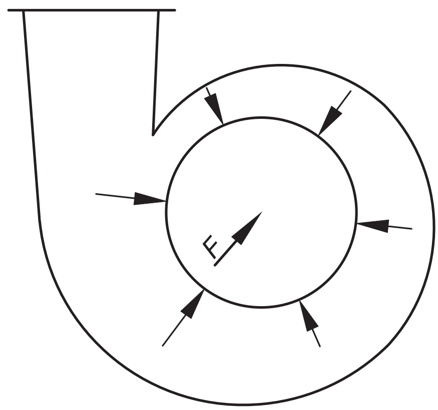


Fig. 3.145 Radial reaction in a single-volute casing (uniform pressures do not exist at reduced capacities).

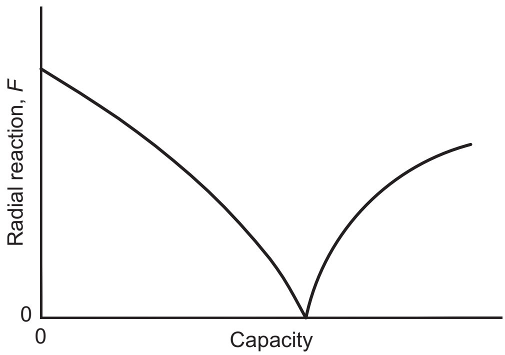


Fig. 3.146 Magnitude of radial reaction in single-volute casing.

force is greatest at shutoff. "F" decreases from shutoff to design flow rate and then increases with as flow rates increase. As flow rates increase above the design flow rate the reaction is roughly in the opposite direction from that with less than design flow rate.

The radial reaction force is a function of total head, and the width and diameter of the impeller. Thus a high head pump with a large diameter impeller will have a much greater radial reaction force at partial capacities than a low head pump with a small-diameter impeller. A zero radial reaction is not often realized; the minimum reaction occurs at design capacity. In a diffuser-type pump, which has the same tendency for overcapacity unbalance as a single-volute pump, the reaction is limited to a small area repeated all around the impeller, with the individual forces canceling each other.

Shaft diameter and bearing size can be affected by allowable deflection as determined by:

- Shaft span
- Impeller weight
- Radial reaction forces
- Torque to be transmitted

As the fluid leaves the top of the rotating impeller, it exerts an equal and opposite force on the impeller, shaft, and radial bearings. At BEP, the summation of all radial forces nearly cancels each other out. At capacities below or above BEP, forces do not cancel out completely because the flow is no longer uniform around the periphery of the impeller. Radial forces can be significant especially for standard volute pumps (Fig. 3.147).

In the past, when flow rates exceeded 50% of the design flow rate, standard designs compensated for reaction forces by using maximum diameter pump impellers. When flow rates are to be operated at flow rates less then design the pump manufacturer would supply a heavier shaft. However, sustained operation at less than design flow

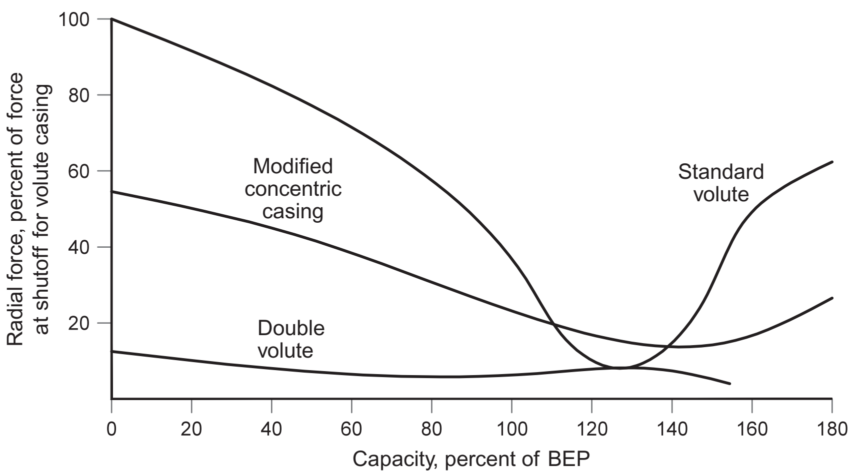


Fig. 3.147 Radial forces acting on pump impellers as a function of casing design and flow percentage of BEP.

is much more common practice today. This results in broken shafts, especially on high-head units.

Pumps are increasingly being operated at reduced flow rates and it has become desirable to design standard units to accommodate such conditions. One solution is to use heavier shafts and heavy-duty bearings when a pump operates significantly away from the BEP and it is not possible to change the casing design (e.g., to a double volute).

3.13.2 Double volute

Fig. 3.148 is a schematic of a double-volute design. This design consists of two 180-degree volutes; a passage external to the second joins the two into a common discharge. A pressure unbalance exists at partial capacity through each 180-degree arc, forces F_1 and F_2 are approximately equal and opposite, thereby producing little, if any, radial force on the shaft and bearings. In large-capacity medium- and high-head single-stage vertical pump applications, the rib forming the second volute and separating it from the discharge passageway of the first volute strengthens the casing (Fig. 3.149). A typical twin volute is shown in Fig. 3.150.

3.14 Materials of construction

Proper material selection is highly dependent on specific service conditions and the liquids handled and will not be discussed in detail in this section. Some of the service conditions that affect the selection of materials are as follows:

- Corrosion resistance
- Electrochemical action
- Abrasiveness of suspended solids

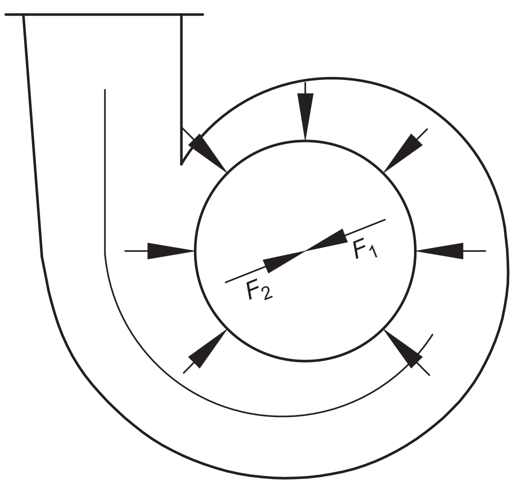
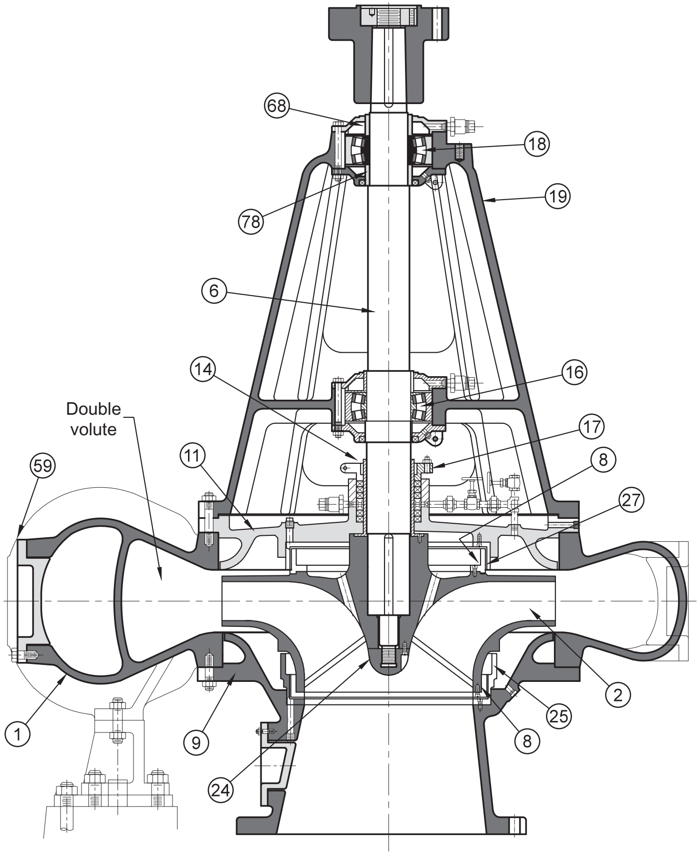


Fig. 3.148 Radial reactions in a double-volute pump.



Item no.	Name of the part	Item no.	Name of the part
1	Casing	33	Bearing housing (outboard)
1A	Casing (lower half)	35	Bearing cover (inboard)
1 B	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	Sudion cover	46	Coupling key
11	Stuffing box cover	48	Coupling bushing
13	Pocking	50	Coupling lock nut
14	Shaft sleeve	52	Coupling pin
15	Discharge bowl	59	Hondhole 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	Seal cage	125	Grease (oil) cup
31	Bearing housing (inboard)	127	Seal piping (tubing)
32	Impeller key		

Fig. 3.149 (1 of 2) Cross-sectional view of a vertical-shaft end-suction pump with a double-volute casing. Courtesy of Hydraulic Institute. (Numbers refer to parts listed in Fig. 3.149. (2 of 2) Centrifugal pump part identification. Courtesy of Hydraulic Institute.

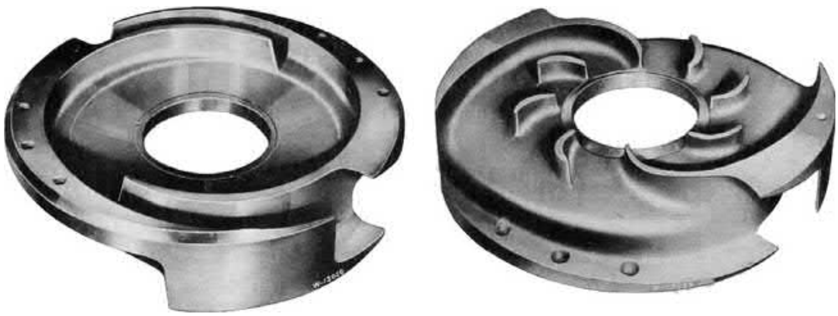


Fig. 3.150 A twin-volute of a multistage pump.

- Pumping temperature
- Head per stage
- Operating pressure
- Suitability of the material from a structural perspective
- Expected life and load factor

A listing of materials commonly recommended for various liquids can be found in:

- API 610 Standards
- Hydraulic Institute Standards
- Manufacturer/Company Standards and Specifications

Pump components can be made from any material that can be cast, molded, welded, or shaped. The suitability of the material rests in its compatibility with the pumped liquid or in some cases, the environment. Common materials used in pumps are as follows:

- *Cast iron*: This is relatively easy to cast and it holds up well in water and noncorrosive services. It has low tensile strength and becomes brittle below about 20°F (−7°C). Cast iron is not readily repairable.
- *Bronze and brass*: Easy to cast, zincless bronze holds up well in saltwater service. These materials can be repaired by brazing.
- *Carbon steel*: Easily cast and weld repairable. Suitable for temperatures as low as −40°F (4.5°C).
- *400 series stainless steel*: A little more difficult to cast than carbon steel. This material can be weld repaired with proper post weld heat treatment.
- *300 series stainless steel*: More difficult to cast than 400 series stainless steel. Can be weld repaired. When using 300 series stainless steel, it is best to specify "L" grade which is low carbon content. Regular 300 stainless steel has about 0.08% carbon content, whereas the "L" grade has a maximum of 0.03% carbon. This makes weld repair more feasible with less post weld heat treatment.
- *Special purpose alloys, such as Monel or titanium*: Some are easily cast or fabricated and are repairable. Costs vary widely, but are justified when special metallurgy is required.
- *Fiberglass reinforced plastics (FRP)*: Easily cast and machined. These materials have excellent corrosion resistance to most aggressive chemicals. If the pump manufacturer knows what is being pumped, they will select the proper grade of plastic for the application.

Unless the pumped fluid is exotic, or some special characteristics are expected, the materials most commonly used for specific parts of the pump are listed as follows:

3.14.1 Casing materials

At normal temperatures and pressures, centrifugal pump casings are usually made of cast iron. Cast iron casings are seldom used for pressures over 1100 psi and temperatures over 350°F (177°C). If higher pressures or higher temperatures are expected, due to higher than design speeds or higher suction pressures, the designer must either modify the design strength or another metal (cast steel) substituted. If pumping fluids are at high pressure and/or high temperature the casing is usually made of steel so as to match the piping materials. Alloyed cast iron casings are commonly used, to prevent low temperature brittle fracture, when pumping liquids at low temperatures. Bronze casings are typically used when handling mildly corrosive fluids, for example, seawater. Stainless steel is used if the pumped liquid is corrosive or highly abrasive.

3.14.2 Impeller materials

At normal temperatures and pressures, centrifugal pump impellers are usually made of bronze when handling normal liquids for the following reasons:

- Easy to cast for complicated cored sections
- Easy to machine
- Does not rust
- Provides smoother surfaces

Since pumps are normally assembled at ambient temperatures, the radial clearance between the hub of a bronze impeller and the steel shaft will increase (bronze expands 40% more than steel) when handling liquids at higher temperatures. The increased clearance can loosen the impeller on the shaft and increases the possibility of leakage and erosion. Thus bronze impellers are seldom used for temperatures above 250°F (121°C).

Bronze impellers are also limited by the pump speed. As the pump speed increases the impeller stretches at the hub. For example, a pump handling liquids at ambient temperatures with a 12-in. (300 mm) bronze or cast iron impeller mounted on a 3-in. (75 mm) shaft operating at 3600 rpm will stretch approximately 0.0011 in. If the temperature increases to 250°F (121°C), the bronze impeller will undergo an additional expansion of 0.0014 for a total excessive looseness of 0.0025 in. between the shaft and the impeller. To avoid the cumulative effect of thermal and centrifugal expansion, the speed of a bronze impeller handling hot liquids is 160 fpm or a head of 375 ft per stage.

3.14.3 Wear ring, shaft, shaft sleeve, and packing gland materials

Wear rings are usually made of bronze for the same reasons that impellers are. Sometimes the hardness or other properties require the wear ring be made out of another material, for example, cast iron, cast steel, stainless steel, or Monel.

Shafts without shaft sleeves are usually made of stainless steel, phosphor bronze, or Monel, depending on the liquid being handled. Shafts with shaft sleeves are usually made of steel or, if high stresses are to be encountered, high strength alloy steels. If corrosive liquids are handled, seepage may occur through the joint between the impeller and the sleeves, thus requiring shafts of a noncorrosive metal, for example, stainless steel, phosphor bronze, or Monel.

Shaft sleeves are usually made of bronze. If bronze is not satisfactory due to its relatively low abrasion resistance, stainless steel is substituted. In recent years, stainless steel alloys have superseded steel due to its popularity and reduced cost.

Packing glands are normally made of bronze but cast iron or steel may be used in all iron fitted pumps. If pumps handle hydrocarbons, iron or steel packing glands are brushed with bronze to avoid the potential of sparking which could ignite flammable vapors.

When using packing, gland studs and nuts should be 300 series stainless steel, especially when pumping water. Plain carbon steel studs and nuts will rust and eventually seize, preventing adjustment to the packing. Operators will typically use whatever force is necessary to turn the seized nut and move the gland. Typically the stud snaps and the pump must be taken out of service to have the stud removed, the hole retapped, and a new carbon steel stud inserted so the process can be repeated in 5 or 6 months. After a few of these futile exercises, someone will discover 316 stainless steel as a solution. As a general rule, mechanical seals have stainless steel gland studs and nuts as standard material.

3.14.4 Fitting materials

"Fittings" refer to the general construction features of pump parts (Table 3.13). Table 3.14 lists materials commonly used for various pump parts shown in Figs. 3.151–3.154.

3.14.5 Piping considerations

Auxiliary piping such as vent, drain, and seal-flush piping cannot be seal welded to seal the threads. The welding will introduce severe stress concentrations at all locations where the weld toe intersects the root of the thread. The situation is further aggravated by the fact that the threads are cut into the piping components, resulting in

Table 3.13 Materials commonly used in pump designs

Metallic	
Ferrous	Cast iron Ductile iron Carbon steel
Nonferrous (nickel alloy)	Ni-resist 304 SS 316 SS
Nonferrous	Bronze, brass
Nonmetallic	
Synthetic rubber	Neoprene Butyl
Plastics	Fluorocarbons

Table 3.14 Materials for various fittings

Part	Part no.[a]	Standard fitting	All iron fitting	All bronze fitting
Casing	1	Cast iron	Cast iron	Bronze
Suction head	35	Cast iron	Cast iron	Bronze
Impeller	4	Bronze	Cast iron	Bronze
Impeller ring	12	Bronze	Cast iron or steel	Bronze
Casing ring	3	Bronze	Cast iron	Bronze
Diffuser		Cast iron or bronze	Cast iron	Bronze
Stage-piece	5	Cast iron or bronze	Cast iron	Bronze
Shaft (with sleeve)	2	Steel	Steel	Steel, bronze, or monel
Shaft (without sleeve)	2-A	Stainless steel or steel	Stainless steel or steel	Bronze or monel
Shaft sleeve	10	Bronze	Steel or stainless steel	Bronze
Gland	15	Bronze	Cast iron	Bronze

Materials for bearing housings, bearings, and other parts are not usually affected by the liquid handled.
[a]Parts in this list and in Figs. 3.151–3.154 are numbered according to a proposed standard listing suggested to the Hydraulic Institute by Charles J. Tullo, Chief Engineer, Worthington Corporation. This proposed standard gives stationary parts odd numbers and rotating parts even numbers.

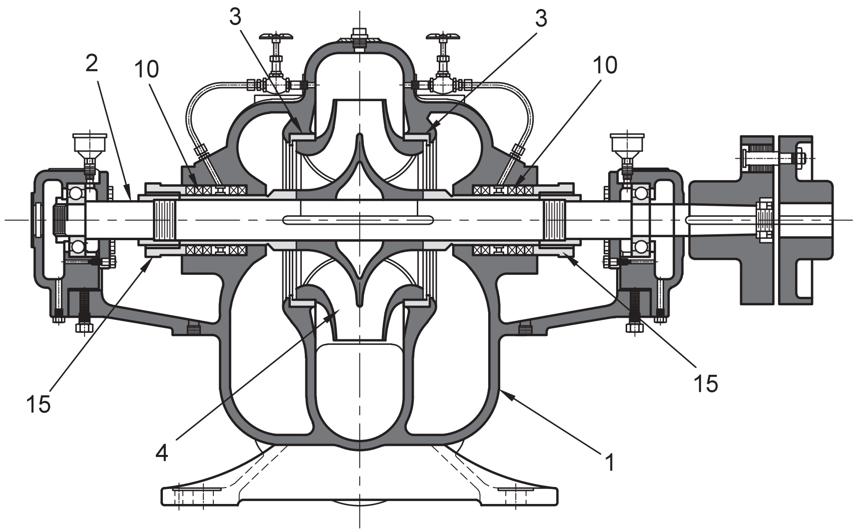


Fig. 3.151 Sechmatic of a double-suction, single-stage centrifugal pump with shaft sleeves (numbers refer to parts listed in Table 3.14).
Courtesy of Hydraulic Institute and Worthington Corporation.

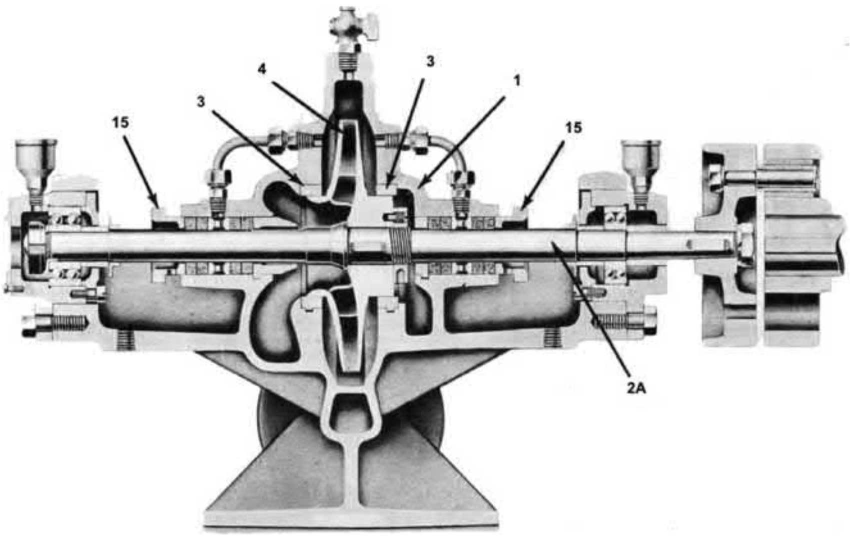


Fig. 3.152 Section of a single-suction, single-stage centrifugal pump without shaft sleeves (numbers refer to parts listed in Table 3.14).
Courtesy of Hydraulic Institute and Worthington Corporation.

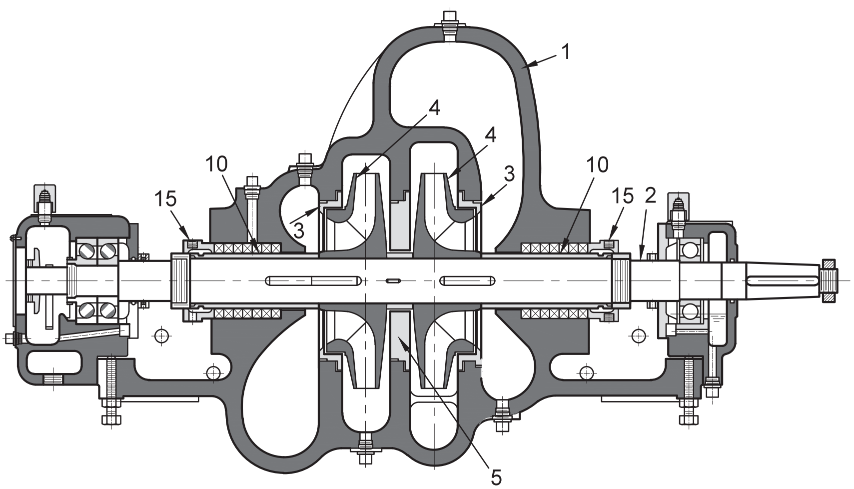


Fig. 3.153 Section of a two-stage centrifugal pump (numbers refer to parts listed in Table 3.14).
Courtesy of Hydraulic Institute and Worthington Corporation.

a reduced cross-section at the thread root. These factors when taken together result in a severe concentration in the root area.

The second reason which has been readily accepted as it applies to socket weld connections involves the fact that upon cooling, seal fillet welds need the ability to

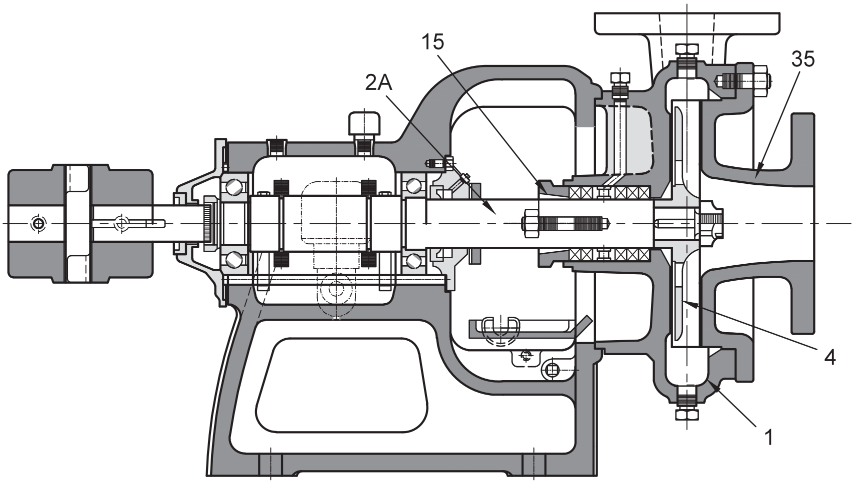


Fig. 3.154 Section of an end-suction, single-stage centrifugal pump used for light duty (numbers refer to parts listed in Table 3.14).
Courtesy of Hydraulic Institute and Worthington Corporation.

shrink due to thermal contraction. In the case of steel welded threaded connections, the threads resist contraction and another stress is introduced across the fillet weld.

If auxiliary piping is used and it must be sealed, socket welding is the only acceptable method of sealing. The proper method for socket welding the piping to the casing is to provide a 1/16-in. gap between the end of the pipe and the base of the socket in the casing. This gap provides room for the pipe to freely expand and contract during and after the welding.

The only other auxiliary connection is heavy walled tubing, which cannot be welded. Tubing is not an ideal product, as it is not rigid enough. Operations personnel typically use any convenient item handy as a ladder or step to do their work; auxiliary piping is fair game.

Flat face cast iron flanges cannot be mated to raised face flanges. Cast iron has limited tensile strength. The edge of the raised face portion of a raised face flange will act as a fulcrum point. When the flange bolts are tightened, the back side of the cast iron flange will be placed in tension and cause the flange to break. If the flange is on a pump, the casing will make a good boat anchor or paper weight.

3.15 Centrifugal pump selection criteria

3.15.1 Overview

Centrifugal pumps are well accepted in production operations. The advantages of centrifugal pumps are as follows:

- Relatively inexpensive
- Operate at high rotational speeds (900, 1200, 1800, and 3600 rpm)

- Have a smooth, nonpulsating discharge
- Have few moving parts and therefore tend to have greater on-stream availability and lower maintenance costs than positive displacement pumps
- Have relatively small space (footprint) and weight requirements in relation to throughput
- There are no close clearances in the fluid stream, therefore, they can handle liquids containing dirt, abrasives, large solids, and so on
- Because there is very little pressure drop and there are no small clearances between the suction flange and the impeller, they can operate at low suction pressures without causing damage to the pump
- Due to the shape of the head-capacity curve, centrifugal pumps automatically adjust to changes in head. Thus capacity can be controlled over a wide range at constant speed

The limitations of centrifugal pumps include:

- Have extremely low efficiencies when handling viscous fluids and are not recommended for fluids with viscosities above 100 cp (0.1 Pa s). However, sometimes there are no good alternatives, as in the case of down-hole submersible pumps handling heavy crude oils.
- Have low maximum efficiencies when compared to reciprocating pumps. Since the efficiency also declines as the flow rate varies from the design point, in actual operation the pump will operate at still lower efficiencies. Efficiencies between 55% and 75% are common, as opposed to 85%–90% for positive displacement pumps.
- Are sensitive to downstream pressure, thus, if pressure increases flow is reduced and are therefore not typically used in metering applications.
- Not self-priming, thus the suction must be flooded before starting the pump.

This section provides general information for selecting a centrifugal pump for a specific application. Selection should be based on total lifecycle cost and not just initial cost. The material contained herein provides background information on selecting a pump for upstream productions.

3.15.2 Casing duty ratings

The pressure-temperature ratings of pumps provide an index for duty rating. Fig. 3.155 is a decision flowchart that assists in determining whether a particular pumping application should be classified as medium or heavy duty.

Light duty: The characteristics of light duty pumps are not well standardized nor clearly defined. They are used in domestic and intermittent industrial use.

Medium duty: Medium duty pumps have the following characteristics:

- Operating pressures less than 300 psig
- Operating temperatures less than 300°F
- Flow rates less than 500 gpm
- Speeds less than 3600
- Based on American National Standards Institute Standards (ANSI)
- Not in a process area
- Less expensive than medium duty
- Have standardized dimensions
- Readily available

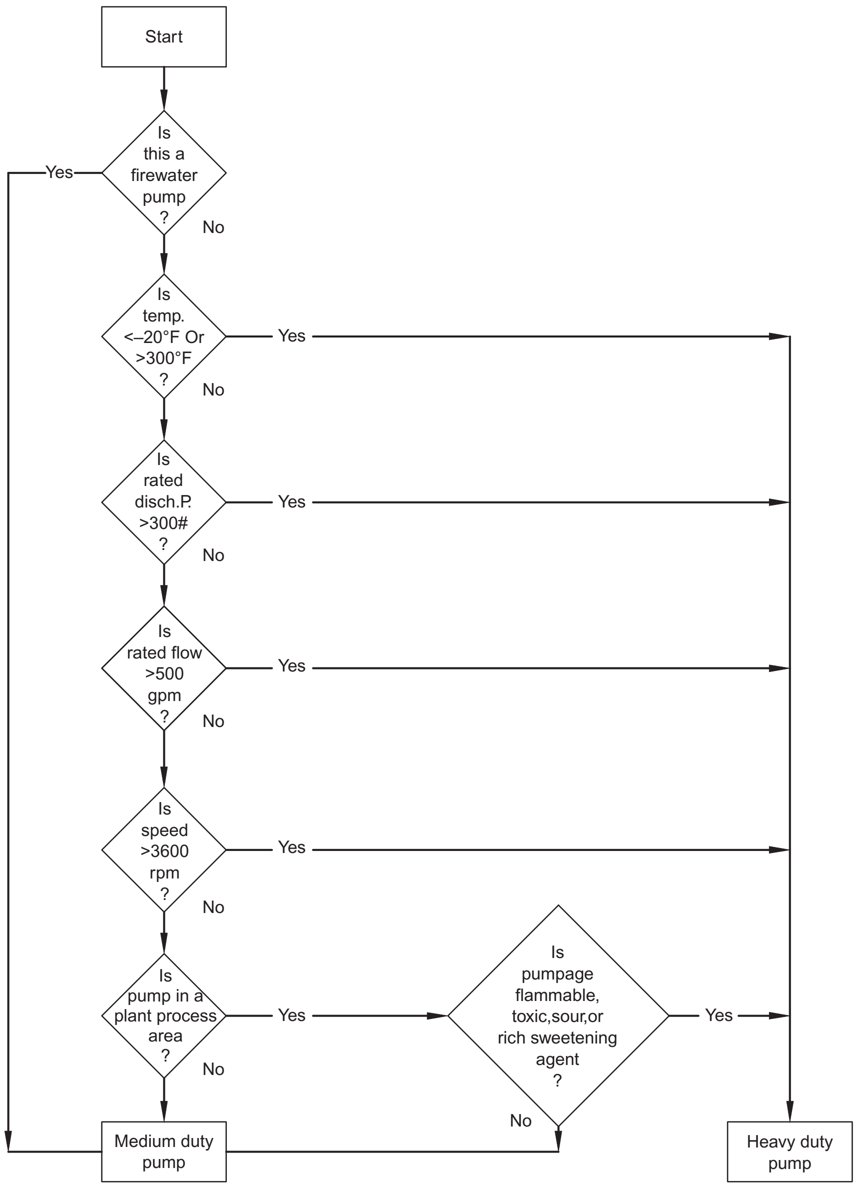


Fig. 3.155 Decision flowchart that assists in determining whether a pump is classified as medium or heavy duty.

Heavy duty: Heavy-duty pumps are pumps that operate above those of Medium Duty. They are based on American Petroleum Institute (API) Standards. They are more expensive, have longer deliveries, offer better efficiencies, more reliable, and have lower operating costs than medium duty pumps.

Industry practice is to specify any pump used in

- Heavy duty (all continuous-duty, process plant, hydrocarbon pumps) or critical service (high-pressure water-flood, etc.) meet API Standard 610.
- Light duty (smaller than 150 hp and noncritical services) meet ANSI Standards or a general-purpose pump to supplier standards.

The majority of pumping needs within a production facility are met with the "work horse" pump—a single-suction, single-stage, 3600/1800 rpm horizontal centrifugal pump. Vertical in-line single-stage centrifugal pumps are reliable and usually less expensive to purchase and install. In all cases, the designer should consider operator preference, maintenance considerations, spare part availability, and engineering judgment.

3.15.3 Pump standards

The Hydraulic Institute (HI), formed in 1917, consisted of representatives from several companies. The purpose was to solve engineering problems brought about by production problems encountered during World War One.

The American Petroleum Institute (API), formed in 1919, consisted of manufacturers and end users. API issues Standards based on the accumulated knowledge and experience of manufacturers and users in order to establish a pump with maximum reliability and efficiency at the least operational cost.

The American Voluntary Standard (AVS), formed in the 1950s, consisted of a committee of the Manufacturing Chemists Association (MCA) in association with the large pump manufacturers. The purpose was to develop a standard for the chemical process industry. In 1971, the standard was adopted by the American National Standards Association (ANSI).

There are significant construction and design differences between ANSI and API pumps. These differences will impact the pump selection. Fig. 3.156 lists the major differences between ANSI and API Standards. The two major differences are pressure rating and materials of construction. ANSI pumps are:

- limited to ASME 150 class ratings
- standard materials are cast and ductile iron
- not readily available with carbon steel casings or impellers
- not as efficient or reliable as API pumps

Limitations with use of cast or ductile iron:

- castings (case and impeller) cannot be repaired by welding
- susceptible to cracking due to thermal shock, for example, when exposed to cold extinguishing fluids it may crack. If pump is handling a flammable or combustible fluid, it could feed a fire

316 SS is commonly substituted in ANSI pumps that meet the required service conditions but where cast or ductile iron is not acceptable.

	ANSI	API
Type pump and specification	ANSI B73.1 for horizontal end suction top discharge pumps ANSI B73.2 for vertical in-line pumps All are single stage	API 610 for horizontal single and multistage pumps, vertical in-line, vertical single and multistage centrifugal pumps
Maximum allowable working pressure (MAWP)	275 PSIG	Minimum 700 PSIG Some API pumps are designed for pressures above 5000 PSIG
Hydrostatic test pressure	415 PSIG	Minimum 1050 PSIG API pump hydrostatic test pressure will be 1.5 times the MAWP
Flange rating	150# flat faced is standard. 150# raised face is available	300# raised face is standard. 600, 900, 1500, and higher ratings are available if required by the service
Maximum temperature	250°F Pump casing is foot mounted which limits allowable thermal growth	800°F pump casing is centerline mounted. No casing thermal growth limitations
Materials of construction (casing and impeller)	Ductile Iron 316 SS Alloy 20 A carbon steel casing or impeller is not commonly available	Carbon steel casing is standard; stainless steel is also available. Impeller materials are cast iron, carbon steel, and stainless steel
Maximum head differential	550–600 feet ANSI pumps are only single stage. Maximum impeller diameter is about 13 in.	Practical limit is 10,000 ft Horizontal API pumps can have as many as 14 stages
Impelller design and attachment	Open impellers are common. Some enclosed impellers are available No standard for attachment to the shaft. Most are threaded on the end of the shaft	All are enclosed design. Some open designs are available for special coke crushing services. Impellers must be key driven with a lock nut attachment
Standard dimensions	ANSI pumps are built for interchangeability between manufacturers	No standard dimensions apply
Shaft sleeves	Not required but are available. Fit to the shaft and extension past the gland are not ANSI specification requirements	Are required to prevent shaft damage in the seal or packing area. Sleeve and stuffing box design is part of the API 610 specification
Lubrication	Can be grease or oil lubricated	Oil lubrication is required. Usually ring oil system is provided
Thrust bearing and life	Antifriction bearings only. B-10 bearing life of 17,500 h at design load is required	Antifriction ball bearings must be duplex, single-row, 40-degree angular-contact type, installed back to back. L-10 bearing life must exceed 25,000 h at rated conditions, or 16,000 h at maximum axial and radial loads at rated speed
Wear rings	Not required and not available in most designs due to the use of open impellers	Case and impeller, front and back wear rings are required Wear ring clearances, attachment, and hardness differential are specified

Fig. 3.156 Comparison of ANSI and API pump standards.

3.15.4 Head-capacity considerations

As discussed earlier, the performance of a centrifugal pump over a range of heads and capacities is a function of the impeller and case design. There are three general impeller designs (Fig. 3.157): radial flow, mixed flow (Fig. 3.158), and axial flow (or propeller)(Fig. 3.159). Fig. 3.48 shows the general shape of the characteristic curves for radial-, mixed-, and axial-flow pumps. It shows the head, brake horsepower, and efficiency plotted as a percent of their values at the BEP.

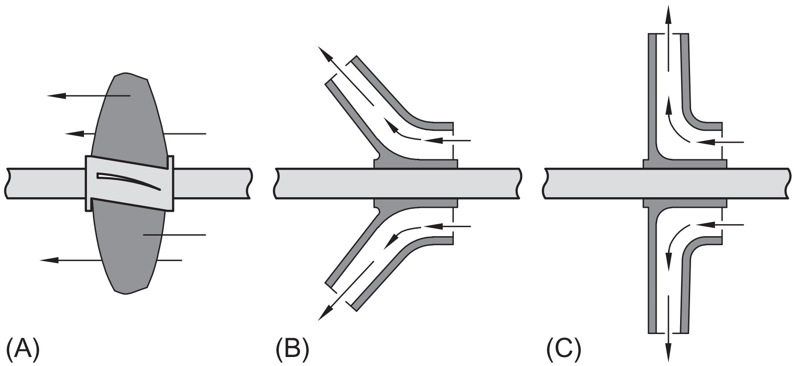


Fig. 3.157 Impeller classification: (A) axial flow; (B) mixed flow; and (C) radial flow.

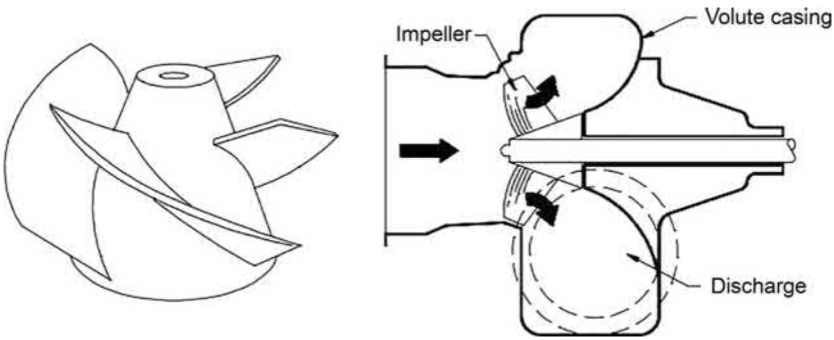


Fig. 3.158 Example of an open mixed flow impeller.
Courtesy of Worthington Pumps.

Fig. 3.159 Example of an axial flow impeller.
Courtesy of Worthington Pumps.

The head-capacity curve for a *radial-flow* centrifugal pump is relatively flat. As the head gradually decreases, the flow and brake horsepower increase. The brake horsepower reaches maximum at maximum flow.

Mixed-flow and *axial-flow* pumps exhibit different characteristics. The head curve for a mixed-flow pump is steeper than for a radial-flow pump. The shutoff head is usually 150%–200% of the design head. The brake horsepower remains fairly constant over the entire flow range. For a typical axial-flow centrifugal pump, the head and brake horsepower both increase drastically near shutoff.

Figs. 3.160 and 3.161 tabulate centrifugal pump selection guidelines.

3.15.5 High-head low-flow considerations

When an application lies outside the typical range of a single stage, for example, high-head and low-flow rate, a high-speed, single-stage vertical-in-line pump should be considered. If conditions exceed the limits of the high-speed, single-stage vertical-in-line pump, a multistage centrifugal or positive displacement pump may be appropriate.

Axially split, horizontal multistage pumps should be limited to 2000-psig discharge pressure. Higher heads require double case or barrel pumps, which are more expensive. In some cases, such as high-pressure pipelines with limited $NPSH_A$, pumps in series should be considered, but shaft sealing becomes difficult as the inlet pressure increases.

3.15.6 Low-head high-flow considerations

When an installation calls for low-head (50–200 ft) (15–60 m) combined with a high flow rate (>5000 gpm) that does not fall within the pump performance range provided by horizontal or in-line pumps, high-capacity pumps and double-suction pumps that provide higher heads than mixed-flow or axial-flow pumps should also be considered. Both of these pumps are designed to move large capacities without the usual $NPSH_R$ by high-capacity suction pumps.

3.15.7 NPSH considerations

Suction considerations can dictate the pump selection. For example, cavitation can occur if $NPSH_A$ is limited or if suction lift is required. Pumps which operate at low speed, have high N_{SS} (suction specific speed), or have double-suction impellers require less $NPSH_R$. In some situations, vertical-turbine barrel or self-priming pumps may be the most reasonable solution. If the TDH requirement is not too high, vertical sump pumps may be used.

3.15.8 Temperature considerations

Pumps normally operate at 250°F (121°C) or less, and as such pump design temperature is normally not an issue. Attention must be given to pump materials and mechanical design in high temperature applications (>450°F). Auxiliary cooling of bearings and seals is recommended at temperatures above 300°F (149°C), plus pedestal cooling at temperatures above 500°F (260°C). Some applications require pumps to operate above 800°F (427°C). Recommended bearing, seal, and pedestal cooling arrangements are given in API Standard 610.

Pump application guidelines horizontal centrifugal

Pump description	End suction							Between BRG design	
	ANSI	Self primer	Magnetic	Electric canned	API	Slurry	Double suction	Two stage	Multiple stages
Number of stages	1	1	1	1	1	1	1	2	2
Diff. press. range (HP)	20–600	20–250	50–700	20–700	60–800	60–600	30–1000	150–1500	800–7000
Row range (GPM)	20–3500	20–1000	4–2000	8–700	40–10,000	40–8000	50–50,000	40–3000	20–5000
Temp. range (F)	<250	<250	<250	<250	<750	<750	<400	<750	<400
Horsepower range (HP)	<200	<70	<200	<150	<4000	<800	<3000	<600	<5000
Speed range (RPM)	<3600	<3600	<3600	<3600	<3600	<1800	<3600	<3600	<7000
Efficiency range (%)	<75	<70	<75	<75	<85	<85	<85	<75	<85
Viscosity range (SSU)	<1000	<1000	<1000	<700	<1000	<1000	<1000	<1000	<1000
NPSH range recommended (FT)	>4	> 4	> 4	>4	>4	>4	>4	>4	>10
Can meet API specs?	N	N	Y	N	Y	N	Y	Y	Y
Can meet ANSI specs?	Y	Y	Y	N	Y	N	Y	Y	Y
Self-priming	N	Y	N	N	N	N	N	N	N
Must operate close to BEP	N	N	N	N	N	Y	N	N	Y
Can run dry with special design	Y	Y	N	N	Y	Y	N	N	N
Will emulsify	Y	Y	Y	Y	Y	Y	Y	Y	Y
Filed alignment required	Y	Y	Y	N	Y	Y	Y	Y	Y
Remove seal W.O. pulling pump	N	N	N/A	N/A	N	N	Y*	Y*	Y*
Back pullout fearure	Y	Y	Y	N	Y	Y	N	N	N
Good for some entrained gas	N	Y	N	N	N	N	N	N	N
Good for abrasives	N	N	N	N	N	Y	N	N	N
Sealless	N	N	Y	Y	N	N	N	N	N
Easier to add stages	N	N	N	N	N	N	N	N	N
Hard surface wetted parts	N	N	N	N	N	Y	N	N	N
Parallel or series is best	P	P	P	P	P/S	P	P	P/S	P/S
Brg. lub. (oil, grease, stock)	O.G	O.G	S	S	O.G	O.G	O.G	O	O
Thrust brg. in pump or motor	P	P	P	M	P	P	P	P	P
Coulinq rlqid or rexible	F	F	F	N/A	F	F	F	F	F
Open or closed impeller	O.C	O.C	C	C	O.C	O.C	C	C	C
B. Holes/W. Rings/PO Vanes	Either	PO Vanes	BH/WR	BH/WR	BH/WR	PO VaneS	BH/WR	BH/WR	BH/WR

Legend. *Y*, yes; *N*, no; N/A, not applicable

*If cartridge seal used

Fig. 3.160 Horizontal centrifugal pump selection guidelines.

Pump manual

200 Centrifugal pumps

Fig. 200-25 Vertical centrifugal pump application guidelines

Pump description	In-line flexible coupling	In-line rigid coupling	Integral NOx coupling	High speed	Sump	Cantilever	Vertical turbine
Number of stages	1	1	1	1	1	1	>1
Can meet API specs?	N	Y	Y	Y	N	N	Y
Can meet ANSI specs?	Y	Y	Y	N	N	N	N
Self-priming	N	N	N	N	Y	Y	Y
Must operate close to BEP	N	N	N	Y	N	N	N
Can run dry with special design	N	N	N	Y	N	N	N
Will emulsify	Y	Y	Y	Y	Y	Y	Y
Field alignment required	N	N	N	N	N	N	N
Back pullout feature	Y	Y	N	N	N	N	N
Good for some entrained gas	N	N	N	N	N	N	N
Good for abrasives	N	N	N	N	N	Y	N
Sealless	N	N	N	N	Y	Y	N
Easier to add stages	N	N	N	N	N	N	Y
Parallel or series is best	P	P	P	P	N/A	N/A	N/A
Brg. lub. (oil, grease, stock)	G	G	G	O	S	G,S	S
Thrust. brg. in pump or motor	P	M	M	Gear	Either	Either	M
Coupling rigid or flexible	F	R	None	None	Either	R	R
Open or closed impeller	O	O,C	O,C	O	Either	O	O
B. holes/W. rings/PO vanes	Either	Either	Either	Neither	Either	PO vanes	Either

Legend: *Y* = yes; *N* = no

N/A = not applicable

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Fig. 3.161 Vertical centrifugal pump selection guidelines.

3.15.9 Hazardous and toxic fluid considerations

Care should be given when handling hazardous (H_2S, CO_2, etc.) or toxic fluids or hydrocarbons above their flash point. Pumps with dual mechanical seals or seals with external flush should be considered. Pump materials must be carefully selected to make certain they are compatible with toxic, hazardous, or corrosive liquids. Recommended seal flush arrangements are provided in API Standard 610.

3.15.10 Impeller efficiency considerations

Selection of the proper impeller size and the proper number of stages can significantly affect pump efficiency. Wear ring design, material selection, and running clearances can improve efficiency. Impeller disc friction is a major factor affecting overall efficiency. Disc friction effects are more evident in low specific speed (N_S) pumps. These pumps tend to have large diameter, narrow shaped impellers as shown in Fig. 3.48. Fig. 3.162 shows the typical variation of pump losses with N_S. For example, for low N_S impellers ($N_S < 1000$), disc friction accounts for approximately 15% or less in efficiency.

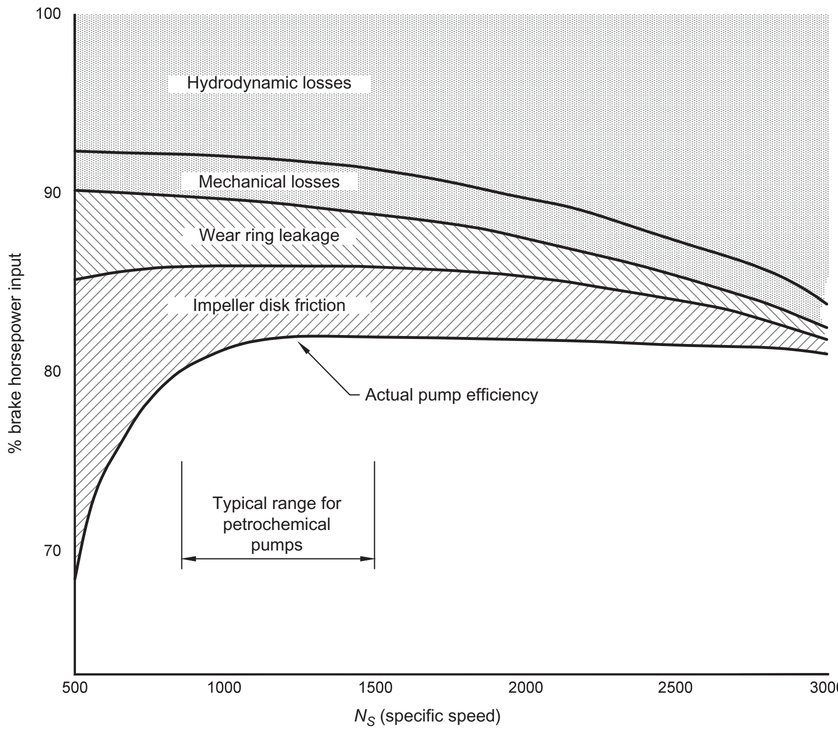


Fig. 3.162 Factors affecting overall pump efficiency.

3.16 Centrifugal pump types

Several different types of centrifugal pumps have been developed to fill both general and specialized needs. The following discussion examines the most commonly used types of centrifugal pumps.

3.16.1 Horizontal centrifugal pumps

3.16.1.1 Horizontal-single-stage, API, top/end suction, top discharge (Fig. 3.163)

Design	The end-suction type centrifugal pump is the largest selling pump in industry today. This style of pump is cost effective and lends itself to small space applications as it occupies less space than axially split-case type pumps. Due to the design, most piping connections can be made in limited space Most end-suction pumps are designed so they can be serviced without disconnecting the system piping. These pumps are available as a uni-type design or frame mounted design. Manufacturers provide these pumps to meet minimum standards as outlined by the Hydraulic Institute
Construction	These pumps are provided in many metallurgies including cast iron, carbon steel, stainless steel, and special coatings and can come with special mechanical seals and different styles of impellers
Applications	Used in process systems, water treatment, sewage systems, industrial applications, chemical processing, hydrocarbon processing and similar systems. They are limited to the capacity ranges mentioned and in general should have a double volute design for 3-in. and larger discharge. This will extend bearing life and provide for higher efficiencies
Typical service	Continuous duty, refinery, upstream facility, and critical water service
Head range	10–800 ft (3–244 m)
Capacity range	5–50,000+ gpm
Speed range	Up to 3600 rpm
Max allowable temperature	350°F (177°C) without cooling 500°F (260°C) with bearing cooling 800°F (427°C) with bearing and pedestal cooling
Standard materials	Available in cast and alloy steel Not available ductile or cast iron
Suction	Single or double suction end-suction
Casing	Radially split, back pullout for maintenance
Impeller	Normally closed impellers
Lubrication	Oil lubrication
Packing/seals	Packing, single or multiseals
Support	Centerline mounted, frame mounted, flexible coupled
Specification	API 610
Typical control method	Throttled discharge on flow, level or pressure

Advantages	More rugged and reliable than ANSI or Industry Standard pumps Available in a wide range of pressures and capabilities Lower operating costs since efficiency is usually high Available in overhung design up to 900 HP (671 kW)
Limitations	Most expensive standard centrifugal pump

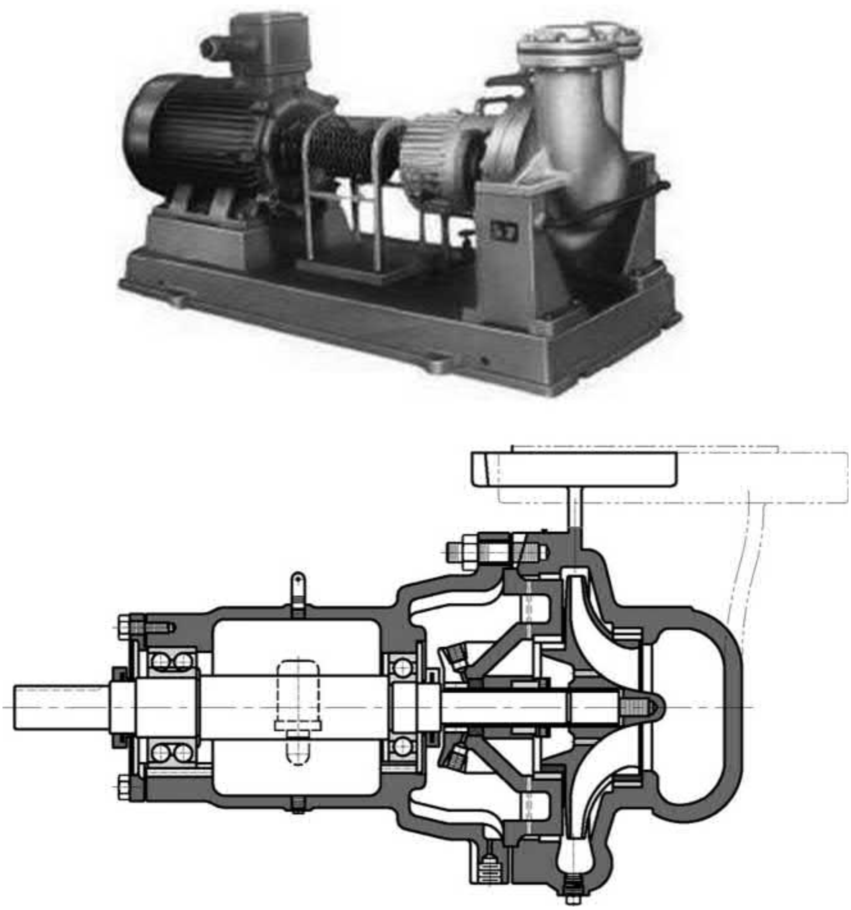


Fig. 3.163 Horizontal, single-stage, API, top/end suction, top discharge overhung pump. Also available a standard with an end-suction arrangement. (Top) Assembled; (bottom) sectional. Courtesy of Peerless Pump Company.

3.16.1.2 Horizontal, single-stage, ANSI B73.1, end suction, top discharge (Fig. 3.164)

Typical service	General purpose. chemical, water, noncritical hydrocarbon
Head range	50–600 ft (15–183 m)
Capacity range	50–3500 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast or ductile iron 316 SS and alloy 20 Not available in carbon steel
Suction	End suction, top discharge
Casing	Back pullout for maintenance
Impeller	Open or closed
Lubrication	Ball bearing grease or oil lubricated
Packing	Single, tandem or double seals
Support	Foot mounted
Specification	ANSI B73.1
Typical control method	Throttled discharge on flow, level, or pressure control
Advantages	ANSI pumps (for each size) are dimensionally interchangeable from any manufacturer Less expensive than API pumps less expensive than API pumps Wide variety of alloy construction materials available
Limitations	150 HP (112 kW) maximum recommended Not available in carbon steel Pressures limited to 275 psig @ 60°F (1896 kPa @ 16°C)

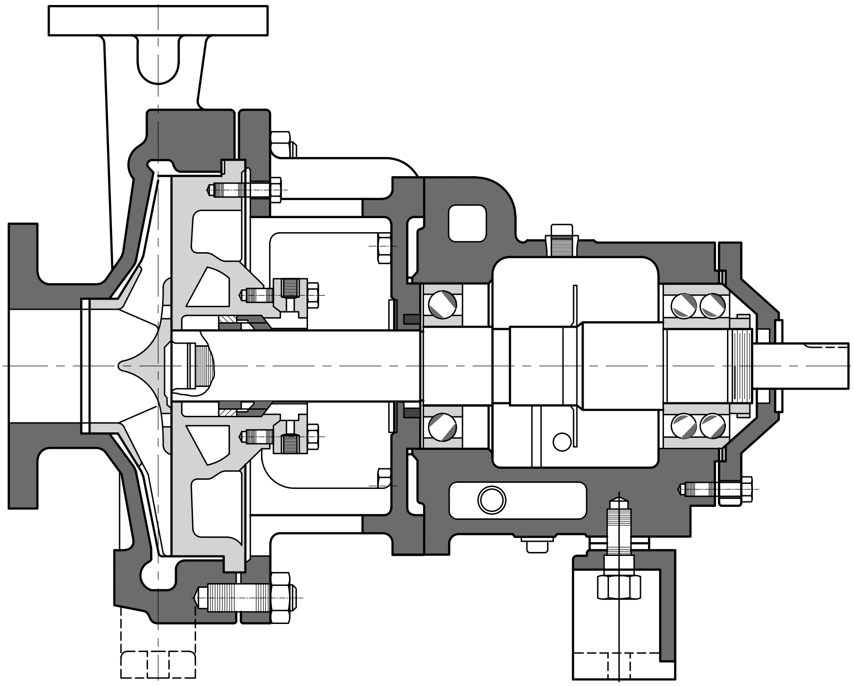


Fig. 3.164 Horizontal, single-stage, ANSI B73.1, end-suction, top discharge. (Top) Assembled; (bottom) sectional.
Courtesy of Worthington a division of Ingersoll Dresser Pumps.

3.16.1.3 Horizontal, single-stage, ANSI B73.1, end suction, top discharge, self-priming (Fig. 3.165)

Typical service	Used for vertical lift when nonpulsating flow is required sump pump-out
Head range	150–250 ft (46–76 m)
Capacity range	0–1000 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast or ductile iron 316 SS and alloy 20 Not available in carbon steel
Suction	End suction, top discharge
Casing	Back pullout for maintenance
Impeller	Open or closed
Lubrication	Ball bearing grease or oil lubricated
Packing	Single, tandem, or double seals
Support	Foot mounted
Specification	ANSI B73.1
Typical control method	Discharge on/off level control
Advantages	Up to 20 ft (6 m) effective static lift Eliminates need for foot valve Dimensionally interchangeable with all ANSI pumps More reliable than submerged vertical sump pumps
Limitations	Less efficient than standard non-self-priming pumps May take too long to prime on large suction lines Mechanical seal may run dry without an external flush

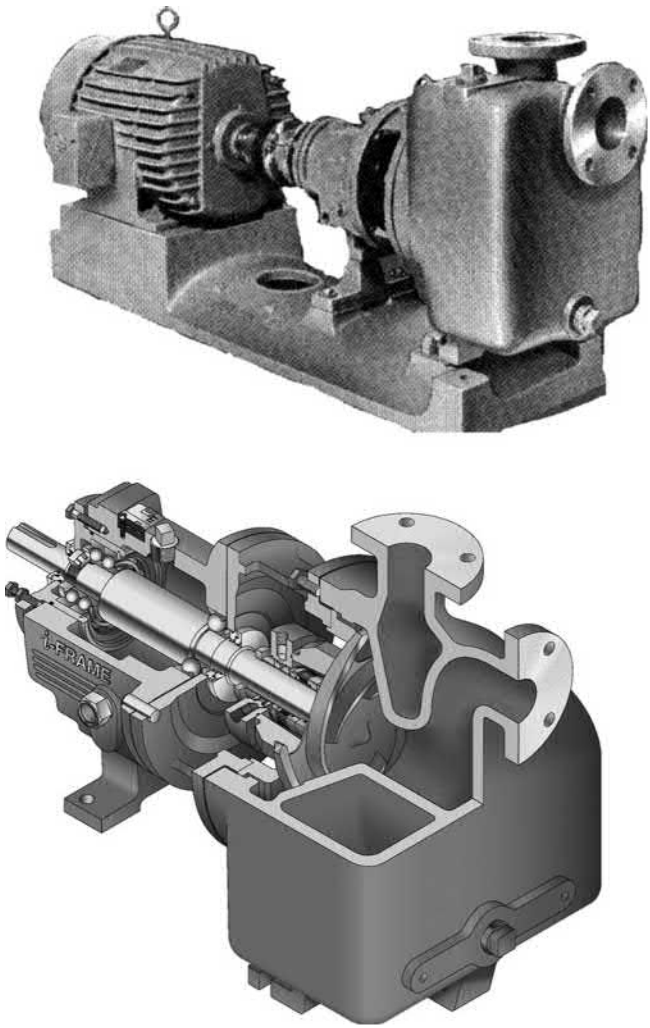


Fig. 3.165 Horizontal, single-stage, self-priming, ANSI B73.1 pump. (Top) Assembled; (bottom) sectional.
Courtesy of Goulds Pumps, Inc.

3.16.1.4 Horizontal, single-stage, double-suction, horizontally split pump (Fig. 3.166)

Design	The axially split-case centrifugal pump has been in use for many years as the heavy-duty work horse. This pump is designed for larger flows and higher heads than for end-suction type pumps. It is a double suction design allowing for flow to enter the impeller from opposite sites. This provides for a more balanced thrust load, which in turn produces a smoother running pump with somewhat higher efficiencies. This pump is very easy to service due to the accessibility to the entire rotating assembly without disturbing any of the connecting piping. The design also contributes to having a longer service life for the bearings. Pumps are also designed for vertical mounting, which allows for less floor space to be used
Construction	These pumps can be provided in many metallurgies. Standard construction casings are rated for ASME 125 class with options up to 300 lb. Pumps are available with special alloys and accessories to accommodate severe applications. Pumps are also available in multistage construction for high pressure applications with flange ratings up to ASME 2500 class
Applications	These pumps are used in heavy-duty industrial and commercial services, including water treatment facilities, irrigation projects, process water heating, air-conditioning, and so on. They are sometimes limited due to space required for adequate piping and valves
Typical service	Cooling water circulation Fire pump Crude oil transfer pump
Head range	20–1100 ft (6–335 ms)
Capacity range	200–500,000 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast iron or bronze case (steel case for hydrocarbons) and bronze trim
Suction	Horizontal inlet and outlet
Casing	Back pullout for maintenance
Impeller	Closed Stainless steel impellers available for higher cavitation resistance
Lubrication	Ball bearing grease or oil lubricated
Packing/seals	External sleeve or antifriction bearings
Support	Foot mounted
Specification	API 610 (hazardous, flammable, and special purpose)
Typical control method	Throttled discharge on/off level control
Advantages	Balanced thrust on shaft will maintain pump in place Low NPSH requirements Wide range of sizes and capacities
Limitations	More expensive than single suction, overhung design Suction lines must be carefully designed to avoid nonsymmetrical flow that would channel to one side, resulting in unbalanced thrust and possibly cavitation

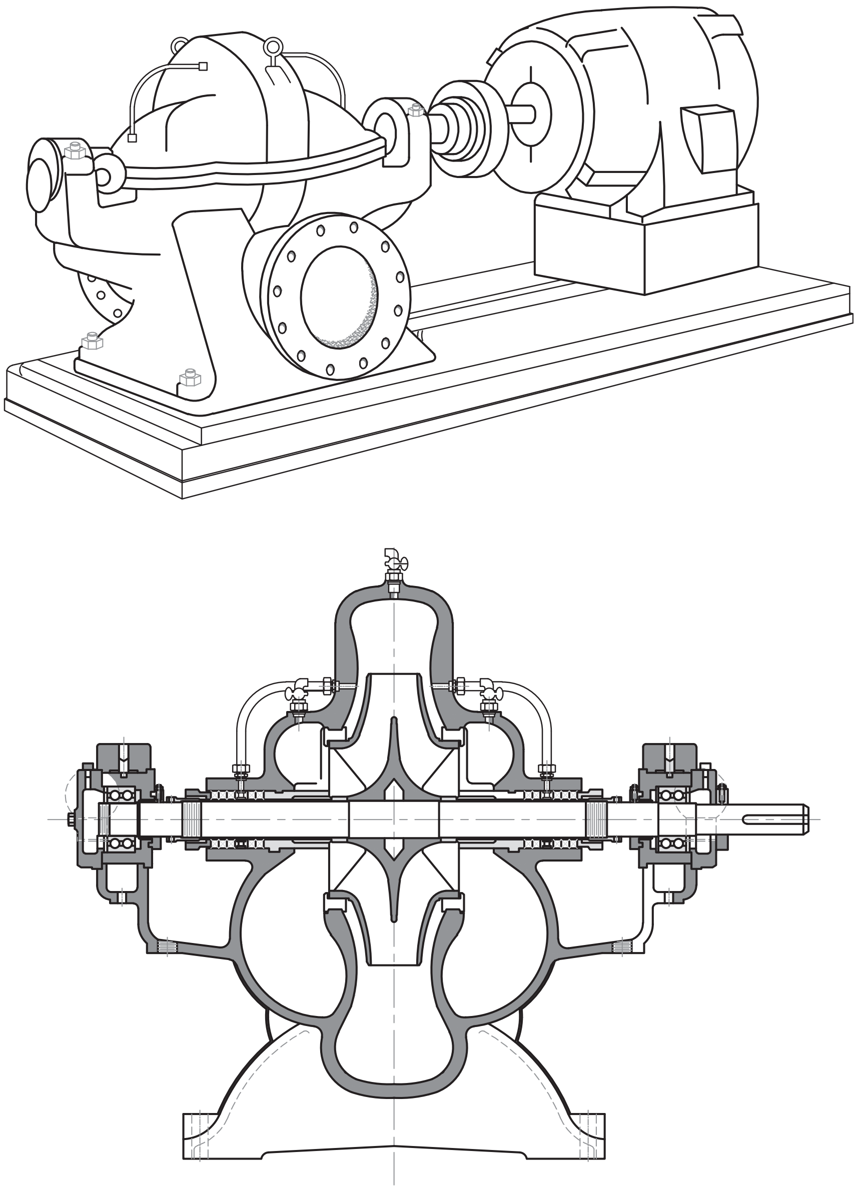


Fig. 3.166 Horizontal, single-stage, double-suction, horizontal split case. (Top) Assembled; (bottom) sectional.
Courtesy of Goulds Pumps, Inc.

3.16.1.5 Horizontal, multistage, API, horizontally split pump suction (Fig. 3.167)

Typical service	Crude oil feed pumps Waterflood Process Pipeline Boiler feed-water
Head range	200–7050 ft (61–2149 m)
Capacity range	100–7000 gpm
Speed range	Up to 7000 rpm
Max allowable temperature	250°F (122°C) without cooling 400°F (204°C) with cooling
Standard materials	Carbon steel case Cast iron, stainless steel, or bronze impellers
Suction	Horizontal nozzles, both suction and discharge nozzles located in bottom half casing
Casing	Horizontally split
Impeller	Closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals	Bearing
Support	Centerline supported
Specification	API 610
Typical control method	Throttled discharge on flow, level, or pressure control
Advantages	Ease of in-line assembly and inspection Can be designed with balanced axial thrust Eliminates multiple in-line series pumps
Limitations	API 610 limits horizontally split case design to applications below 400°F (204°C) and pumped fluids with specific gravity above 0.70 More complex than single-stage pumps (note that pressures to 2000 psig (13,790 kPa) are common in producing waterflood applications)

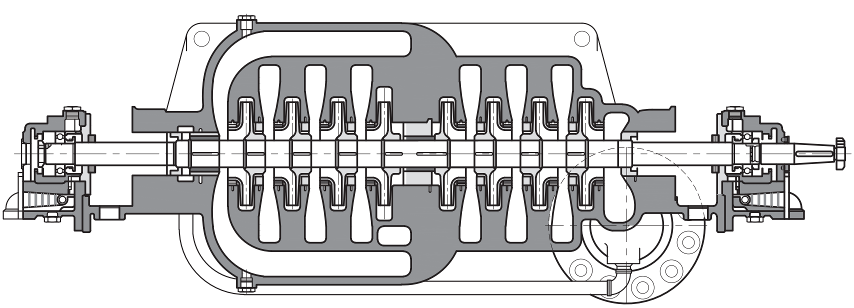


Fig. 3.167 Horizontal, multistage, API 610, horizontally split case.
Courtesy of Flowserve Corporation.

3.16.1.6 Horizontal, multiple-stage, API, vertically split double-case (high pressure and high temperature) (Fig. 3.168)

Typical service	High-pressure process feed Crude pipelines Boiler feed-water
Head range	0–10,000 ft (0–2134 m)
Capacity range	100–5000 gpm
Speed range	1800–7000 rpm
Max allowable temperature	850°F (454°C) w/pedestal, bearing, and seal cooling
Standard materials	Carbon steel
Suction	Top suction/discharge Nozzle location may vary with installation requirements
Casing	Vertically (radially)-split, double-case
Impeller	Closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals	Bearings and seals Water-cooled pedestals, bearings, and seals available
Support	Centerline
Specification	API 610
Typical control method	Spillback on external flow control
Advantages	High pressures are possible without series pump operations Double-casing allows in-line assembly/disassembly
Limitations	Clearances sensitive to differential temperatures in pump Slow pump start-up mandatory with hot pumps Proper assembly difficult with many stages Important not to run with blocked discharge

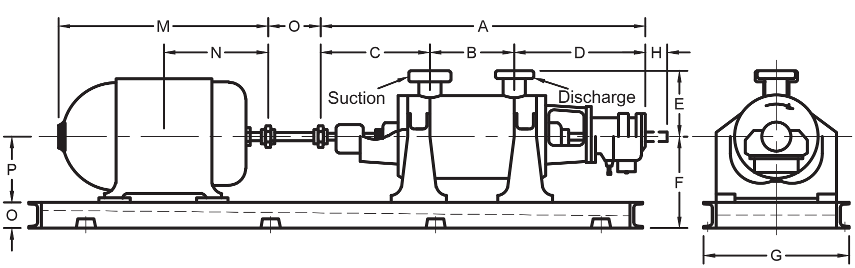


Fig. 3.168 Horizontal, multistage, API 610, vertically split case (high pressure, high temperature).
Courtesy of Pacific a division of Ingersoll Dresser Pumps.

3.16.2 Vertical centrifugal pumps

3.16.2.1 Vertical, single-stage, ANSI B73.2, rigid-coupling pump (Fig. 3.169)

Typical service	Chemical Water Noncritical hydrocarbons General purpose
Head range	600 ft (183 m)
Capacity range	3000 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast iron and ductile iron 316 SS and alloy 20
Suction	Suction/discharge flanges with common centerline which intersects shaft axis
Casing	Integral
Impeller	Closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings	Motor bearings carry pump loads
Support	Motor supported by pump
Specification	ANSI B73.2
Typical control method	Throttled discharge on flow, level or pressure control
Advantages	Can remove seal and impeller without disturbing motor Unit is interchangeable with all other vertical ANSI designs Simpler and cheaper to install than horizontal Occupies less floor space No field alignment of pump and motor needed (as long as it fits and remains within tolerance)
Limitations	150 HP (112 kW) maximum Cannot install dual mechanical seals Vapor or gas in liquid tends to collect at mechanical seal faces. This promotes failure unless properly vented during start-up (needing a vent), and flushed during operation Single-stage rigid couplings are always troublesome to keep in alignment (causes short bearing and seal life typically)

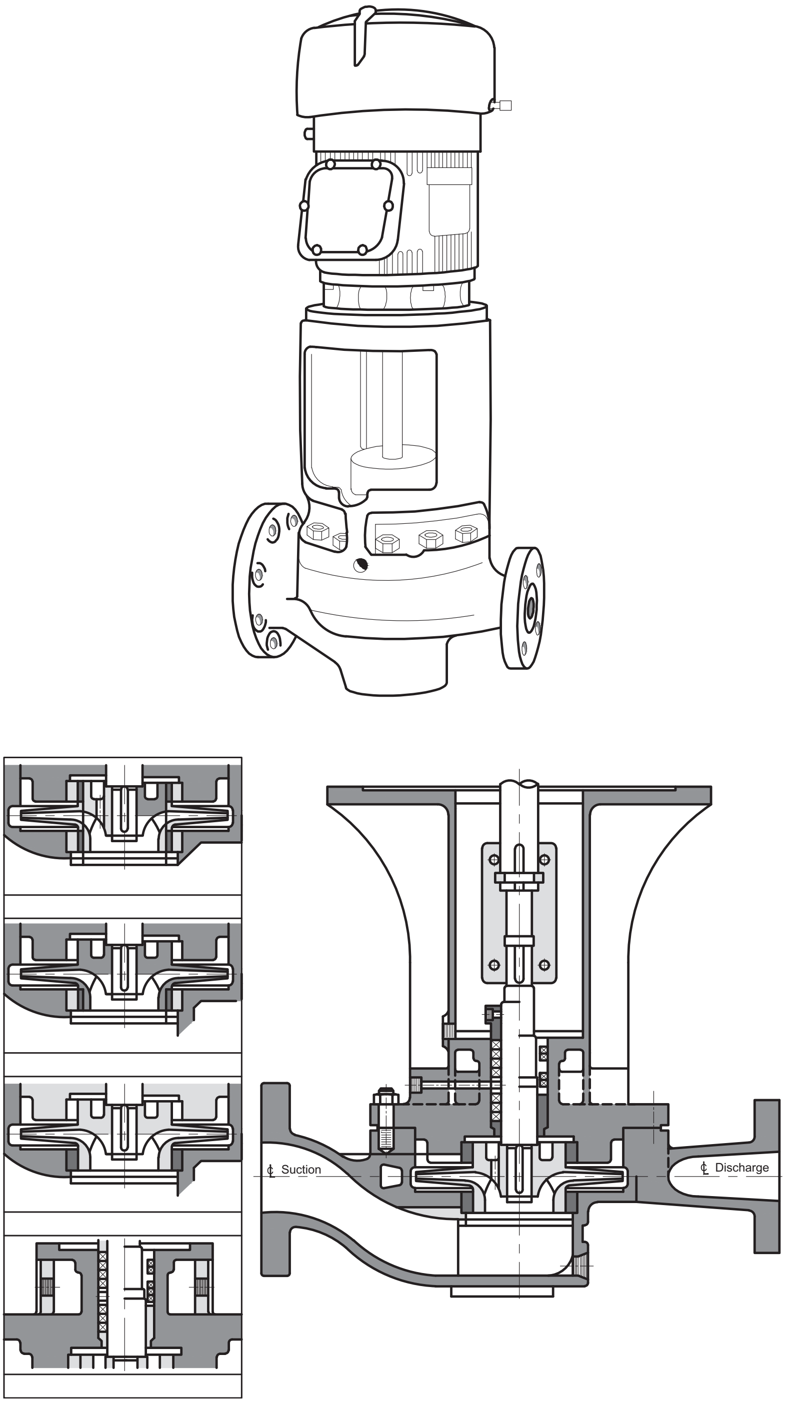


Fig. 3.169 Vertical, single-stage, in-line, ANSI B73.2, rigid coupling: (top) Assembled; (bottom) sectional.
Courtesy of Flowserve Corporation.

3.16.2.2 Vertical, single-shaft, in-line, ANSI B73.2, integral-shaft pump (Fig. 3.170)

Design	The vertical in-line centrifugal pump was developed to allow for space savings and ease of servicing. The capacity ranges are about the same as end-suction pumps but will use much less floor space. Design is similar to end-suction, but provided with a special volute to guide the fluid to the eye of the impeller on a straight through in-line connection. The pumps can be mounted horizontally as well. It is important to make sure the mechanical seal is properly vented when the pump is mounted vertically. Most construction designs allow for uni-type mountings, but some are available with flexible couplings. Flexibly coupled pumps with a pump bearing housing is the preferred design as the shaft length is minimized, resulting in lower shaft deflection and longer mechanical life
Construction	These pumps are available in many metallurgies with pressure ratings up to ASME 300 class. Most are equipped with mechanical seals
Applications	In-line pumps are used for almost any commercial and industrial application where space is a premium
Typical service	Chemical Water Noncritical hydrocarbons
Head range	600 ft (183 m)
Capacity range	3000 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast iron and ductile iron 316 SS and alloy 20 Carbon steel not available
Suction	Suction/discharge flanges with common centerline which intersects shaft axis
Casing	Integral
Impeller	Closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings	Bearings are in the motor—none in the pump
Support	Motor supported by pump
Specification	ANSI B73.2
Typical control method	Throttled discharge on flow, level, or pressure control
Advantages	Unit is interchangeable with all other vertical ANSI designs Simpler and cheaper to install than a horizontal pump Occupies less floor space No field alignment of pump and motor needed Provides better seal and bearing life than rigidly coupled in-line
Limitations	Must remove motor for access to seal or impeller Cannot accommodate dual mechanical seals

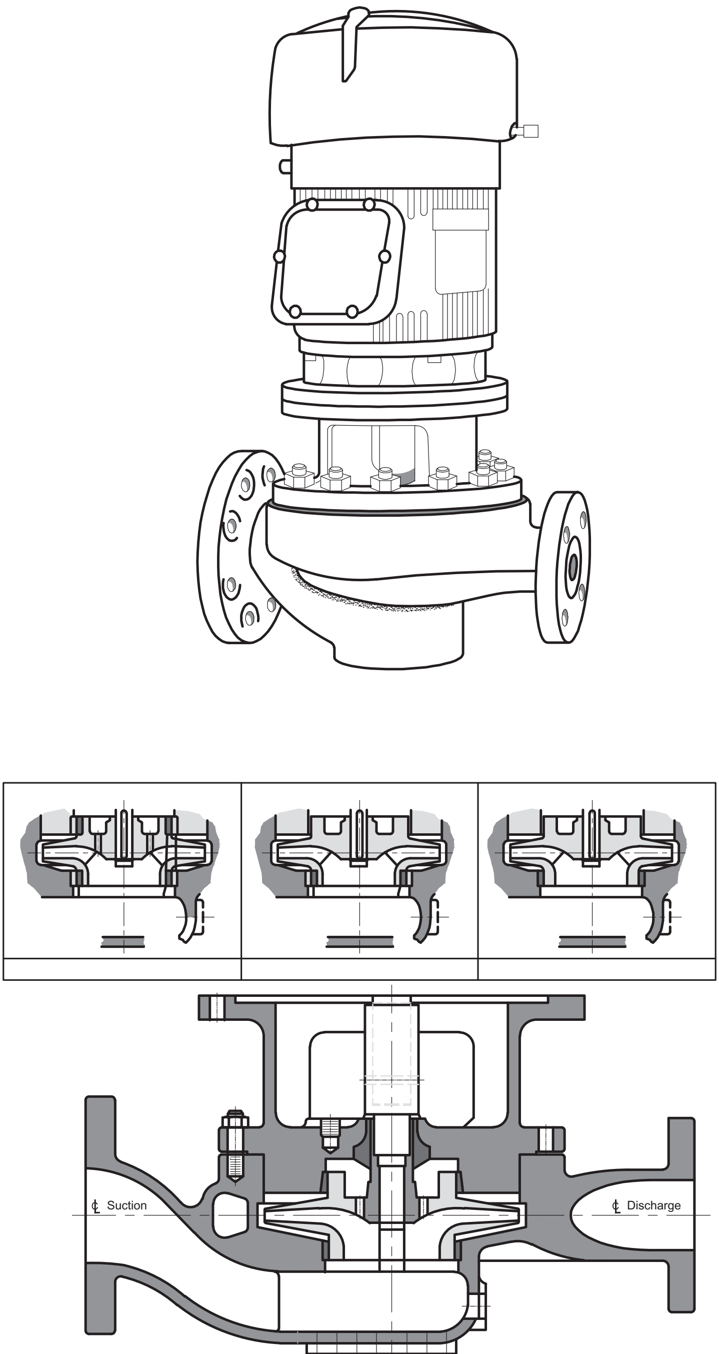


Fig. 3.170 Vertical, single-stage, in-line, ANSI B73.2, integral-shaft. Courtesy of Flowserve Corporation.

3.16.2.3 Vertical, single-shaft, in-line, ANSI B73.2, flexible-coupling pump (Fig. 3.171)

Typical service	Chemical Water Noncritical hydrocarbons General purpose
Head range	600 ft (183 m)
Capacity range	3000 gpm
Speed range	Up to 3600 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast iron and ductile iron 316 SS and alloy 20 Carbon steel not available
Suction	Suction/discharge flanges with common centerline which intersects shaft axis
Casing	Integral
Impeller	Closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings	Pump has own bearings. Otherwise, same as rigid coupling pump
Support	Motor supported by pump
Specification	ANSI B73.2
Typical control method	Throttled discharge on flow, level, or pressure control
Advantages	Field alignment of pump and motor shafts is maintained by register fits Hydraulic loads not carried by motor bearings Can remove seal and impeller without disturbing motor Interchangeable with all other vertical ANSI designs Simpler and cheaper to install than horizontal Occupies less floor space No field alignment of pump and motor needed
Limitations	Complete bearing bracket/pump rotor must be sent to shop for seal repairs More expensive than rigid coupling or integral shaft pumps; otherwise, same as rigid coupling pump Taller and heavier installed height than other vertical in-line options

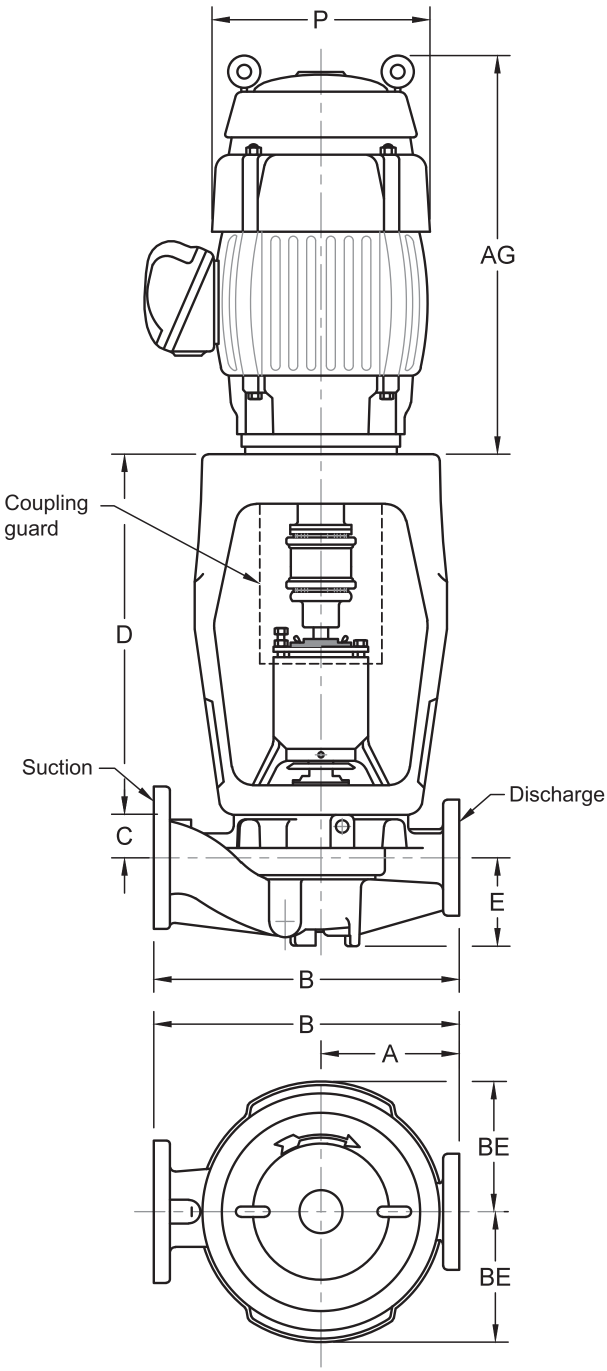


Fig. 3.171 Vertical, single-stage, in-line, ANSI B73.2, flexible-coupling, centrifugal pump. Courtesy of Worthington a division of Ingersoll-Dresser Pump Company.

3.16.2.4 Vertical, single-shaft, in-line, high-speed pump (Fig. 3.172)

Typical service	High head/low flow service for water and hydrocarbons
Head range	0–4500 ft (0–1372 m)
Capacity range	0–800 gpm
Speed range	Up to 15,000 rpm
Max allowable temperature	400°F (204°C) with cooling
Standard materials	Carbon steel 316 SS and alloy 20
Suction	Suction/discharge flanges with common centerline which intersects shaft axis
Casing	Integral
Impeller	Open
Lubrication	Integral gear box with self-contained lube system
Packing/seals/ bearings	Built-in seal flush, dual seals available
Support	Motor supported by pump
Specification	API 610
Typical control method	Minimum flow bypass with flow control
Advantages	Less expensive to purchase and install than comparable moderate high-pressure horizontal, centrifugal, and plunger pumps Field alignment of pump/motor not required Occupies less floor space than equivalent horizontal or PD pumps
Limitations	Special prelube system for higher suction-pressure applications Separate minimum flow bypass with controller for each pump High speed creates seal face problems Vapor collecting at top of case can cause seal failure if not flushed Accidental reverse rotation can loosen impeller and cause failure Must dismantle to replace seals May have unstable performance curve at low flows 400 HP maximum More $NPSH_R$ and much less efficient than equivalent horizontal pumps Better metallurgy required for impeller/diffuser due to sensitivity of performance vs internal clearances There are numerous ports (seal flush, vents, etc.) which are complex and must be carefully piped up

Fig. 3.172 Vertical, single-stage, in-line, high speed pump. Courtesy of Sundstrand Fluid Handling Company.

3.16.2.5 Vertical, sump, single-shaft, bearing supported pump (Fig. 3.173)

Typical service	Sump pump-out Sewage Nonabrasive solids Sludge
Head range	20–250 ft (6–76 m)
Capacity range	50–2000 gpm @30 ft
Speed range	Up to 1800 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Cast iron, plastic and 316 SS
Suction	Submerged
Casing	Integral

Impeller	Open or closed
Lubrication	Ball bearing lubricated with grease, fluid injection, or pumped fluid
Packing/seals/ bearings	Bearings are in the motor—none in the pump
Support	Motor supported by pump
Specification	–
Typical control method	On/off level control and throttled discharge
Advantages	Simple mounting Foundation not required No stuffing box or seal leakage Submerged impeller
Limitations	20 ft (6 m) is a practical limit Less reliable than self-priming horizontal or vertical cantilever pump Line shaft bearings require lubrication from one of the following: (1) grease, (2) continuous water/pumped fluid injection, (3) pumped fluid

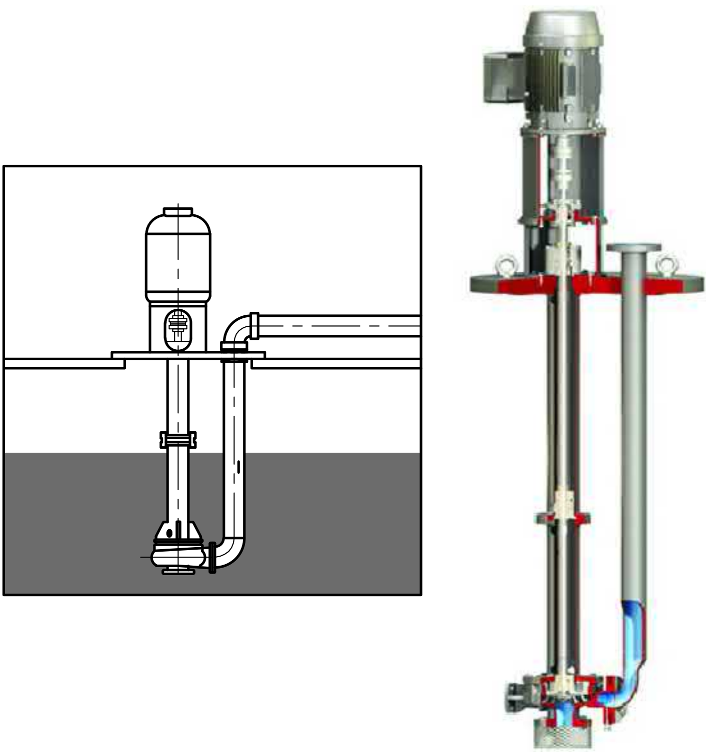


Fig. 3.173 Vertical, sump, single-stage, bearing supported pump. Courtesy of Goulds Pumps Corporation.

3.16.2.6 Vertical, sump, single-stage, cantilever impeller and shaft pump (Fig. 3.174)

Typical service	Sump pump-out Sewage Abrasive solids, sludge and slurry
Head range	0–200 ft (0–60 m)
Capacity range	0–5000 gpm @100 ft
Speed range	Up to 1800 rpm
Max allowable temperature	200°F (93°C)
Standard materials	Cast iron, plastic, and 316 SS
Suction	Submerged
Casing	Integral
Impeller	Open or closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings	No bearing/pumped liquid contact
Support	Large-diameter shaft to support cantilevered impeller
Specification	–
Typical control method	On/off level; throttled discharged
Advantages	No bearing/pumped liquid contact More reliable than bearing supported vertical sump pumps in abrasive or sludge service Simple mounting; no foundation No stuffing box or seal Submerged impeller Pump can run dry for short periods
Limitations	10 ft (3 m) practical shaft limit Requires rigid, large diameter shaft More expensive than bearing supported vertical sump pump

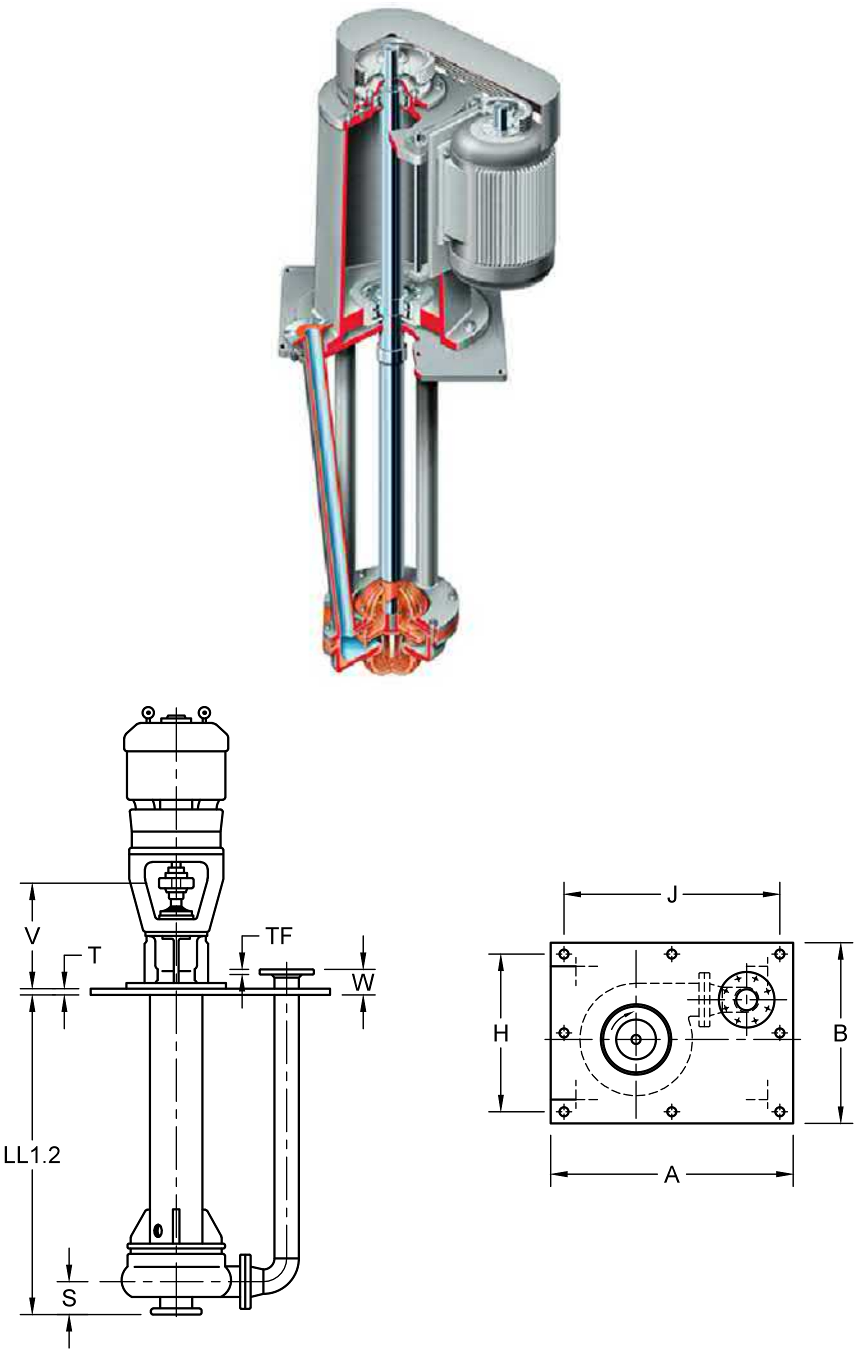


Fig. 3.174 Vertical, sump, single-stage, cantilever impeller and shaft pump. Courtesy of Worthington a division of Ingersoll Dresser Pump Company.

3.16.2.7 Vertical, multistage, barrel pump (Fig. 3.175)

Typical service	Low $NPSH_A$ applications Boiler feed-water Flashing liquid
Head range	0–3500 ft (0–1067 m)
Capacity range	0–80,000 gpm
Speed range	Up to 3600 rpm; however, 1800 rpm is the preferred maximum speed for improved reliability
Max allowable temperature	650°F (343°C)
Standard materials	Steel barrel and steel or cast iron head with cast iron bowls and cast iron steel or bronze impellers
Suction	Submerged
Casing	Integral
Impeller	Open or closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings:	Weight of pump and pump thrust by motor thrust bearing
Support	Motor supported by pump
Specification	API 610 (hazardous, flammable, and special purpose services)
Typical control method	Throttled discharge on flow, level, or pressure control
Advantages	Has a small "foot print" and requires less floor space Low $NPSH_R$ Offers high efficiency
Limitations	Shaft sleeve bearings exposed to pumped liquid Must remove pump for all maintenance except mechanical seal changes

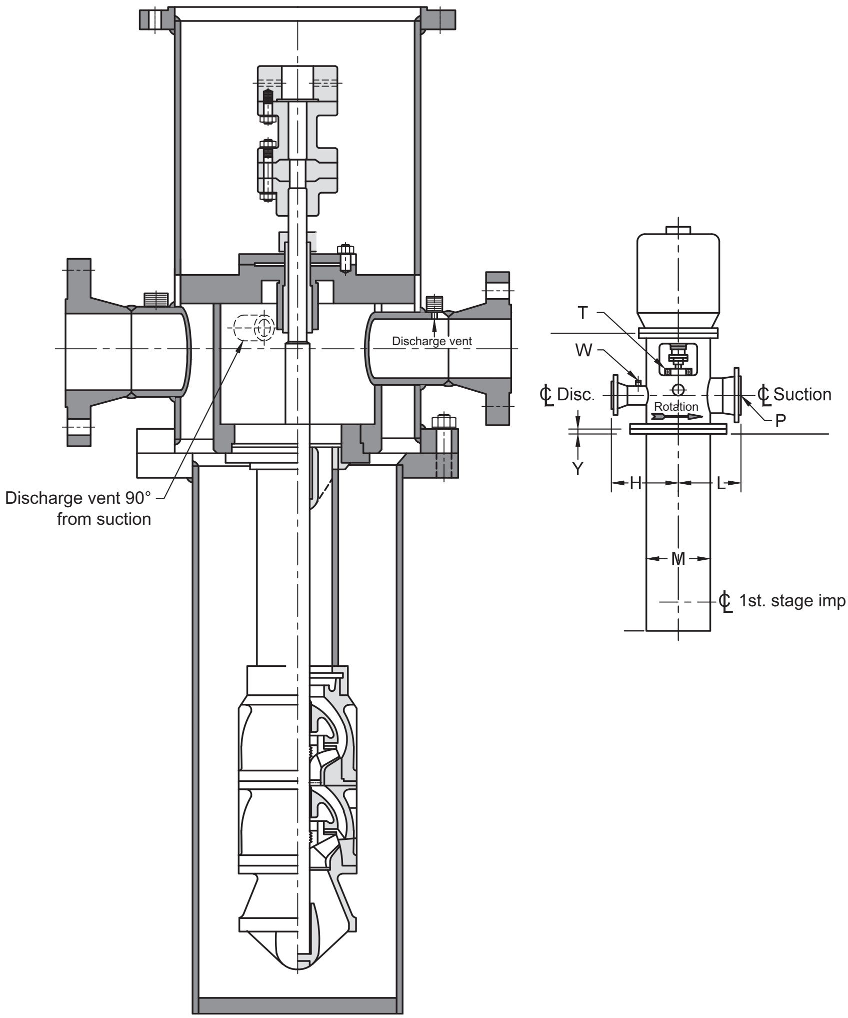


Fig. 3.175 Vertical, multistage, barrel pump. Courtesy of Flowserve Corporation.

3.16.2.8 Vertical, multistage, deep-well (vertical turbine) pump (Fig. 3.176)

Design	The vertical turbine pump was designed for applications where standard centrifugal pumps could not handle the application. They are very popular for well water pumping in both the column type and the submersible for deeper well applications. This design allows for single-stage performance, to multiple stage performance, to develop higher heads, while maintaining flow. Pumps may have many types of discharge heads and bearings may be lubricated with special oil lubricators, as well as the product being handled. Pumps can be provided with flexible line shafting or direct connected line shafting, depending on the service required. The vertical turbine has a very definite place in industrial, commercial, municipal, and irrigation projects
Construction	Vertical turbine pumps can be manufactured in many different alloys. Most pumps will have cast iron bowl assemblies, bronze impellers, steel columns, cast iron and fabricated steel discharge heads, stainless steel line shafting, bronze sleeve bearings, and a variety of upper seal chambers. To provide better alignment and to facilitate repair disassembly/reassembly, the pump bowls should be flanged
Applications	These pumps are typically a heavy-duty construction and are often used for condenser water service on large cooling tower projects, large cooling and heating systems, and many more heavy-duty applications
Typical service	Firewater Seawater Platform
Head range	0–3500 ft (0–1067 m)
Capacity range:	60,000 gpm
Speed range	Up to 1800 rpm
Max allowable temperature	250°F (122°C)
Standard materials	Steel barrel and steel or cast iron head with cast iron bowls and steel or cast iron impellers
Suction	Submerged
Casing	Integral
Impeller	Semiopen of closed
Lubrication	Ball bearing grease or oil lubricated
Packing/seals/bearings	Bearings are in the motor—none in the pump
Support	Weight of shaft and hydraulic thrust supported by vertical motor bearings
Specification	–
Typical control method	Throttled discharge. Level control for sumps
Advantages	Exhibits high efficiency Can be installed in wells or wet pit sumps Can be provided with engine driver with right angle drive
Limitations	Size limited to diameter or well casing Practical maximum setting depth is 1000 ft @ 1800 rpm (305 m @ 1800 rpm)

	Bowl bearings are process fluid lubricated Abrasiveness will shorten pump life Available with submersible motors which eliminates long drive shafts; however, submersible motor installations are less reliable and are not recommended above 50 HP (37 kW)

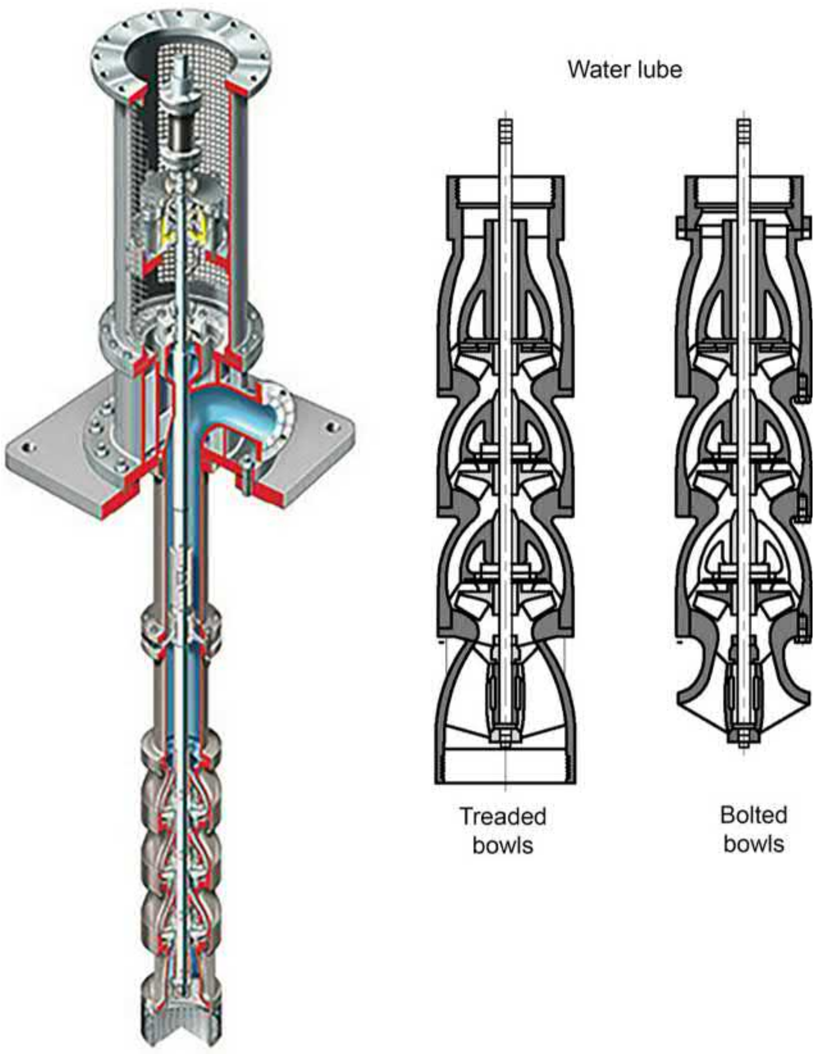


Fig. 3.176 Vertical, multistage, deep-well (vertical turbine) pump. Courtesy of Flowserve Corporation.

3.17 Troubleshooting

Troubleshooting is the key to solving problems properly. If one understands their system, understands how to determine a system curve, and knows what type of pump works best for a specific system, then one can look at a realistic approach to solving a problem. It does not need to be harder than it is. The operation of a centrifugal pump may be affected by hydraulic or mechanical troubles. It is important to note that there is often a definite connection between these two kinds of difficulties. For example, increased wear at the running clearance would be classified as a mechanical issue, but will result in a reduction of the pump's capacity—a hydraulic symptom—without necessary causing a mechanical breakdown or excessive vibration. Therefore it is important to classify symptoms and causes separately and to develop a schedule of potentially contributory causes for each symptom. Table 3.15 can serve as a starting point to solve problems and can save considerable time and effort in the office and field.

Table 3.15 Troubleshooting centrifugal pumps

Check		Problem				
		Pump does not deliver liquid	**Insufficient capacity delivered**	**Insufficient pressure developed**	**Loss of prime after starting**	**Excessive power required**
1	Pump not primed					
2	Suction pipe or pump 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					

Continued

	Check	Problem				
		Pump does not deliver liquid	Insufficient capacity delivered	Insufficient pressure developed	Loss of prime after starting	Excessive power required
7	Air leakage into suction line		■		■	
8	Air leak into pump through stuffing box		■		■	
9	Foot valve too small		■			
10	Foot valve partially clogged		■			
11	Inlet of suction pipe insufficiently submerged	■	■		■	
12	Water seal pipe plugged				■	
13	Seal case improperly located thus preventing seal formation				■	
14	Speed too low	■	■	■		■
15	Speed too high					■
16	Wrong direction of rotation	■		■		■
17	Total system head higher than pump design head	■	■	■		■
18	Total system head lower than pump design head					■
19	Specific gravity of liquid different from design					■
20	Viscosity of liquid different from design		■	■		■

Continued

Check		Problem: Pump does not deliver liquid	Insufficient capacity delivered	Insufficient pressure developed	Loss of prime after starting	Excessive power required
21	Operation at very low capacity					
22	Parallel operation of pumps is unsuitable					
23	Foreign matter in impeller					
24	Misalignment					
25	Foundation 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 the packing					
33	Packing improperly installed					
34	Incorrect type of packing for operating conditions					
35	Shaft running off center due to worn bearings or misalignment					

Continued

Check		Problem				
		Pump does not deliver liquid	Insufficient capacity delivered	Insufficient pressure developed	Loss of prime after starting	Excessive power required
36	Rotor out of balance resulting in vibration					
37	Gland too tight: No flow to lubricate packing					
38	Lack of lubrication					
39	Dirt getting into bearings					
40	Rusting of bearing					

Check		Problem				
		Excessive leaking of stuffing box	Packing has short life	Pump vibrates or is noisy	Bearings have short life	Pump overheats and ceases
1	Pump not primed					
2	Suction pipe or pump 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 leakage into suction line					

Check		Problem: Excessive leaking of stuffing box	Problem: Packing has short life	Problem: Pump vibrates or is noisy	Problem: Bearings have short life	Problem: Pump overheats and ceases
8	Air leak into pump through stuffing box					
9	Foot valve too small					
10	Foot valve partially clogged					
11	Inlet of suction pipe insufficiently submerged					
12	Water seal pipe plugged					
13	Seal case improperly located thus preventing seal formation					
14	Speed too low					
15	Speed too high					
16	Wrong direction of rotation					
17	Total system head higher than pump design head					
18	Total system head 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 is unsuitable					
23	Foreign matter in impeller					
24	Misalignment					

Continued

	Check	Problem				
		Excessive leaking of stuffing box	Packing has short life	Pump vibrates or is noisy	Bearings have short life	Pump overheats and ceases
25	Foundation 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 the packing	■	■			
33	Packing improperly installed	■	■			
34	Incorrect type of packing for operating conditions	■	■			
35	Shaft running off center due to worn bearings or misalignment	■	■	■	■	■
36	Rotor out of balance resulting in vibration	■	■	■	■	■
37	Gland too tight: No flow to lubricate packing		■			
38	Lack of lubrication			■	■	
39	Dirt getting into bearings			■	■	
40	Rusting of bearing			■	■	

3.17.1 Common causes of pump failure

The most common causes of pump failure are as follows:

- Casing/casting failure
- Bearing failure
- Motor failure (for close-coupled pumps)
- Seal failure

One should look for these possible failures first before evaluating other items. If the earlier possibilities are eliminated then one can start checking items in the system, such as check valves, gate valves, strainers, and piping. *Note: Strainers should always be checked and cleaned routinely to ensure the performance of any pump.*

3.17.1.1 Casing/casting failure considerations

Casing/Casting failure considerations include:

- Casing failure generally occurs due to improper material compatibility with pumped fluid, chemical incompatibility, or abrasives.
- Material selections are best made with the assistance of an experienced pump designer.
- Seal, bearing, and motor failures are more often the result of improper operation in the system.
- Casting failure could be due to bubbles, porosity, slag inclusions, fissures, and so on, in the mold, or contamination in the metallurgy.

Pumps that are used in caustic or corrosive environments should be brought to the attention of the pump manufacturer. In addition, high temperature or high-pressure applications also require additional attention.

3.17.1.2 Bearing failure considerations

Some of the more common bearing failure considerations include:

- Overheated bearing.
- Noisy bearing.
- Replacements are too frequent.
- Vibration.
- Unsatisfactory performance.
- Bearing is loose on shaft.
- Hard turning shaft.

Assuming a perfect installation (good lubrication, installation without damage, and no bearing defects), bearings will still fail due to fatigue. Fatigue stresses are the result of shear stresses and are observed as spalling. Fig. 3.177A shows incipient fatigue spalling. Fig. 3.177B illustrates advanced spalling and Fig. 3.177C shows greatly advanced spalling. If a bearing is replaced, it should always be kept to examine it for the root cause of failure and use the load path patterns as clues. If the same bearing is replaced every six months with the same wear characteristics, chances are the same cause is at work and must be dealt with.

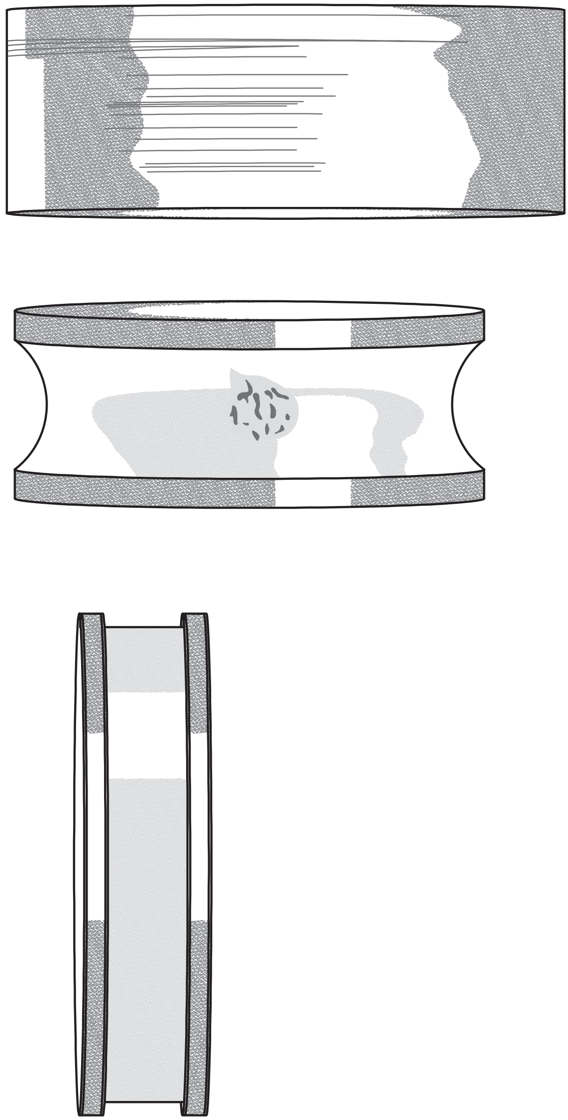


Fig. 3.177 (Top) Incipient fatigue spalling; (middle) advanced spalling; (bottom) greatly advanced spalling.

The kind of bearing grease used can be as much of a problem as not using any grease at all. Lithium-based grease is used for pumps while Polyurea-based grease is used for motors. If mixed, bearing failure will result. Polyurea has better water/condensation resistance. Common Polyurea greases include Chevron NB 2, Shell NRLB 2, and Texaco NRLB 2. Lithium (axle grease) has better availability.

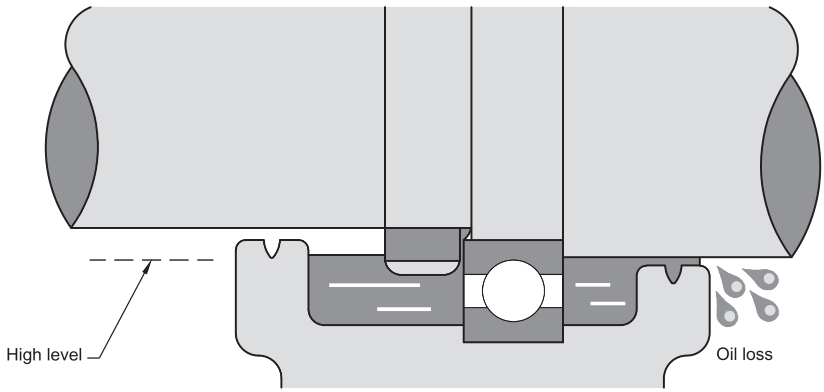


Fig. 3.178 Too much oil or grease may cause overheating and increase friction.

As shown in Fig. 3.178, too much oil or grease may overheat the bearing or make the shaft hard to turn due to increased friction. As shown in Fig. 3.179, too low an oil level or insufficient grease may overheat the bearing, may cause bearing noise, may cause the bearing to be replaced too frequent and/or increase friction. The oil level should be just below the center of the lowest ball or roller in the bearing. Using grease, the lower half of pillow block should be ½ to 1/3 full.

3.17.1.3 Stuffing box and mechanical seal failure considerations

As shown in Fig. 3.94C, the shaft of the pump passes through the casing (back-plate), there is a space between the shaft and the casing. The fluid will flow from the inside to the outside (from high to low pressure). In suction lift applications, the stuffing box

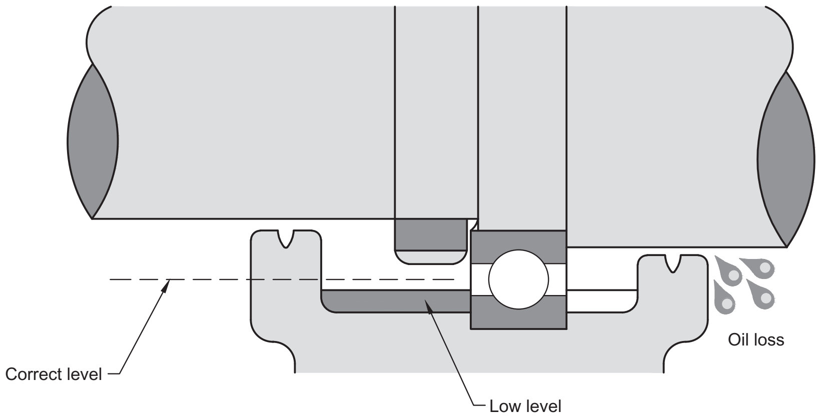


Fig. 3.179 Too low oil level or insufficient grease may overheat the bearing, cause noise, and increase friction.

will be under a partial vacuum. Air will flow into the stuffing box unless supplied with sealing liquid to the lantern ring. To stop this flow, packing and seals are used to plug this gap. They:

- Preserve the pumped fluid.
- Protect personnel and the environment.
- Save money.

To perform these functions properly, it is extremely important to pick the right kind of seal or packing. They must be selected to match the fluid being pumped, the temperature, and other system and site requirements. The simplest form of shaft sealing is packing.

Every pump except for canned rotor, submersible, and magnetic drive pumps has one or two stuffing boxes. This one component of a pump causes more problems than all other components combined. To minimize these problems, much thought should go into the selection of the best sealing system, whether it be some sort of packing or mechanical seals. Included in this best sealing system is a properly selected pump with, for example, minimal shaft deflection, pipe strain, and an optimum foundation. There are three sealing systems:

1. *Square braided packing*: Can be made of any flexible material with suitable lubricants added to reduce the rubbing friction of the packing on the rotating shaft or shaft sleeve. This lubricant can be the pumped liquid if it is free of solid particulate that could damage the shaft or sleeve surface, or packing itself. If the stuffing box is under vacuum, the liquid flush will prevent air from being drawn into the stuffing box where it would migrate to the lowest pressure area—the impeller eye which would cut off or reduce the pump flow. Note: It is assumed that the materials used in the stuffing box are compatible with the liquid being pumped.
2. *Floating seal rings*: Are only used in high pressure boiler feed pumps installed in central station electric power generating plants. This is a type of square ring packing except that it acts more like a labyrinth in that the rings float in a cold stream of cold condensate injected into the packing box near the box throat, with a maximum number of rings available to act as the pressure breakthrough path to minimize leakage out of the box. The injected liquid must be compatible with the pumped liquid, which in this case is treated steam condensate. Floating seals are less frequently used with the improvement in mechanical seal design and materials.
3. *Mechanical seals*: Provide as close to zero leakage as can be found with the exception of canned motor pumps and magnetic drive pumps. As there is a pressure breakthrough path across the seal faces, there is a minute amount of leakage to the atmosphere from the box. In some applications, any product leakage to the atmosphere might be too much, such as hydrogen sulfide (H_2S) gas which is entrained and is present in many crude oil and gas streams. H_2S is also present in liquids used to remove this contaminant from the crude oil or gas streams. These are termed sour services in refining or gas treating terminology.

Mechanical seals offer advantages, and disadvantages, over packing. As shown in Fig. 3.180, the rotating component of a mechanical seal may rub against the stationary component which could cause seal failure. Possible causes of rotating component rubbing against the stationary component include:

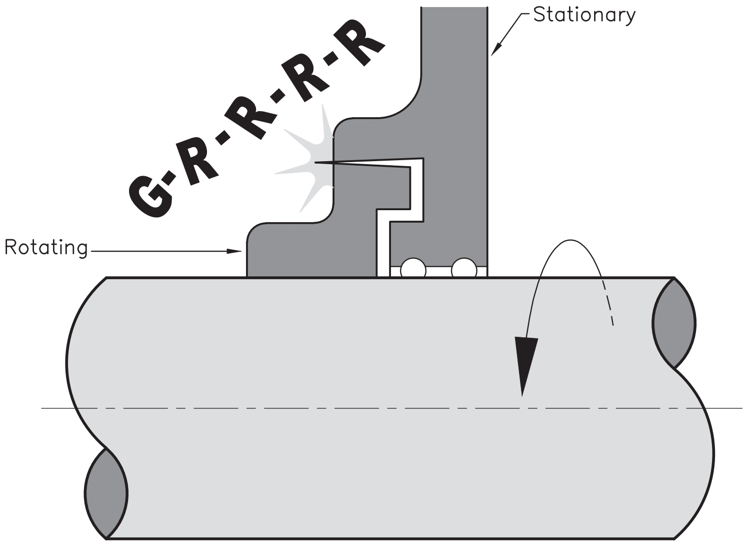


Fig. 3.180 Rotating component rubbing against the stationary component which causes mechanical seal failure.

- Stationary component warped—striking in one spot.
- Rotating component out of round—wear on stationary part.
- Shaft misalignment.
- Bent shaft or shaft deflection.
- Stationary component out of round—wear on rotating part.
- Could show up as a hot spot on thermographic view.

Fig. 3.181 shows the typical wear pattern for a nonleaking mechanical seal. If the seal is leaking one should examine the secondary seal. Common causes of mechanical seal failures include:

- *Coning*: Fig. 3.182 shows the effects of coning (negative rotation). Under high pressures there is little leakage. Coning is caused by overpressure or improper lapping.
- *Thermal distortion*: Fig. 3.183 shows the effects of thermal distortion (positive rotation). When stationary there is no leakage. Caused by thermal distortion or improper lapping.
- *Mechanical distortion*: Figs. 3.184 and 3.185 show the effects of mechanical distortion. With mechanical distortion the seal leaks steadily and the seal faces are usually not flat. One should check the stuffing box for over-torqued, check for squareness and flatness of box. Fig. 3.186 exhibits high spots at bolt locations. Due to over-torqued, change to softer gasket material.
- *High wear or thermally distressed surface*: Fig. 3.187 shows the effects of high wear or thermally distressed surface. The seal leaks steadily. Flashing and popping occurs. Carbon deposits on the atmosphere side. One should check the pressure, working height, cooling, and clearance.

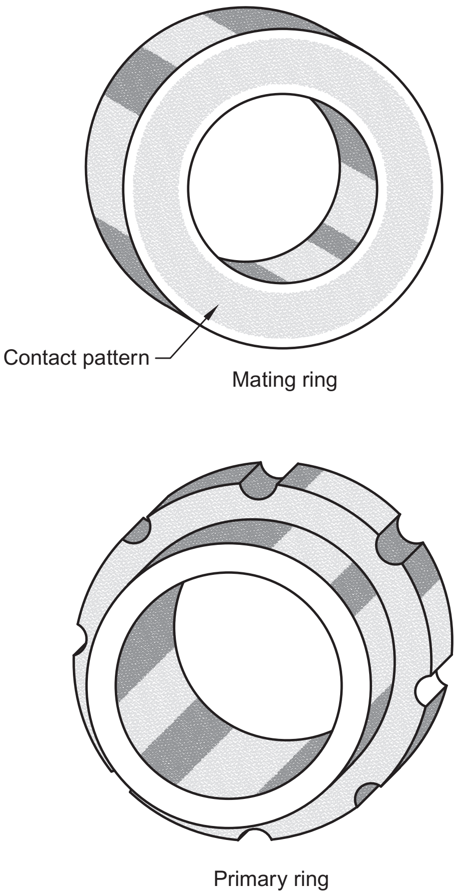


Fig. 3.181 Typical contact pattern for a nonleaking mechanical seal.

- *High wear and grooving*: Fig. 3.188 shows the effects of high wear and grooving. The seal leaks steadily. May be a result of inadequate cooling. One should check for abrasives and dead ended box.
- *Out-of-square mating ring*: Fig. 3.189 shows the effects of out-of-square mating ring. When rotating the seal leaks steadily. One should check the gland surface, drive pin extension, shaft alignment, and pipe strain.
- *Wide contact pattern*: Fig. 3.190 shows the effects of wide contact pattern. The seal leaks when rotating. One should check alignment, bearings, bent shaft, and pipe strain.
- *Eccentric contact pattern*: Fig. 3.191 shows the effects of an eccentric contact pattern. The seal may not leak unless the shaft is contacted. One should check for misaligned mating ring and shaft-stuffing box concentricity.

When replacing bearings do not use precision bearings or sealed bearings. Use C3 grade bearings as they are slightly forgiving. Use shielded bearings with a Zz code as they run cooler than sealed bearings.

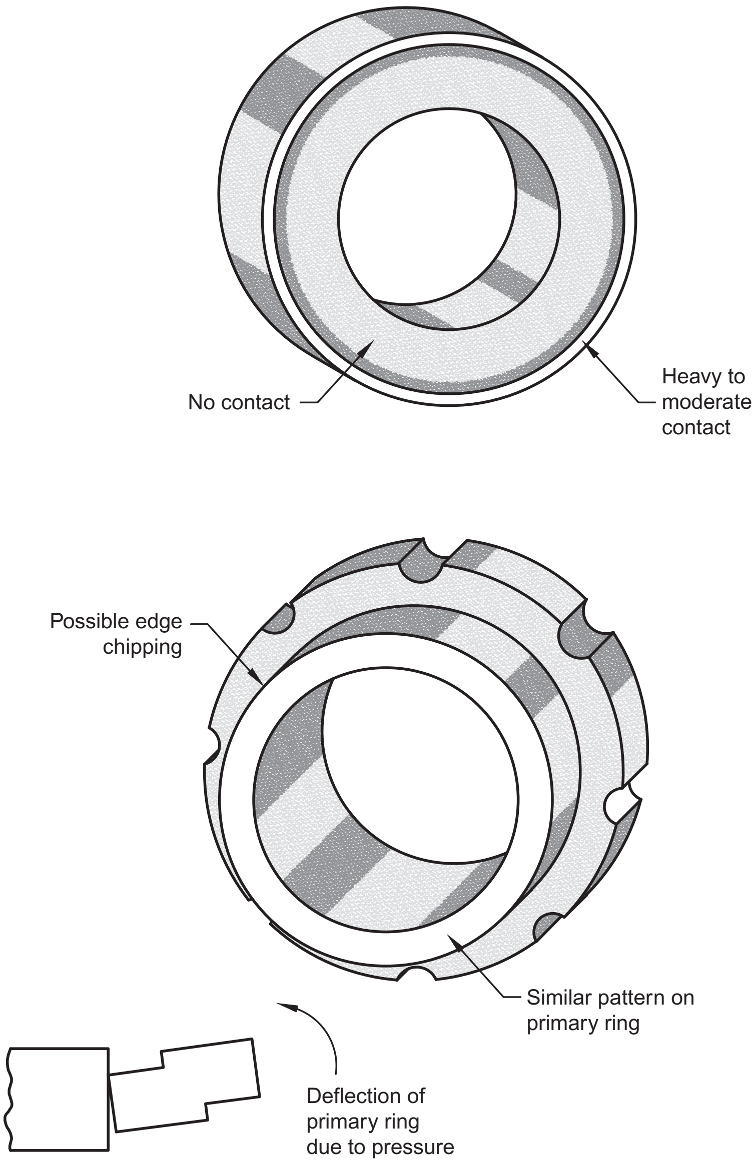


Fig. 3.182 Mechanical seal contact pattern due to coning (negative rotation).

3.17.2 Inadequate flow considerations

Fig. 3.192A and B are easy to follow troubleshooting methods of checking a pump and system for problems regarding inadequate flow. It is important to get the pumping system into proper design flow. All other parts of the system control system are engineered to function based on a given design flow. Thus the designer needs to

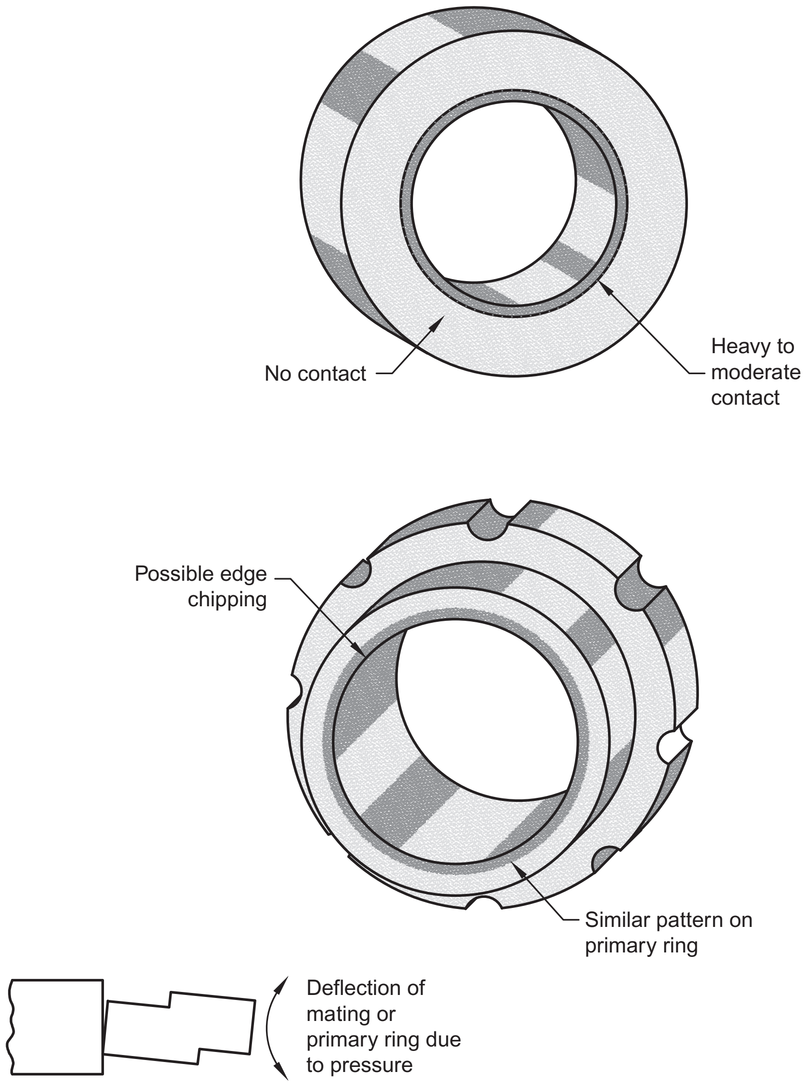


Fig. 3.183 Mechanical seal contact pattern due to thermal distortion (positive rotation).

balance flow in order for all components to work efficiently. The other basic problem is that the system is using excessive horsepower which is a waste of money due to the unnecessary horsepower usage.

3.17.3 Mechanical problems

Mechanical problems generally require shutting down the system to correct. If the system is vital to production, the costs of these types of problems can rise into tens of thousands or even hundreds of thousands of dollars when looking at lost production time.

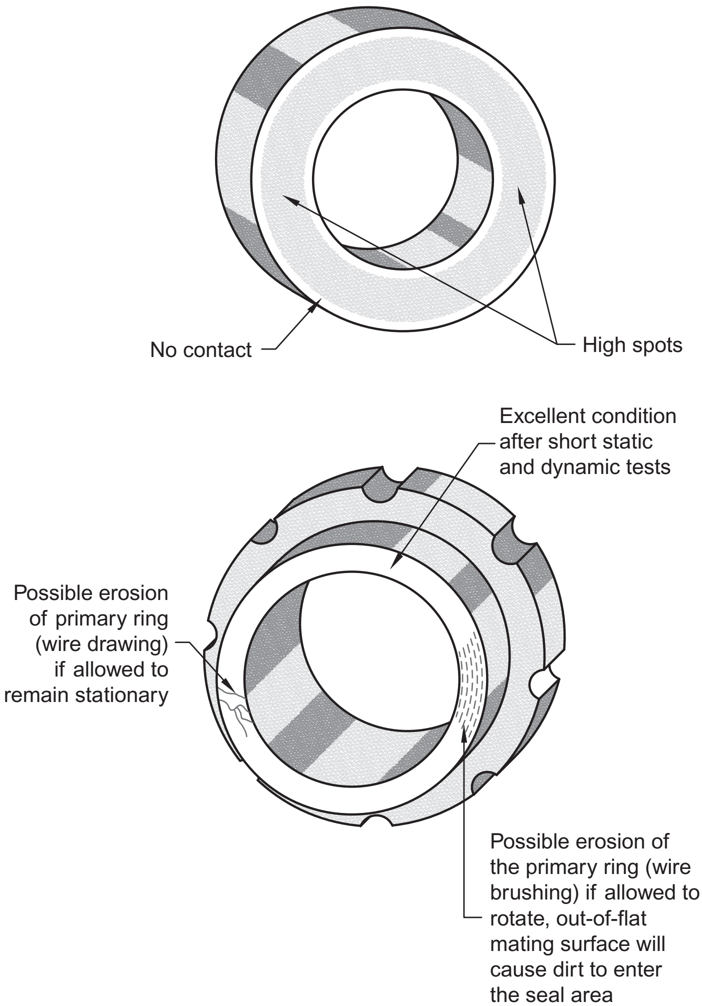


Fig. 3.184 Mechanical seal contact pattern due to mechanical distortion.

It is critical to increase the mean time between failure (MTBF) to an optimum level. Fig. 3.193 is an easy method for troubleshooting mechanical problems.

3.17.4 Motor overload

Motor overload is generally an indication that there is a pump system problem. It is important to check the electrical side of the system, but usually the problem can be located on the hydraulic side of the system. Fig. 3.194 can be used to troubleshoot the problem.

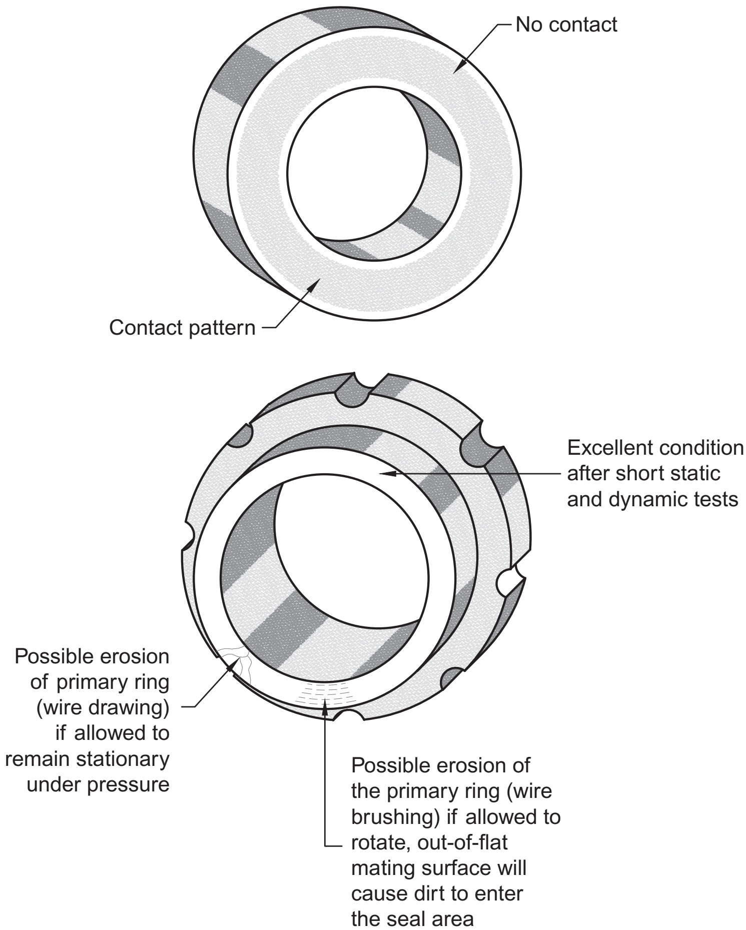


Fig. 3.185 Mechanical seal contact pattern due to mechanical distortion.

3.17.5 Systematic approach to troubleshooting centrifugal pumps

When troubleshooting it is important to use a systematic approach. Fig. 3.195 is a centrifugal pump troubleshooting flowchart which can be useful when troubleshooting centrifugal pumps.

The first step in troubleshooting is to collect the right data and understand it. Fig. 3.196 is a pump performance monitoring worksheet that can be used to collect useful data for performance evaluation. One must always work safely around all equipment and samples, if necessary. As soon as possible, one should discuss the problem with those personnel that witnessed the problem. Use all available resources, coordinate efforts, and stay organized.

The second step is to analyze the data. Apply engineering principles, resolve loose ends, and make sure "bad data" is not really something you do not understand.

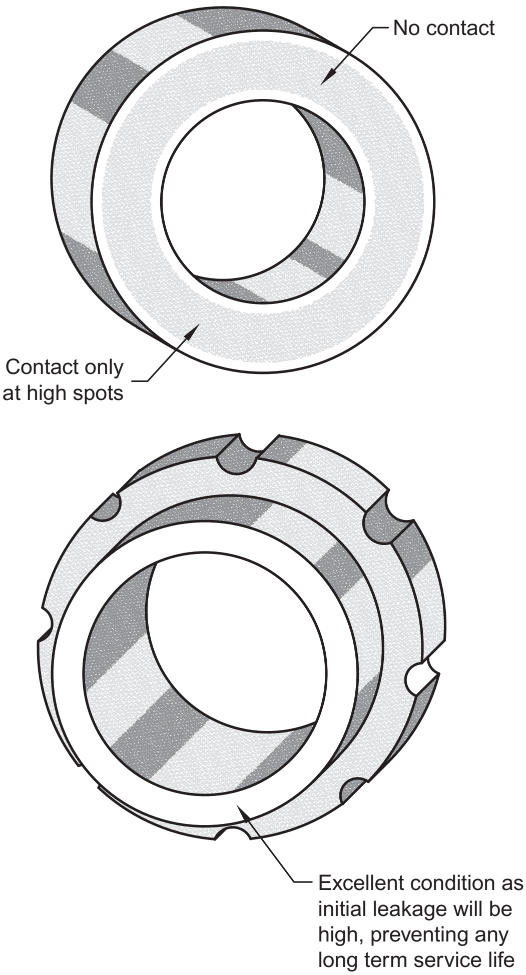


Fig. 3.186 Mechanical seal contact pattern due to mechanical distortion.

Identify potential causes. Involve appropriate resources and test to verify or eliminate specific causes. Identify solutions by involving appropriate resources/stakeholders. Evaluate the solutions by keeping in mind good engineering practices, time/money within resources, and will get organizational "buy-in."

Table 3.16 lists typical data error sources. Fig. 3.197 is an example of field data performance test tolerance boxes. The following data gathering errors need to be considered:

- Pressure measurement locations
  - Pick the right pressure gauge range.
  - Calibrate for critical work-being new is no guarantee the gauge is calibrated.
  - Be aware of gauge location effects:
    - Elevation
    - Pressure drops

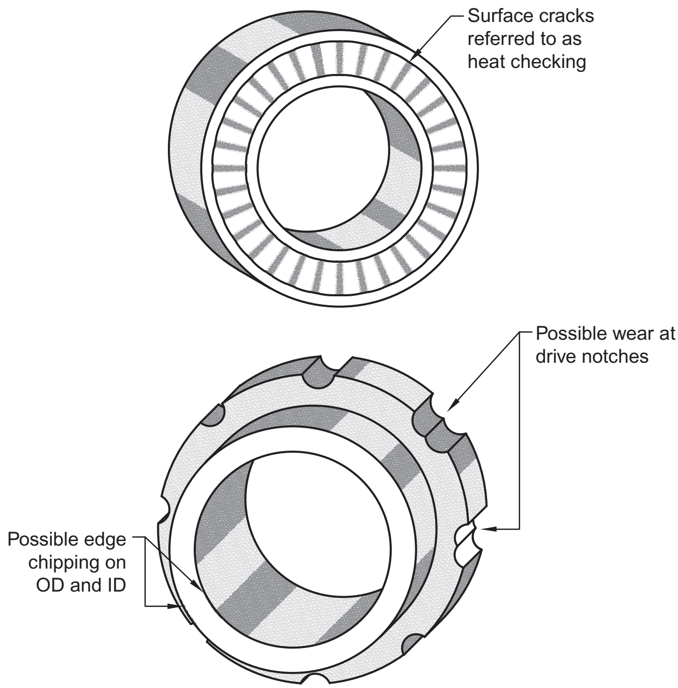


Fig. 3.187 Mechanical seal contact pattern due to high wear or thermally distressed surface.

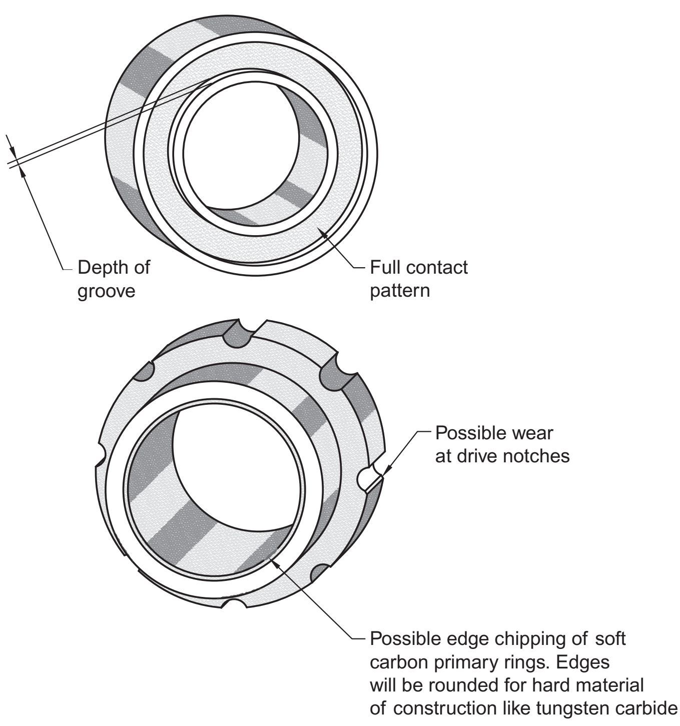


Fig. 3.188 Mechanical seal contact pattern due to high wear and grooving.

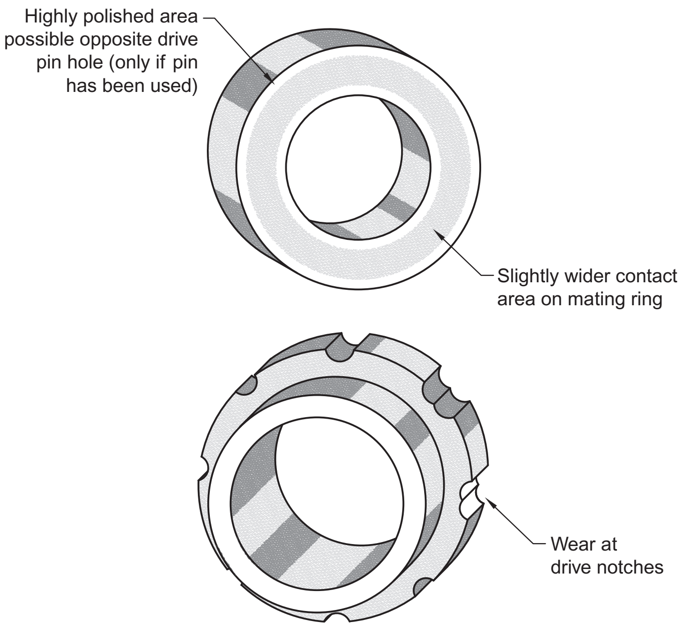


Fig. 3.189 Mechanical seal contact pattern due to out-of-square mating ring.

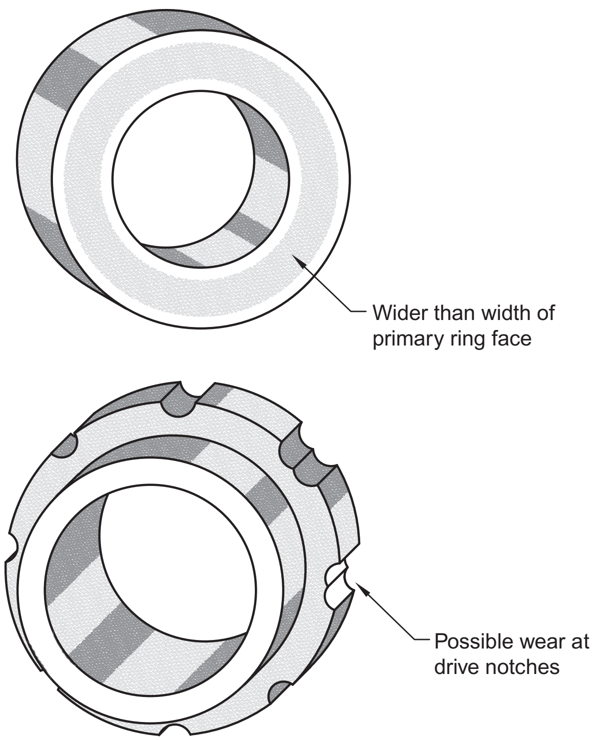
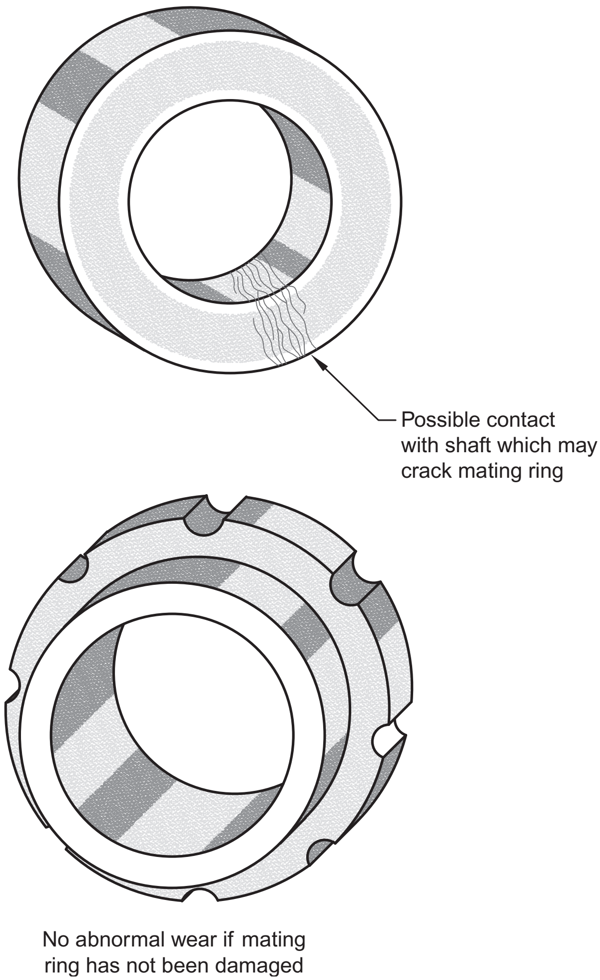


Fig. 3.190 Mechanical seal showing wide contact pattern.

Fig. 3.191 Mechanical seal showing eccentric contact pattern.



- Orifice flow meters
  - Typically gathered from distributed control system (DCS)
  - Must be temperature corrected
  - Confirm operating conditions are close to design conditions
- Specific gravity (SG) should be obtained for sample lab test for API gravity (refer to Fig. 3.198)
  - $SG = (141.5)/[(131.5 + {}^{o}API)]$
  - Must be corrected for pumping temperature
- Motor power
  - (1.73)(amps)(volts)(PF)(eff/746)

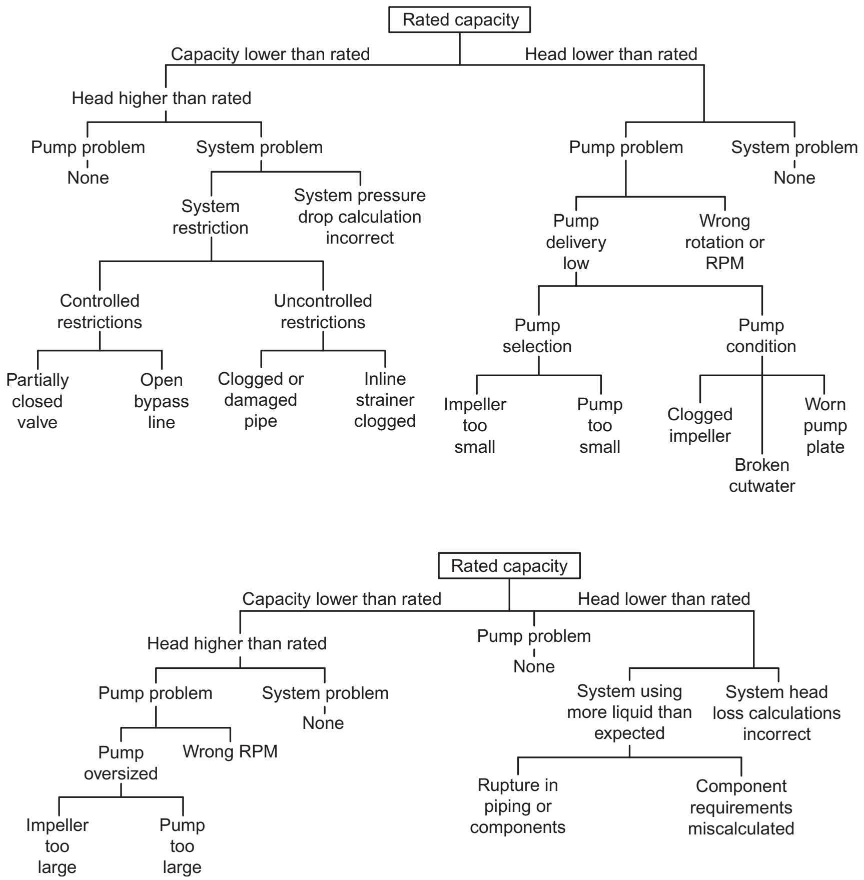


Fig. 3.192 Diagram to troubleshoot inadequate flow.

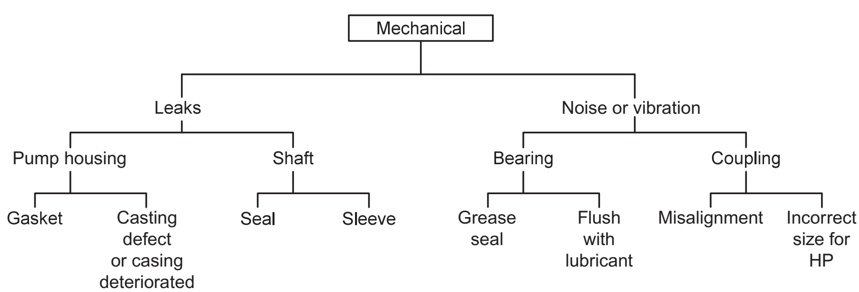


Fig. 3.193 Diagram to troubleshoot mechanical problems.

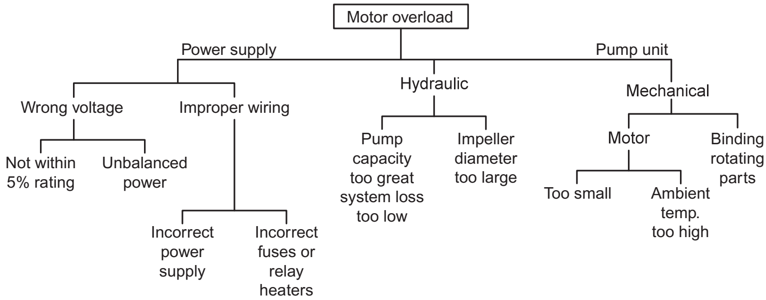


Fig. 3.194 Diagram to troubleshoot motor overload.

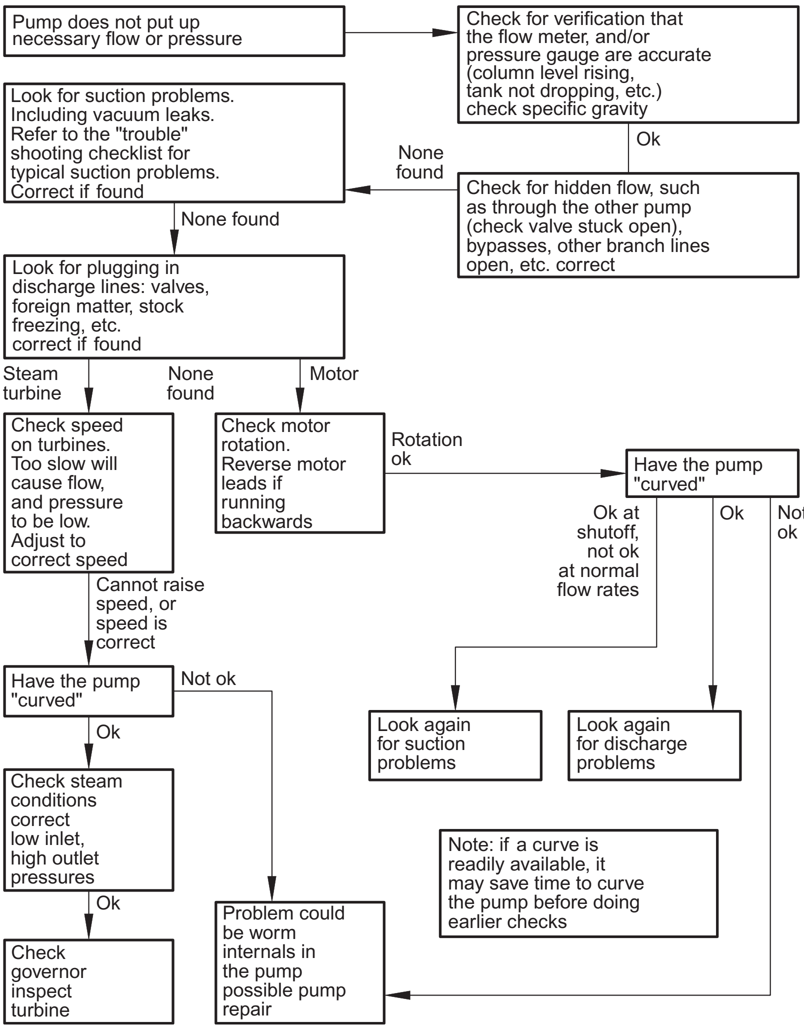
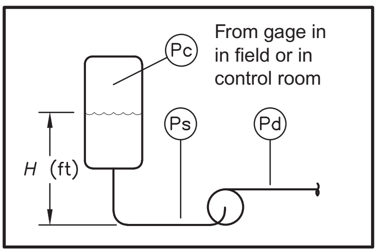


Fig. 3.195 Centrifugal pump troubleshooting flowchart.

PUMP NO. ________________________ SERIAL NO. ____________________
MFGR ____________________________ DRIVER SPEED _______ HP ________
TYPE & MODEL _____________________ PUMP CURVE NO. ______________
IMPELLER DATA ____________________

	SHUTOFF	<BEP	BEP	BEP>
1. Suction pressure, P_S (PSIG)				
2. Discharge pressure, P_D (PSIG)				
3. Differential pressure, $P_{DP} = P_D - P_S$, PSI				
4. Flow rate at 60°F in GPM, Q<B60±				
5. Liquid temp, °F				
6. Specific gravity at flow temp., SG_F				
7. Specific gravity at 60°F, SG_{60}				
8. Change P_{DP} (PSI) to H (ft) $H = \frac{P_{DP} \times 2.31}{SG_F}$				
9. Flowrate at temp. $Q = Q_{60} \times \frac{SG_{60}}{SG_F}$				
10. Hydraulic horsepower $HP_{HYD} = Q_F \times H \times SG_T/3960$				
11. Pump horsepower $BHP_P = HP_{HYD}/EFF_P$				
12. Motor horsepower $BHP_{MTR} = V_{AVG} \times I \times PF \times EFF_{MTR}/431$				



Conversion formulas:

$$SG_{60°F} = \frac{141.5}{API + 131.5}$$

PSIA = ATMOS. PRESS. + PSIG

$$P_S = P_D \frac{H \times SG_F}{2.31}$$

P_S and P_D can be PSIG or PSIA

GPM = BPD x .0292 = BPH x 07

Fig. 3.196 Pump performance and monitoring worksheet.

Table 3.16 Typical data error sources

Orifice flowmeter, direct tap reading (without S.G. error)	**1%**
Pressure gage (new)	2%
Compound pressure gage	2%
Tank level gage	16%
Specific gravity (S.G.)	2%
Motor amps	1%–4%
Steam turbine speed	0–?

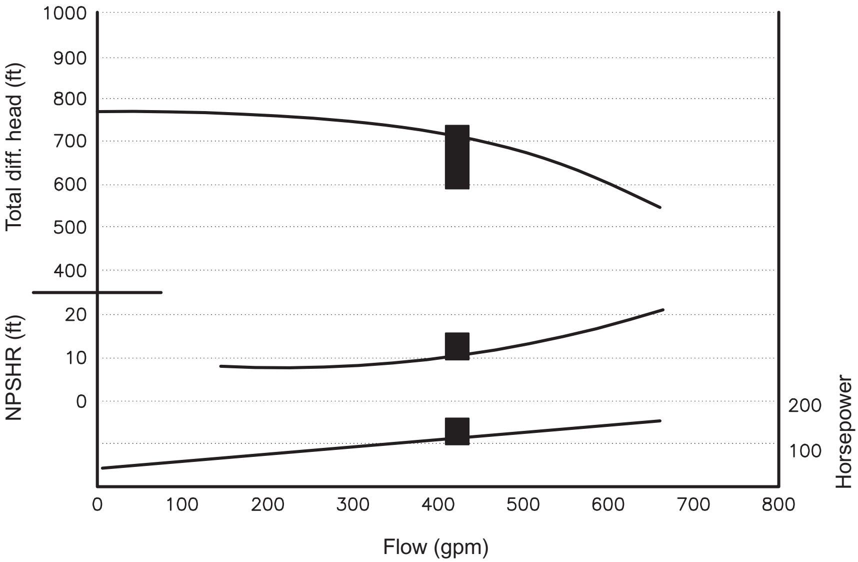


Fig. 3.197 Example of field data performance test-tolerance boxes.

Example 3.3. Determining pump performance (field units)
Given:

Pump "P1" is a crude unit bottoms pump that has been in service for over 5 years. There are suspicions that the pump is underperforming. You have been asked to evaluate its performance. The following data was obtained during a field test:

- API gravity of fluid being pumped: 23
- Suction pressure, measured near the pump suction: 28 psig
- Discharge pressure: 160 psig
- Flow rate (directly from the DCS): 28.5 KBPD
- Tower operating temperature: 590°F

You have also requested electrical assistance and have obtained the following information for the motor:

- Measured amps during test: 138 amps
- Voltage (est. at motor): 460 V
- Motor efficiency: 0.94
- Motor power factor: 0.90

In addition, you have been able to find an old copy of the pump performance curve and a copy of a volume expansion chart.
Determine:

(1) Calculate and plot the pump performance on the curve, including head, flow, and motor horsepower.
(2) Is the pump underperforming? If so, by how much?

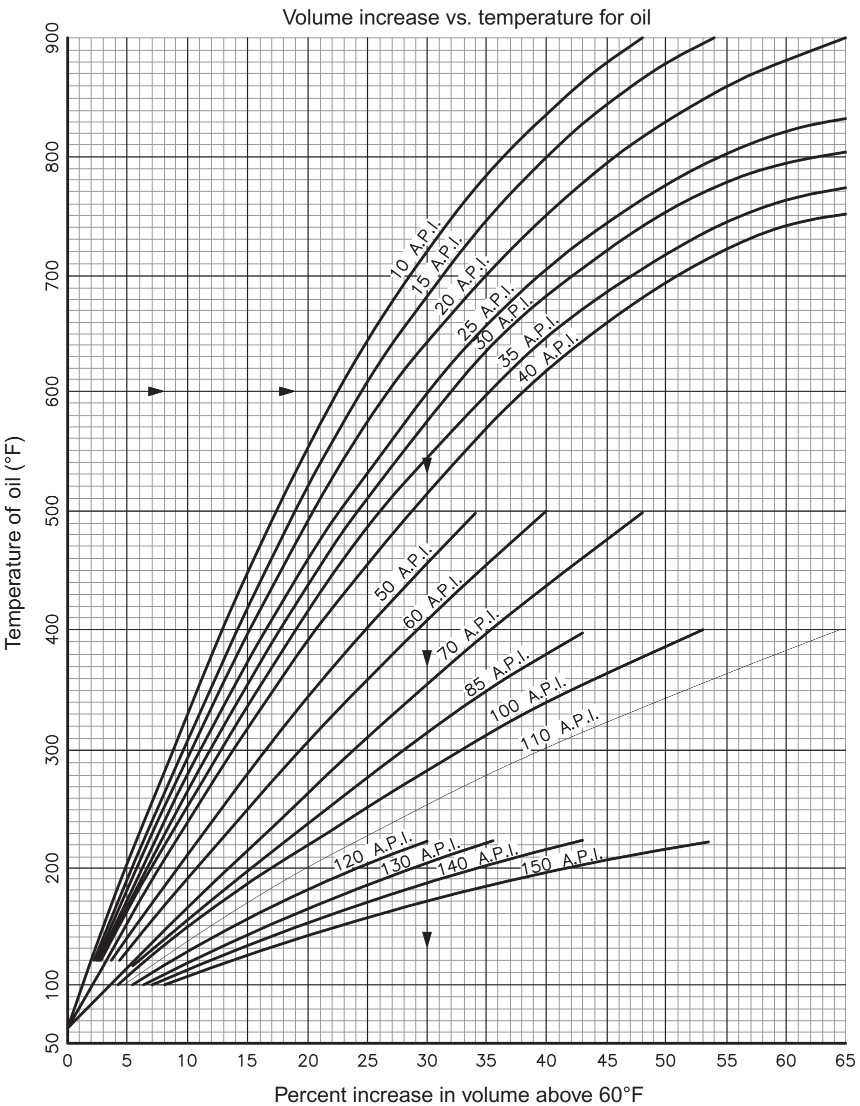


Fig. 3.198 Specific gravity temperature correction.

(3) Is the second head/flow curve drawn for a 13″ diameter impeller valid? How do you know?
(4) Would you repair this pump? What other things should be considered in this evaluation?

Solution:

(1) Refer to Fig. 3.199
(2) Yes, by 18%. Refer to Fig. 3.199
(3) Refer to Fig. 3.200

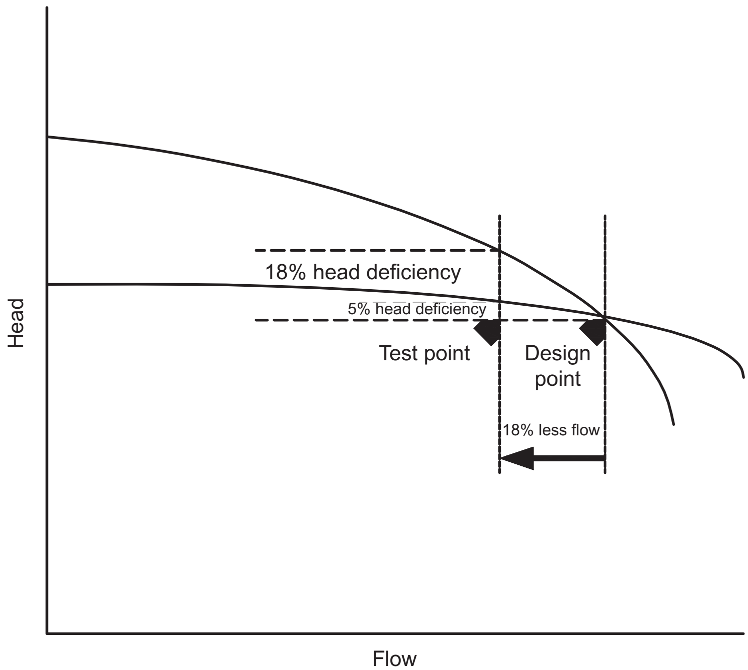


Fig. 3.199 Example 3.3 Pump performance deficiency initial conditions.

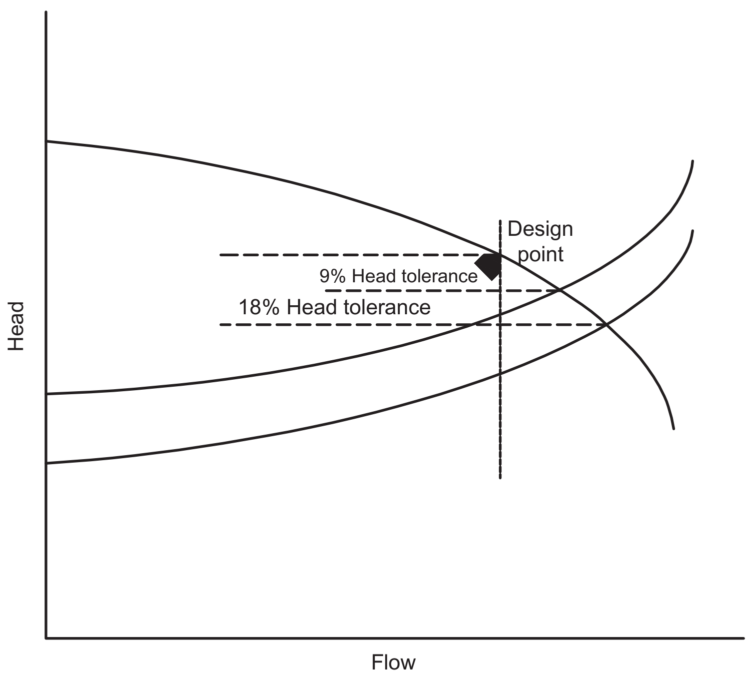


Fig. 3.200 Example 3.3 Performance deficiency—final conditions.

3.18 Start-up considerations

3.18.1 General considerations

Even after much engineering effort has been spent in pump selection, sizing, and design, more effort is required for proper start-up. Several steps must be accomplished before any centrifugal pump can be started. It is advisable to have the pump vendor present on start-ups, especially true for larger and more critical pumps. The designer should review maintenance procedures, ensure that instruction and part books are available, and ensure that the correct number of spare parts is ordered and delivered prior to start-up.

All piping should be thoroughly cleaned before start-up. All piping, including the pump casing, should be filled with the pumping water, properly vented, and then, and only then, the pump can be checked for proper rotation.

Running the pump dry can very quickly gall the wear rings, score the mechanical seal faces, and ultimately cause the pump to seize.

The pump should be started with the discharge valve at least partially closed. This prevents the pump from running off the performance curve and cavitating. Cavitation occurs when the absolute pressure in the suction eye of the impeller is reduced to a value that is equal to or less than the vapor pressure of the liquid. Liquid begins to vaporize under these conditions and the vapor bubbles are carried into the impeller. However, the higher pressure inside the impeller causes the bubbles to collapse resulting in noise, vibration, and a decrease in head and capacity.

Start-up procedures should always be well thought out and reviewed in considerable detail when the pumped fluid is hydrocarbon or when the pump is in cryogenic service. Flashing can be a significant design factor in these services.

3.18.2 Checklist for a centrifugal pump

- Check that vendor instruction books, drawings, outline/elevation, sectionals, and internal pump clearance data on hand.
- Review availability of critical spare parts (as applicable).
- Loosen all bolts connecting the suction discharge piping to the pump. Note any binding of any bolt and springing or movement of the piping. Correct any deficiencies prior to start-up. This is an ounce of prevention which is a lot cheaper than the pound of cure. If you do not have time to do it correctly in the beginning, when will you have time to do it correctly at all?
- Break the coupling between the pump and its driver. Check for smoothness of rotation. Each component should be rotated three or four complete revolutions. Bump the driver to check rotation.
- Review alignment of the driver to the pump and correct if necessary.
- Check that flange gaskets are installed.
- If strainers are installed in the suction, verify metallurgy, mesh size, and properly oriented cone strainer “witches hat” with point pointing upstream. Be sure pressure gauges are installed upstream and downstream of the strainer.
- Check to see that safety requirements and company approved coupling guard is installed.

- Verify that the pump is lubricated with proper lubricant or connected to an operating oil mist lubrication system.
- Verify that seal piping has been cleaned and seal points, if any, are filled with proper buffer liquid. Check that seal instrumentation is functioning and correct.
- Check that there are suction and discharge pressure gauges installed and that they are of a suitable range for the system.

3.19 Operations and maintenance considerations

3.19.1 Background

Centrifugal pumps are dependable and can be long lasting with proper operation and maintenance. They can tolerate some internal corrosion and erosion without substantially decreasing performance. They are quiet, need little attention, and generally operate pulsation free. They can be easily disassembled and have few parts that require tolerances.

Shaft alignment is an extremely important process in the installation of centrifugal pumps. It should also be considered in the ongoing service of the pump within a broader program specifically a predictive/preventive maintenance program.

These types of programs save millions of dollars a year for companies who understand how to use them. It is not the purpose of this text to provide any information on these programs other than basic technologies/techniques that are available. When used properly, these can help predict when a component will fail, given indication that some sort of operational maintenance is required, and even give clues to help troubleshoot failures.

3.19.1.1 Predictive maintenance

3.19.1.1.1 Vibration analysis

Vibration analysis involves using a vibration sensitive transducer and instrumentation to measure and record the vibration characteristic of a rotating machine. Baseline data can be collected and recorded so that trends can be tracked or problems that have developed can be compared to this and analyzed. It can then be used for troubleshooting, root cause failure analysis, as well as quality control issues.

3.19.1.1.2 Thermography

Thermography is the use of infrared thermal imaging devices to observe and measure temperature gradients in equipment and structures. A temperature gradient is an area of uneven temperature and can be seen in a thermograph as areas of different colors. There are many applications of thermography in industry, but for pumps specifically, it can be used to analyze couplings between pumps and motors, motor component temperature, and pump component temperature.

3.19.1.1.3 Oil analysis

Oil analysis is a series of chemical tests that are performed on samples of lubricating oil to determine its condition and whether in this condition it can perform its lubricating functions. The analysis can point out a problem with the lubrication itself or a problem with the machine in which it is being used. Samples of oil are taken by operators and/or maintenance personnel and then shipped to a third-party laboratory for detailed testing.

3.19.1.2 Preventive maintenance

Planned maintenance of pumps and their systems should be done accordingly to facility guidelines. Pumps are generally self-reliant with few preventive maintenance procedures required. Lubrication of bearings, adjustment of packing, and cleaning of strainers are a few of these that if done regularly and properly can add significant life to a centrifugal pump.

3.19.2 Pump operating problems

3.19.2.1 Low or irregular discharge rate

Low or irregular discharge rates can occur because of improper purging prior to start-up but will also happen if the pumps lack the required $NPSH_R$ to prevent fluid from vaporizing at the impeller suction.

3.19.2.2 Excessive discharge pressure

If the discharge pressure is higher than normal, then the problem is probably downstream from the pump (e.g., a partially closed discharge valve or some other blockage). If the discharge pressure is lower than normal, or if it fluctuates between a low and normal reading, the problem is likely to be found on the suction side of the pump (e.g., inadequate NPSH or blockage in the suction line).

3.19.2.3 Loss of prime

A *common problem* that can be *caused by*:

- Incomplete venting
- Blocked suction line, or
- Entrained gases in the liquid

Entrainment can be *caused by*:

- *Vortexing* in the suction vessel
- *Air leaks* through joints, valves, and packing
- *Flashing* of low boiling liquids

Often, the pump can be reprimed while in operation by cracking the vent valve on the top casing and allowing vapor to escape. However, venting may be only a temporary remedy if there is inadequate $NPSH_A$ at the pump suction. Level, pressure, and temperature conditions at the suction source must be checked to ensure that they are within the parameters which the pump was designed to handle.

3.19.2.4 Vibration and overheating

Seal and bearing failures are common mechanical problems. If a pump is continually plagued by seal and/or bearing failure, and the designs have been thoroughly checked, the failures may well be caused by incorrectly installed piping supports and anchors. A piping arrangement that fails to allow for thermal expansion will likely contribute to seal and bearing failure. Seals may also fail if the liquid being pumped is hotter than normal. The heat is conducted along the pump shaft subjecting cooler parts of the pump to thermal stresses which cause seals to crack or shatter. Seals can fail due to inadequate flow of seal oil or jacket water (if the pump is water jacketed).

3.19.2.5 Other common mechanical problems

Other common mechanical problems include:

- Shaft misalignment
- Worn or loose couplings
- Unbalanced (i.e., misaligned) impellers
- Loose pipe supports
- Loose drive belts

3.19.3 Preventing pump seizures

When internal rotating elements of a centrifugal pump heat up and expand due to friction with stationary parts, and this expansion equals the total running clearance between these parts, the stationary parts will stop the unit from rotating. Seizure may occur:

- In new or newly reconstructed pumps with exceptionally close tolerances
- In pumps with inadequate flow to dissipate heat
- When air or vapor gets into the pump casing during normal operations
- When stationary and rotating elements have materials with galling tendencies
- During improper operation such as when pumping against a closed discharge valve, pumping against no backpressure, running at a critical speed, or when pumping solids
- As a result of faculty construction or parts replacement

Preventing seizures during design means avoiding materials for wear rings and throttle bushings that exhibit even moderate galling tendencies unless a hardness differential has been specified. Suction strainers should also be installed to capture particles that would otherwise find a way into close tolerance spaces.

Seizures during commissioning and start-up can be prevented by establishing and observing proper start-up procedures. These should always include:

- Closing of the discharge valve
- Opening the suction valve

- Opening the pump casing vent valve, and
- Filling the suction line and casing with fluid
- Then partially opening the discharge valve
- Starting the pump, and
- Slowly opening the discharge valve to a full open position

3.19.4 Motor overloads

Any one or a combination of the following can overload the pump motor:

- Thermal expansion of moving parts due to poor cooling or lubrication
- A reduction in discharge pressure accompanied by a high flow rate
- An increase in the specific gravity of the fluid being pumped
- A change in fluid viscosity
- Installation of a larger impeller

3.19.5 Vibration

All rotating equipment vibrates to some degree. Sometimes a preventive maintenance shutdown is dictated to prevent more costly, larger failures.

Vibration often happens as a result of misalignment. When checking alignment one should:

- Take a hot reading immediately after the pump is turned off
- Compare the reading to a cold reading taken once the pump has cooled to determine the effects (if any) of thermal growth on the alignment of the pump with the driver

If the pump is handling a hot liquid, thermal growth could have a significant effect on alignment. Vibration also happens as a result of the following (refer to Table 3.15):

- Bent pump shaft
- Coupling wear or looseness
- Impeller damage
- Loose piping supports
- Cavitation
- Inadequate NPSH

As tolerances in centrifugal pumps are often critical, vibration frequencies and amplitudes can warn the designer about impending trouble. Fig. 3.201, vibration severity chart, can be used to help determine impending trouble (Table 3.17) (see Fig. 3.202).

3.19.6 Bearing failure

The purpose of pump bearings is to support the rotating assembly, position the assembly, and absorb the loads imposed. In standard lines of pumps, the bearings are properly sized and selected for the anticipated loads by the design engineer. Only when the pump is operated well away from the best efficiency point (BEP) do loads become excessive and bearing failures occur more frequently.

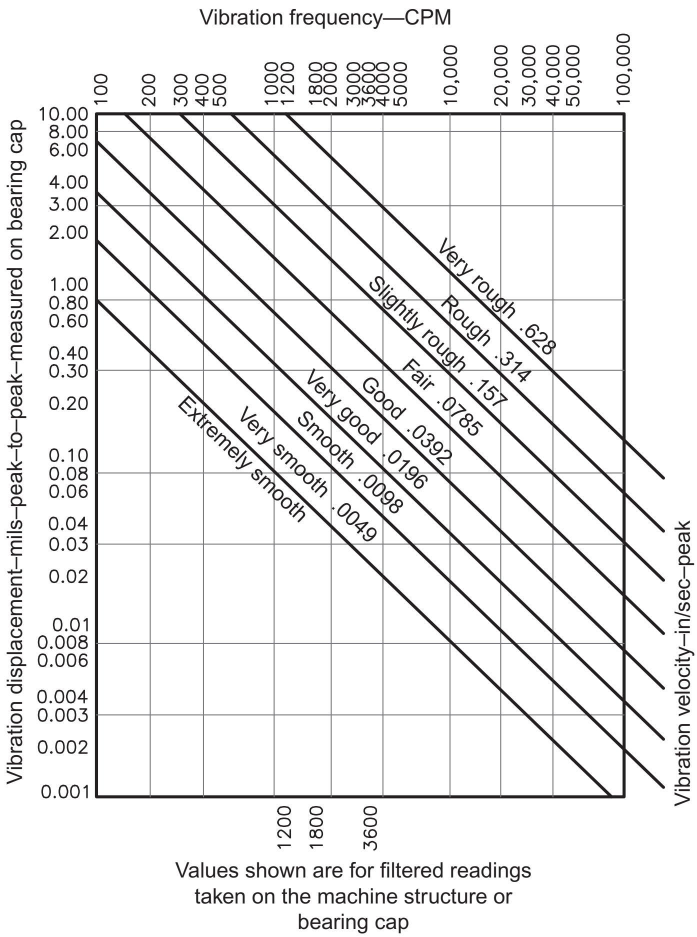


Fig. 3.201 Vibration severity chart.

Bearings should be selected with an L10 rating of at least 25,000 h continuous operation, and at least 16,000 h maximum loading and rated speed. This is a 3- and 2-year continuous operation. API 610 criteria call for a minimum of 3 years continuous operation without maintenance. This is achievable in a well-engineered system.

The bearing housing has three functions:

- To support the rotating assembly of the pump.
- To absorb the loads transmitted by the bearings.
- To provide a reservoir for the bearing lubricant.

The reservoir must be sealed to keep out the contaminants. The least costly seal is a spring-loaded device which mounts in the cavity of the bearing housing covers. The disadvantage of this type seal is that the lip, in time, will score the shaft, making it nearly impossible to seal the housing properly. A more effective way to seal the

Table 3.17 Relationship between frequency and cause of vibration

Frequency	Amplitude	Cause	Remarks
1 × rpm	Largest high-frequency vibration near the bad bearing	Unbalance	
1 × rpm normally	Axial direction vibration 50% or more of radial	Misalignment of coupling or bearings and bent shaft	Easily recognized by large axial vibration. Excessive flange loading can contribute to misalignment
Very high, several times rpm	Unsteady	Bad antifriction bearings	Largest in radial direction. proportional to unbalance
2 × rpm		Mechanical looseness	Check grouting and badplate bolting
1, 2, 3, and 4 × rpm of belts	Erratic or pulsing	Bad drive belts	Use strobe light to freeze faulty belt
1 or 2 × synchronous frequency	Disappears when power is turned off	Electrical	3600 or 7200 CPS for 60 cycle current
No. of impeller vanes × rpm		Hydraulic forces	Rarely a cause of serious vibration
Extremely high, random		Cavitation	

bearing housing is by using a labyrinth-type seal. This employs a rotating member to the shaft, while rotating in a stationary member mounted in the bearing housing cover.

Bearing failure may be caused by water or product contamination in the bearing housing. High suction pressures may create unusual thrust loads on bearings. The designer needs to make certain that thrust bearings are installed in the correct position. Coupling misalignment can also cause bearing failure. Although misalignment always creates vibration, it may not be noticeable, especially at very low speeds or in a particularly rigid system.

3.19.7 Mechanical seal failure

Mechanical seal failure accounts for the majority of pump failures. The two mating faces in the seal must be aligned, well lubricated, and cooled if necessary. When lubrication is lost, the seal faces run dry, overheat, and eventually become scored. If the liquid being pumped contains abrasive materials, particles can force the seal faces apart or score the faces. Even small amounts of scoring can cause leaks.

Additional causes of seal failure include:

- *Overheating*—destroys O-rings and gaskets
- *Material break down*—occurs from chemical incompatibility with the liquid being pumped, for example, O-rings that are not compatible with the liquid being pumped can expand or become brittle and eventually break
- *Mishandling and poor installation practices*—can also cause premature seal failures, seal faces are easily scratched and O-rings and bellows can be easily cut on sharp edges

Many seal problems can be diagnosed by examining the wear pattern on the stationary seal face. A normal wear pattern on the stationary seal appears as an even ring of wear that matches the width of the rotating seal face. The pattern should be continuous and centered on the face. A narrow wear pattern could indicate that the seal faces are not flat or parallel to each other. A wear pattern that is wider than the rotating seal face could indicate a bent shaft or worn bearings. An even, but off center, wear pattern could mean that the rotating ring is not properly positioned against the stationary ring. If the pattern is uneven (i.e., if there are gaps), the seal face could be distorted. This may be caused by overheating or by overtightening the gland. A corroded or pitted seal face can indicate a chemical compatibility problem. Chipped seal faces could indicate the seal was mishandled or that foreign material was trapped in the seal. A grooved appearance is usually caused by foreign material trapped between the seal faces. Hairline cracks in metallic seal faces indicate overheating.

3.19.8 Condition of pumped fluid

A dirty, corrosive fluid will cause rapid extreme wear, creating costly, inefficient operation. Studies reveal that normal centrifugal pump wear occurs when the solid content of the pumped fluid remains below 1.00 lb/1000 barrels.

3.19.9 Operating efficiency

The operating efficiency is a method of establishing if a problem exists. It compares actual operation to design for a given set of conditions. For example, the capacity, operating head, and horsepower must be corrected to the design speed values before a comparison can be made to manufacturer's data. These conditions may be made as follows:

$$GPM_{corr} = GPM_{opn}\left(\frac{NDesign}{NOPN}\right) \tag{3.24}$$

$$TDH_{corr} = TDH_{opn}\left(\frac{NDesign}{NOPN}\right)^2 \tag{3.25}$$

$$BHP_{corr} = BHP_{opn}\left(\frac{NDesign}{NOPN}\right)^3 \tag{3.26}$$

3.19.10 Centrifugal pump data required for specification

Fig. 3.202 summarizes the data required to select a centrifugal pump.

Centrifugal pump data required for specification		
	Information required by pump manufacturer which sets pump size	
	Flow (total flow for facility and flow for each pump)	______
	Head	______
A.	NPSH available	______
	Liquid pumped	______
	Pumping temperature	______
	Pumped fluid = corrosive/erosive	______
	Specify construction grade	
B.	API 610	______
	ANSI B73.1	______
	Other	______
	Specify pump type	
	Barrel	______
	Multistage splitcase	______
C.	Vertical	______
	Sump pump	______
	Inline	______
	API-single stage	______
	Specify major construction features	
	1. Materials of construction	
	Case	______
D.	Shaft	______
	Impeller	______
	Packing gland/mechanical seal	______
	Gland	______
	2. Sealing system	
	Mechanical seal	______
	API seal code	______
	Seal piping plan	______
	Packing	______
		______
		______
	3. Bearing type	
	Manufacturer's standard	______
	Rolling contact	______
	Hydrodynamic	______
	* Unless specific pump application dictates otherwise, bearing type should be manufacturer's standard	
	Specify major construction features (Cont'd)	
	4. Pump baseplate	
D.	Common baseplate, pump skid and driver	______
	Drain pan	______
	Skid material	______
	Skid leveling screws	______
	5. Coupling type	______
	Specify testing	
E.	Performance (witness/nonwitness)	______
	Hydrostatic (witness/nonwitness)	______
	NPSH (witness/nonwitness)	______
	Driver type	
F.	Electric motor	______
	Engine	______
	Other	______

Fig. 3.202 Centrifugal pump data required for specification.

Example 3.4. Centrifugal pump selection (field units)

Given:

An offshore production facility processes 60,000 BOPD of a 19° API crude oil. The produced oil is pumped from a storage tank by a charge pump. This charge pump delivers oil through a metering skid to the main shipping pumps. The inlet pressure to the shipping pump is 205 psig. Crude oil pumping temperature is 80°F.

The shipping pump for the previous facility shall be a centrifugal. The pump must meet the following criteria:

1. Maximum motor size that can be installed on the pump is 200 HP. (Platform generator is not capable of starting a larger motor.)
2. Discharge pressure of pump is to be 585 psig.
3. Platform space is at a premium.
4. Pump is to be direct driven by an electric motor turning 3600 rpm.
5. Pump components and auxiliaries are to be in accordance with safe practices for offshore platforms.

Determine:

1. Approximate number of pumps.
2. Pump configuration.
3. Head and flow for each pump/
4. Pump type.
5. Mechanical features of the pump:
   - **(a)** Seal system with API code
   - **(b)** Bearings
   - **(c)** Material by ASTM number for this application, as recommended by API-610, for the following parts: (1) pressure casing, (2) impeller, and (3) shaft.

Solution:

1. Determine approximate number of pumps.

Total hydraulic horsepower required

$$HP = Q\frac{\Delta P}{58,756}$$

Where:

$Q = 60{,}000$ BPD
$\Delta P = 585–205–380$ psi

$$HP = \frac{(60,000)(380)}{58,756} = 388HHP$$

Assuming pumps are selected with an average efficiency of 0.7

$$HP \approx \frac{388}{.7} = 554HHP$$

∴ Number of pumps required at 200 HP each is

$$\frac{554HP}{200HP/pump} = 2.8 \therefore Use 3 pumps$$

2. Determine pump configuration.

Since 3 pumps are required, the basic pumping configuration must be established.

Option I: Three pumps operating in parallel

This configuration will require three identical pumps. Thus, if one pump is down, pump capacity will be reduced by approximately 33%. Additionally, spare parts for each pump are identical.

Option II: Two pumps operating in parallel with one pump in series

This configuration will require two different pumps. The final pump will have larger impellers and case than the first-stage pumps. This will require additional spare parts. Additionally, if one pump is down, pump capacity is reduced by 50%–100%.

Option III: Three pumps operating in series

This configuration will require three identical pumps. Each pump will be selected with the capability to pump the design flow rate at 33% of the rated differential pressure. If one pump is down, two pumps operating in series will be able to reach the required discharge pressure, but the flow rate will be reduced considerably. Spare parts for each pump will be identical.

Option IV: Four pumps operating in series or parallel

This is similar to Options I and III as far as pump selection is concerned. The only difference is one standby pump is provided so capacity can be maintained if one pump is out of service.

Selection

Only Option I and III will be considered. The selection will depend on pump system interaction curves.

Option I:

3. Determine head and flow for each pump

$$S.G = \frac{141.5}{131.5 + 19} = 0.94$$

$$\text{Head} = \frac{2.31\Delta P}{S.G} = \frac{2.31(585 - 205)}{0.94} = 933\,\text{ft/pump}$$

$$Flow = 60{,}000BPD \times \frac{1}{3} pumps \times .0291 \frac{gpm}{BPD} = 583 gpm$$

4. Pump type

A review of the centrifugal pump types shows that the following types are available:

(a) Horizontal single-stage overhung
(b) Horizontal two-stage overhung
(c) Horizontal single-stage, impeller between bearings
(d) Horizontal multistage
(e) Donut-type pump
(f) Vertical multistage
(g) Regenerator turbine
(h) High speed

Types (a), (c) and (h) can be eliminated as these pumps are single-stage pumps. A multistage will most likely be required.

Types (e) and (g) are pumps that are not designed for applications such as this.

Types (b), (d), and (f) are all suitable for application. Types (b) and (d) are horizontal pumps with horizontal electric motors, while type (f) is a vertical pump with a vertical motor.

Since the vertical pump will take up considerably less space than the horizontal style pumps it is the one that would probably be preferred for this offshore application.

5. Mechanical features of the pump

(a) Seal system. Per seal usage guide, a balanced tandem seal should be provided. Since the fluid pumped is not extremely hot and not sour, the 0″ ring material can be Teflon or Kalrez, fourth letter G or I.

Since crude is liable to contain some sand or other abrasive, for example, products of corrosion, the rubbing surfaces of the seal, the fifth letter should be at least an L. The total seal system is thus BTTGL with API seal piping plan 12 or 13.

(b) Bearings—Manufacturers standard. Bearings should have both a radial and thrust capacity.

(c) The material class from Table F-1 is S-1. Therefore the following material should be specified:

(1) Pressure casing	Carbon steel
(2) Impeller	Cast iron
(3) Shaft	Carbon steel

From Table E-2 the following conversions can be made:

(1) Pressure casing	ASTM-216
(2) Grade WCA or WCB	
(3) Impeller (forging)	ASTM-105
(4) Shaft (bar stock)	ASTM -576
(5) Grade 1015	

Option III

6. Head and flow

$$\text{Head} = \frac{2.31(585 - 205)}{3 \times (.94)} = 311\,\text{ft/pump}$$

$$\text{Flow} = 60{,}000BPD \times 0.291\frac{\text{gpm}}{BPD} = 1750\,\text{gpm}$$

7. Pump type

This type of pump should be identical to that of Option I.

8. Mechanical features of this pump should be identical to those of option I.

3.20 Exercises

1. Energy is imparted to the fluid in a centrifugal pump by the
- **(a)** thrust bearing
- **(b)** shaft
- **(c)** impeller
- **(d)** stuffing box
- **(e)** radial bearing

2. A (n) _______ is a graphical representation of how much head is required to pump various capacities through the piping system.
- **(a)** performance curve
- **(b)** system curve
- **(c)** affinity chart
- **(d)** process flow diagram
- **(e)** P&ID

3. Using Fig. 3.203, identify the single-stage overhung centrifugal pump components.
- **(a)** ____
- **(b)** ____
- **(c)** ____
- **(d)** ____
- **(e)** ____
- **(f)** ____
- **(g)** ____
- **(h)** ____
- **(i)** ____
- **(j)** ____

4. The performance of centrifugal pumps decreases as the viscosity of the pump fluid ____

5. Operating a centrifugal pump at reduced capacities may lead to_________
- **(a)** higher bearing loads
- **(b)** temperature rise
- **(c)** operation at less than BEP
- **(d)** internal recirculation
- **(e)** all of the above

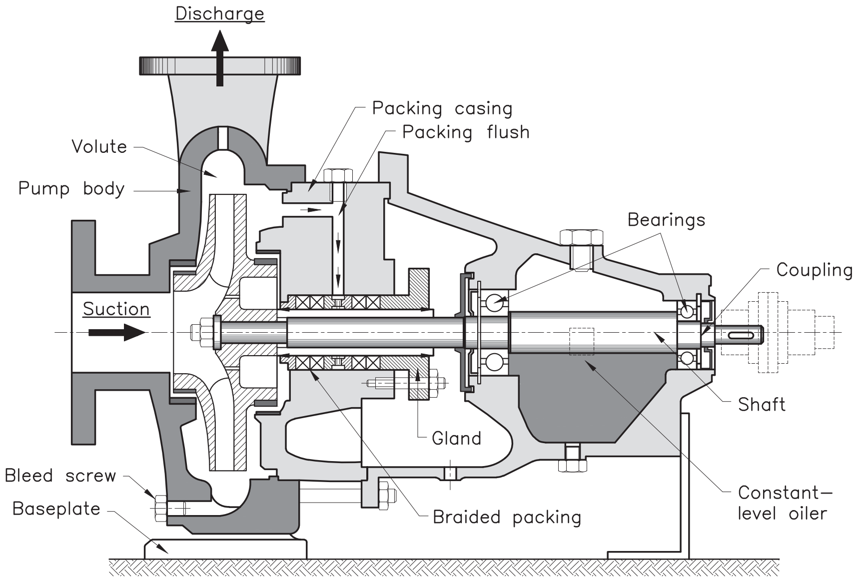


Fig. 3.203 Exercise 3 Single-stage overhung pump.

Questions 6–9

Using Fig. 3.204 and the following data:

Total head: 190 ft
Pump capacity: 150 gpm

6. What is the pump efficiency?
   (a) 31%
   (b) 43%
   (c) 45%
   (d) 82%
   (e) 95%
7. What is the minimum required driver horsepower of the pump for the design conditions?
   (a) 7-1/2hp
   (b) 10 hp
   (c) 15 hp
   (d) 20 hp
   (e) 25 + hp
8. What is the $NPSH_R$?
   (a) 7 ft
   (b) 18 ft
   (c) 24 ft
   (d) 27 ft
   (e) 30 ft

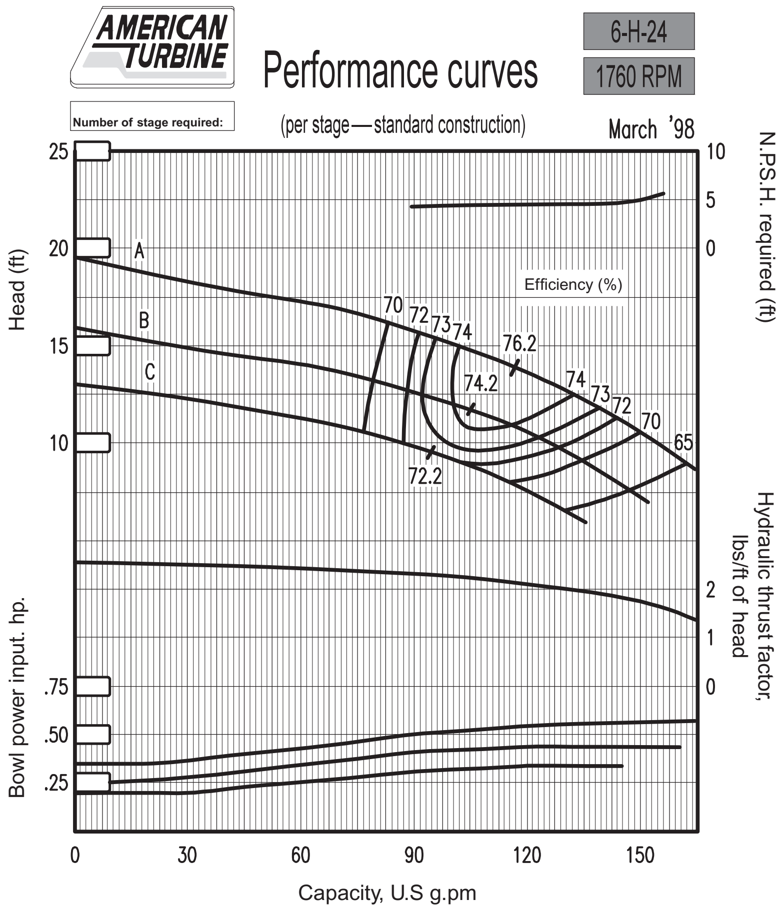


Fig. 3.204 Exercise 6.9 Pump performance curves.

9. What size impeller is recommended?
- **(a)** 6 in.
- **(b)** 6-1/2 in
- **(c)** 7 in.
- **(d)** 7-1/2 in.
- **(e)** 8 in.

10. If a centrifugal pump has either a discharge block valve that may be inadvertently closed or an automatic closing discharge valve a ________________ is recommended to prevent excessive heat buildup.
- **(a)** check valve
- **(b)** drain valve
- **(c)** bypass valve
- **(d)** self-priming pump
- **(e)** belt coupling

11. According to the affinity laws: (fill in the blanks)
Capacity varies as the ______________ of the speed or impeller diameter
Head varies as the ____________ of the speed or impeller diameter
Horsepower varies as the _____________ of the speed or impeller diameter

12. Name three primary references that can be used when specifying a centrifugal pump.
(a) ____
(b) ____
(c) ____

13. Using Fig. 3.205, identify the components of the centrifugal pump installation.
(a) ___________________________
(b) ___________________________
(c) ___________________________
(d) ___________________________
(e) ___________________________
(f) ___________________________
(g) ___________________________
(h) ___________________________
(i) ___________________________
(j) ___________________________

14. Using Fig. 3.206, determine if the centrifugal pumping system is operating properly by first calculating the total dynamic head.
(a) 25.3 ft
(b) 44.6 ft
(c) 59.4 ft
(d) 99.6 ft
(e) 183.1 ft

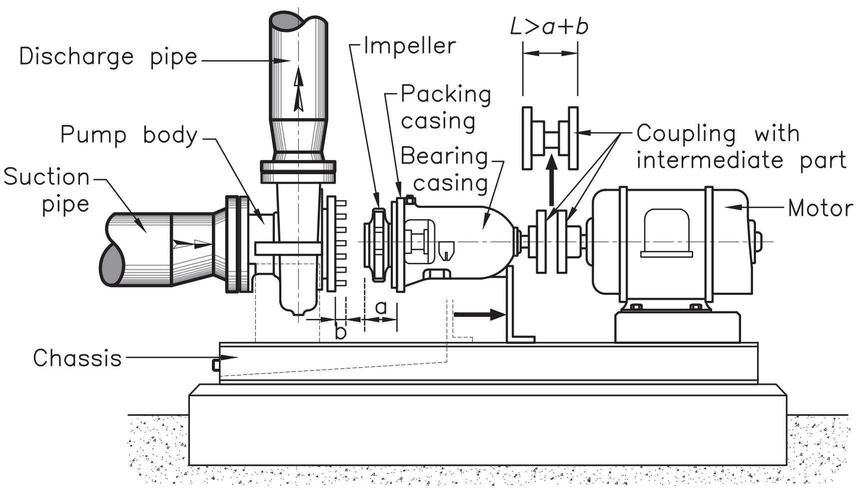


Fig. 3.205 Exercise 13 Centrifugal pump installation.

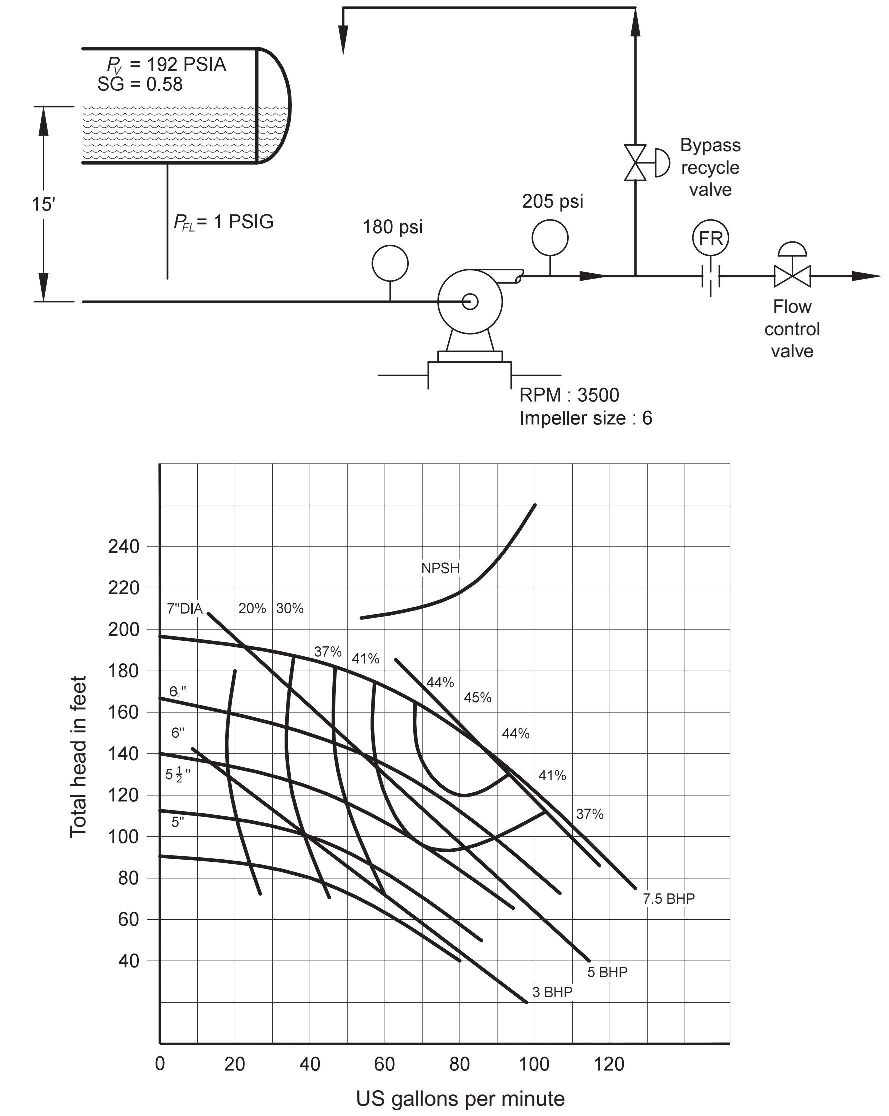


Fig. 3.206 Exercises 14 and 15 Pump performance curves.

15. If the pump in Question 14 is determined not to be performing, and assuming that all instruments are correctly calibrated, pressures correctly read, an accurate specific gravity and no seal leakage, list the most probable causes for poor pump performance.

(a) ____

(b) ____

(c) ____

(d) ____

16. Develop a system head curve based on the following information. The system head curve is a graphical representation of how much total dynamic head (TDH) is required to pump various capacities through the piping system. Use the friction loss tables included in the Hydraulic Institute Engineering Data book.

Suction tank pressure (P_1): 15 psia
Fluid specific gravity (SG): 0.8
Suction piping: Negligible
Minimum static head above
Centerline of pump (H_1): 10 ft
Discharge piping:
Pipe: 5232′ of 10″ Std. WT pipe
Fittings: 2–10″ long radius ells
Valves: 2–10″ ball valves
1–10″ swing check
Flow rate (Q) 2000 gpm
Discharge vessel pressure (P_2) 100 psig
Maximum static head in discharge
Line above centerline of pump (H_2) 100 ft

17. A centrifugal pump is used for propane service (SG = 0.508). The pressure in the vessel is at 190 psia. The liquid is at its bubble point. Elevation of liquid level in the tank is 689 ft; elevation of the pump suction nozzle is 673 ft. The friction loss in the suction line and fittings is 1.5 psi at the desired pumping rate. Atmospheric pressure is 14.6 psia. Neglect velocity head. Determine the NPSH available.

18. A cooling water pump circulates 200 gpm from a cooling tower basin.

(a) Determine the power requirements if the following data apply:

Water density	62.4 Ib/ft^3
Suction pressure	14.5 psia
Discharge pressure	75 psig
Overall efficiency	0.75

(b) Assuming a centrifugal pump, calculate the specific speed at 3600 rpm.

19. Your assignment is to install a spare centrifugal shipping pump during an upcoming upgrade scheduled to begin in 3 months. A multistage centrifugal pump was selected. The appropriate information from the pump data sheet, pump performance curves, and field are attached. A third-party consultant was hired to witness a NPSH required performance test performed by the manufacturer. A copy of the composite NPSH required performance test results is also attached.

Temperature	170°F
Suction pressure	360 psig
Discharge pressure	760 psig
Fluid	Crude oil (30% WTR cut)
Oil specific gravity	0.77
Water specific gravity	1.05
Static head	19 ft
Line losses	1 psi
Atmospheric pressure	15 psia
Flow rate	8000 BPD

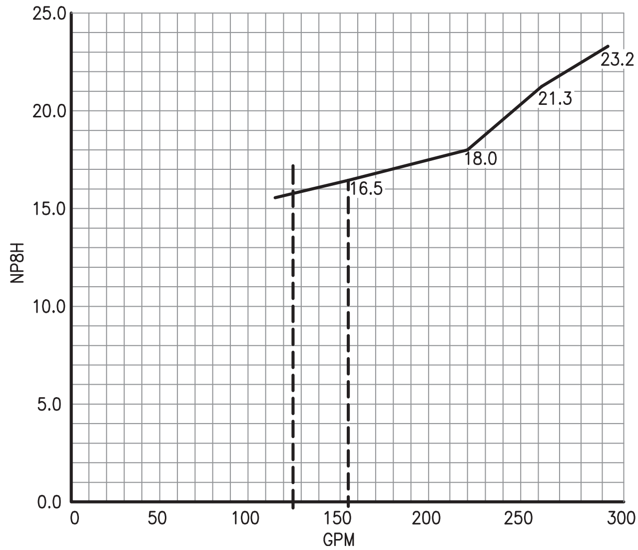


Fig. 3.207 Exercise 19 NPSH required performance test.

Determine the following (Figs. 3.207 and 3.208):

(a) NPSH required
(b) NPSH available
(c) Margin
(d) Will the pump have sufficient NPSH available at the desired flow rate of 8,000 BPD?
(e) If the pump is determined not to have sufficient NPSH available, then what flow rate can be achieved at the available NPSH?
(f) What is the pump efficiency at this flow rate?

20. True or false. The pump curve is the most important tool in performance evaluation.
21. What on the pump curve must be confirmed to assure correct performance evaluation?
22. Which are more common: series pumps or parallel pumps?
23. What is the common name for the first pump in a series arrangement?
24. Two identical centrifugal pumps operating together in parallel in a closed system are most likely to provide how many times the flow of a single pump: 1.3, 2, or 2.1?
25. What is cavitation and how, if necessary, can it be avoided? Name at least three ways.
26. A performance evaluation showing low head and low power draw is likely the result of what?
27. A single-stage double-suction pump was just repaired using all the same major parts with no modifications, yet is putting up much less flow and head than before the repair. What repair error was most likely made?
28. Which condition is least likely to impact mechanical seal life; cavitation, vibration, bad seal flush cooler, or bad wear rings?
29. True of false. Percent head deficiency is the best criteria for determining whether or not a centrifugal pump requires repairs for performance.
30. True of false. Oil mist lubrication is essential to the following pump bearings. Purge (wet sump) and Purge (dry sump).

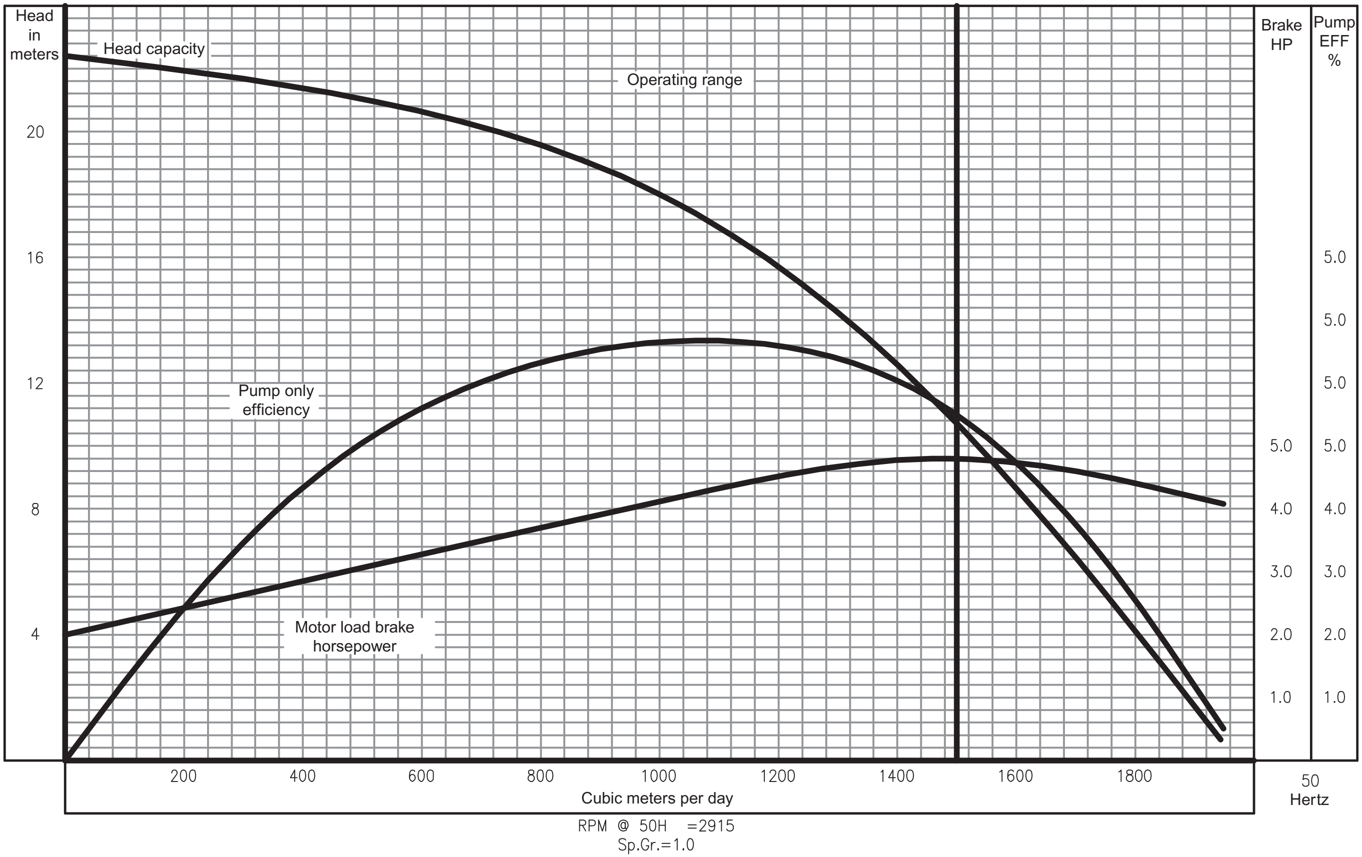


Fig. 3.208 Exercise 19 Composite NPSH required performance test results.

31. Which seal is best suited for high temperature services? Spring pusher, tandem, bellows, or high temperature elastomeric.

32. Adding an extra impeller to a multistage pump will do what to the performance curve? Raise the flow, raise the head, raise both according to the affinity laws, or raise both proportionally.

33. Raising the fluid temperature at a fixed DCS flow rate will impact the following parameters which way (up, down, or not at all)? Volume flow, differential pressure, specific gravity, power consumed, and NSPH.

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Reciprocating pumps

4

Nomenclature

A	plunger or piston cross-sectional area, mm^2 (in^2)
a	piston rod cross-sectional area, mm^2 (in^2)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
D	pump displacement, m^3/h (gpm)
d	displacement per pumping chamber, m^3 (gal)
d'	pump piston or plunger diameter, mm (in.)
d_w	diameter of wrist pin, mm (in.)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/s
g	acceleration due to gravity, 9.81 m/s^2 (32.2 ft/s^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
H_f	pipe friction loss, m (ft)
H_p	total head required for pump, m (ft)
HHP	hydraulic horsepower, kW (hp)
K	A factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in.)
m	number of pistons, plungers, or diaphragms
NPSH	net positive suction head, m (ft)
$NPSH_A$	net positive suction head available, m (ft)
$NPSH_R$	net positive suction head required, m (ft)
n	stroke rate or crank revolutions per min, rps (rpm)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
ΔP	static pressure, kPa (psi)
Q	flow rate, m^3/h (ft^3/s)
Q'	flow rate, m^3/h (BPD)
q	flow rate, m^3/h (gpm)
S	stress, kPa (psi)
S'	valve slip, %
s	stroke length, mm (in.)
(SG)	specific gravity of liquid relative to water
V	velocity in suction line, m/s (ft/s)
Vol	volume of surge tank, m^3 (ft^3)

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00004-1

$(Vol)_g$ required gas volume, m^3 (ft^3)
Z elevation above or below pump centerline datum, m (ft)
ρ density of fluid, kg/m^3 (lb/ft^3)

4.1 Engineering principles

4.1.1 Background

Positive displacement (PD) pumps were developed long before centrifugal pumps. Fluid is positively displaced from a fixed volume container. PD pumps are

- Capable of developing high pressures while operating at low suction pressures
- Self-priming
- Commonly referred to as constant volume pumps

Unlike centrifugal pumps, a PD pump's capacity is not affected by the pressure against which it operates. PD pumps' flow rate is usually regulated by varying the speed of the pump. PD pumps are generally used in

- High head
- Low flow applications
- Where high maintenance costs are more than offset by their higher efficiency

4.1.2 Pumping action

In a PD pump the volume containing the liquid is decreased until the resulting liquid pressure is equal to the pressure in the discharge system. That is, the liquid is compressed mechanically causing a direct rise in potential energy. As shown in Fig. 4.1, PD pumps are classified as either reciprocating or rotary pumps. Most PD pumps are classified as reciprocating pumps where the displacement is accomplished by the linear motion of a piston or plunger is a cylinder. Rotary pumps' displacement is accomplished by a circular motion. Rotary pumps are discussed in Chapter 5.

PD pumps, unlike centrifugal pumps, will in theory produce the same flow at a given speed no matter what the discharge pressure. Thus PD pumps are "constant flow machines." However due to a slight increase in internal leakage as the pressure increases, a truly constant flow rate cannot be achieved. A PD pump must not be operated against a closed valve on the discharge side of the pump, because it has no shutoff head like centrifugal pumps. A PD pump operating against a closed discharge valve will continue to produce flow and the pressure in the discharge line will increase, until the line bursts or the pump is severely damaged, or both. Since the pump will attempt to match whatever system pressure is required, it is necessary that a relief valve be installed on the pump discharge. This procedure ensures that the pump will not overpressure itself or the discharge pipe.

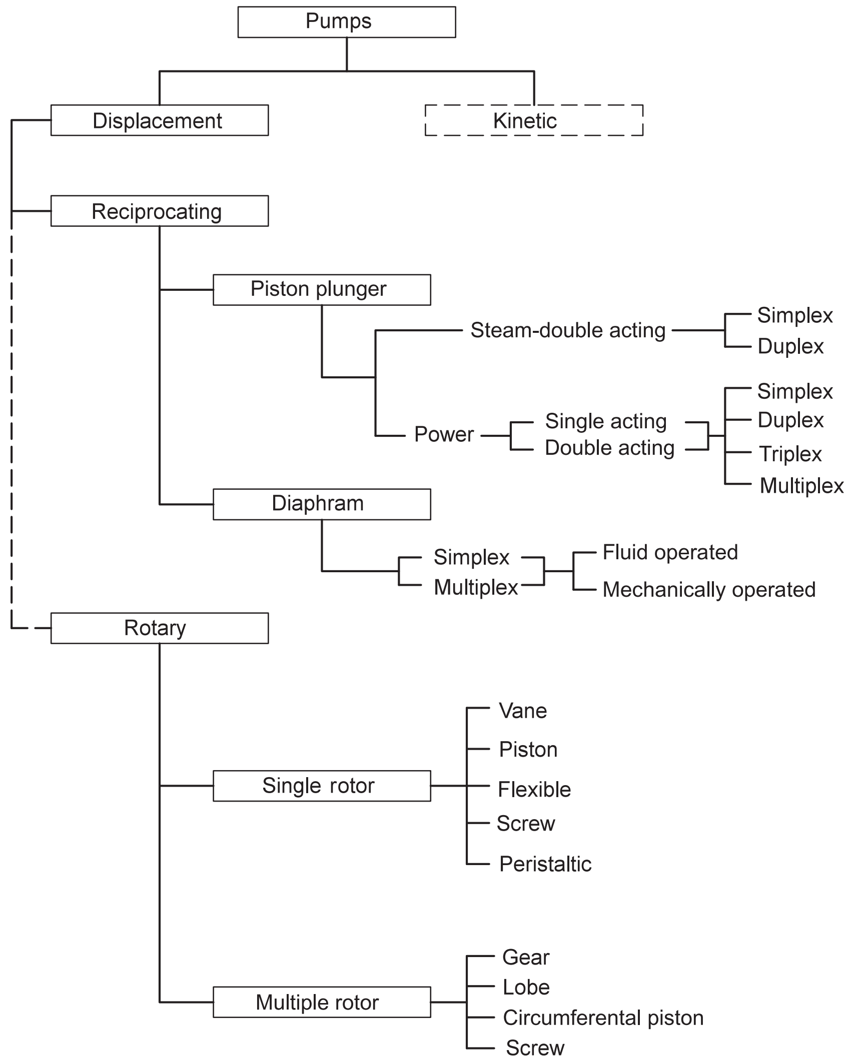


Fig. 4.1 Classification of positive displacement pumps.
Courtesy of the Hydraulic Institute.

A reciprocating pump traps a fixed volume of liquid at near-suction conditions, compresses it to discharge pressure, and pushes it out of the discharge nozzle. Energy is added to the fluid intermittently as one or more boundaries is moved linearly with a piston, plunger, or diaphragm in one or more fluid-containing volumes. Liquid is moved by means of a constant back-and-forth motion of a piston, plunger, or diaphragm within a fixed volume or cylinder.

4.1.3 Pump types

4.1.3.1 Plunger and piston pump (Fig. 4.2)

In a plunger pump, the plunger moves through a stationary packed seal and is pushed into the fluid. In a piston pump, the packed seal on the piston pushes the fluid from the cylinder. Movement of either the plunger/piston creates an alternating increase and decrease of flow. As shown in Fig. 4.3, as the piston, or plunger, moves backward (suction stroke), the available volume in the cylinder increases and a suction valve opens to allow the fluid to enter the cylinder. As shown in Fig. 4.4, as the piston, or plunger, moves forward (discharge stroke), the volume available in the cylinder

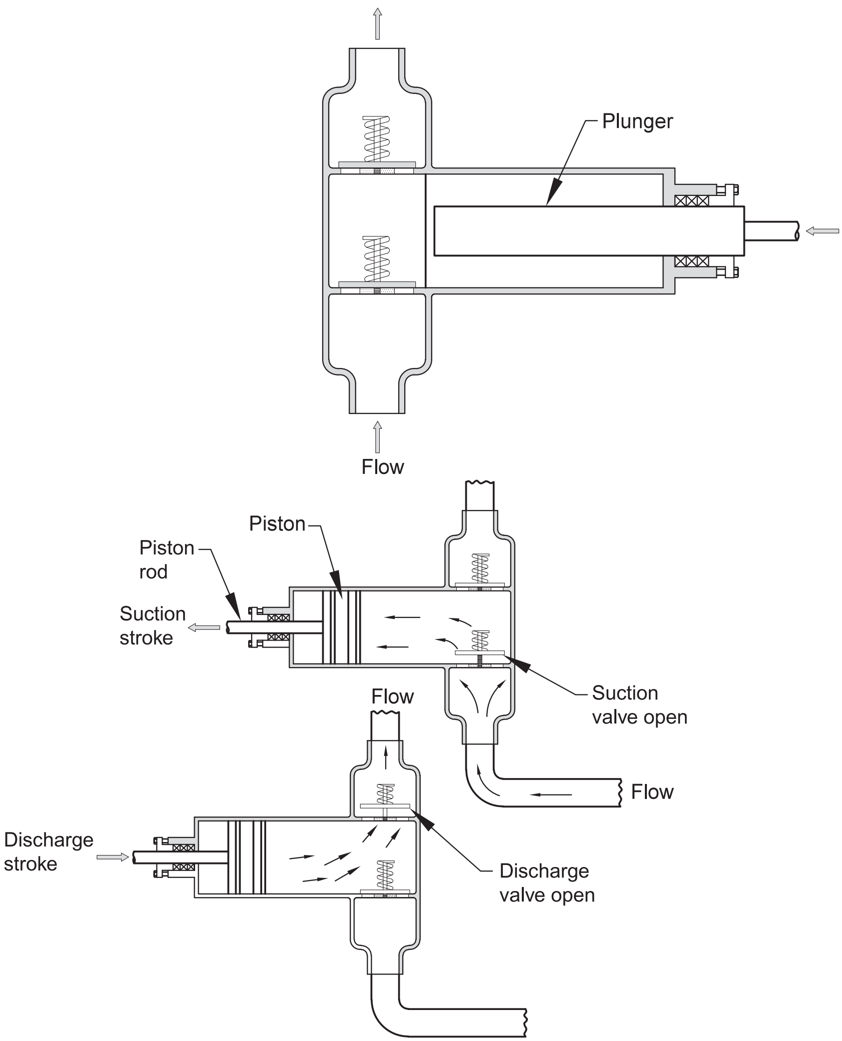


Fig. 4.2 Schematic diagrams illustrating a plunger (top) and piston (bottom) pump.

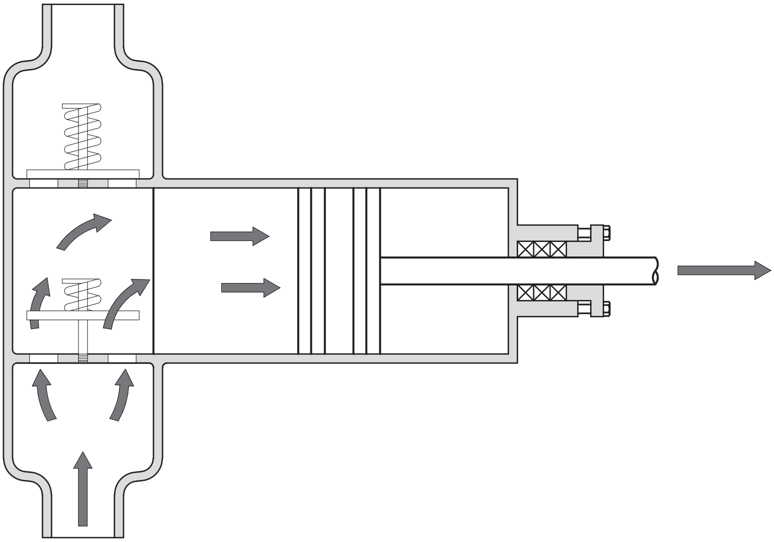


Fig. 4.3 Schematic illustrating the backward "suction" stroke of a single-acting pump.

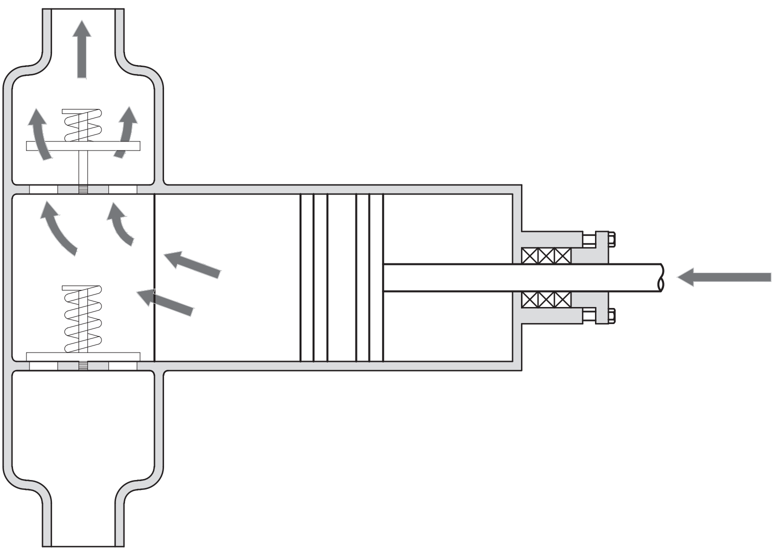


Fig. 4.4 Schematic illustrating the forward "discharge" stroke of a single-acting pump.

decreases, the pressure of the fluid increases, and the fluid is forced out through a one-way discharge valve.

If liquid is pumped during linear movement in one direction only (Fig. 4.5), the pump is classified as "single-acting." If the liquid is pumped during movement in both directions (Fig. 4.6), it is classified as "double-acting."

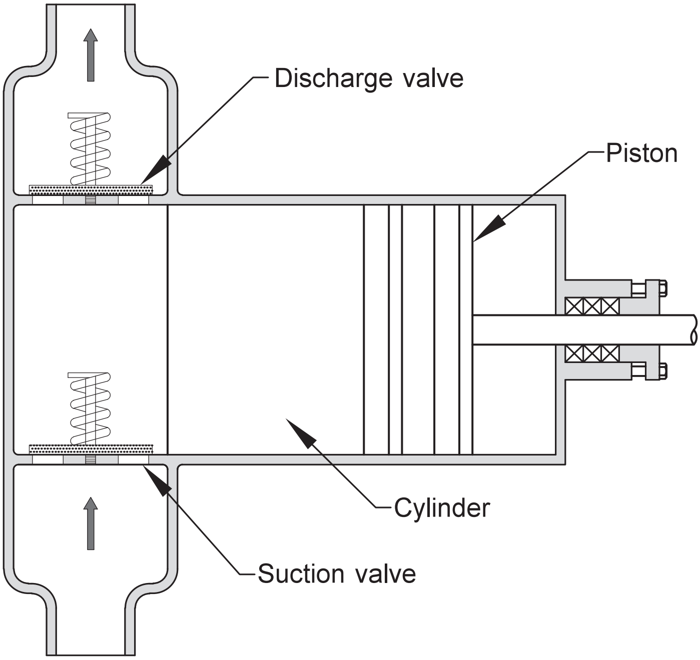


Fig. 4.5 Schematic diagram of a "single-acting" piston pump.

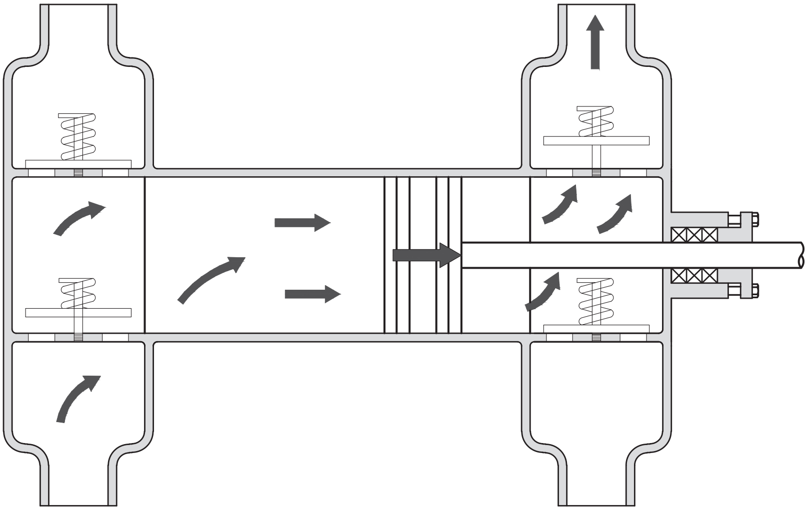


Fig. 4.6 Schematic diagram of a "double-acting" piston pump.

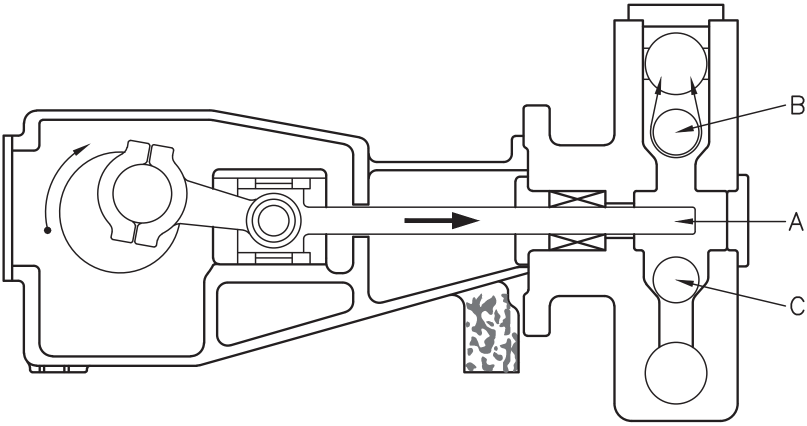


Fig. 4.7 Schematic diagram of a "single-acting" plunger pump (A—plunger; B—discharge check valve; C—suction check valve).

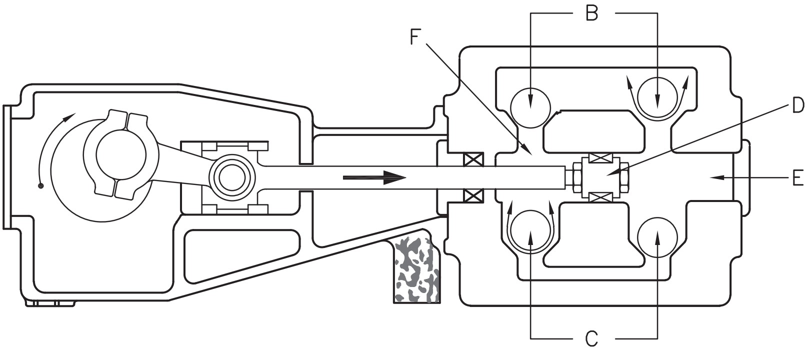


Fig. 4.8 Schematic diagram of a "double-acting" piston pump (B—discharge check valve; C—suction check valve; D—piston; E—discharge pressure; and F—suction pressure).

Figs. 4.7 and 4.8 show a single-acting and double-acting pump. As the plunger (*point A*) moves to the right in the single-acting pump, the fluid is compressed until its pressure exceeds the discharge pressure, and the discharge check valve (*point B*) opens. The continued movement of the plunger to the right pushes liquid into the discharge pipe. As the plunger begins to move to the left, the pressure in the cylinder becomes less than that in the discharge pipe, and the discharge valve (*point B*) closes. Further movement to the left causes the pressure in the cylinder to continue to decline until it is below suction pressure. At this point the suction check valve (*point C*) opens. As the plunger continues to move to the left, the cylinder fills with liquid from the suction. As soon as the plunger begins to move to the right, it compresses the liquid to a high enough pressure to close the suction valve (*point C*), and the cycle is repeated. Thus liquid is discharged only when the plunger moves to the right.

In a double-acting pump, the plunger is replaced by a piston (*point D*). When the piston moves to the right, the liquid in the cylinder to the right of the piston (*point E*) is discharged, and the cylinder to the left of the piston (*point F*) is filled. When the direction of the piston is reversed, the liquid in "*F*" is discharged, and the cylinder at "*E*" is filled with suction fluid. Thus liquid is pumped when the piston moves in either direction.

Reciprocating pumps are also classified by the number of cylinders they have. If the liquid is contained in one cylinder it is called a simplex pump, two cylinders a duplex, three cylinders a triplex, five cylinders a quintuplex, seven cylinders a setuplex, and so forth. Reciprocating pumps normally have odd numbers of cylinders to reduce the rocking coupling forces.

As shown in Fig. 4.9, the head-flow rate curve is a nearly straight vertical line. That is, no matter how high a head is required, the plunger will displace a given volume of liquid for each rotation. The flow rate through the pump can be varied only by changing the pump speed. A throttling valve that changes the system head-flow rate curve will have no effects on the flow rate through the pump.

4.1.3.1.1 Advantages

The advantages of reciprocating pumps are as follows:

- High efficiency regardless of changes in required head; efficiencies on the order of 85% to 95% are common.
- The efficiency remains high regardless of pump speeds, although it tends to decrease slightly with increasing speed.
- Reciprocating pumps' operating speeds are much lower than those of centrifugal pumps. As a result, reciprocating pumps are better suited for handling viscous fluids.
- For a given speed the flow rate is constant regardless of head. The pump is limited only by the power of the prime mover and the strength of the pump parts.
- Capable of producing high pressures and large capacities and are self-priming (Fig. 4.10)

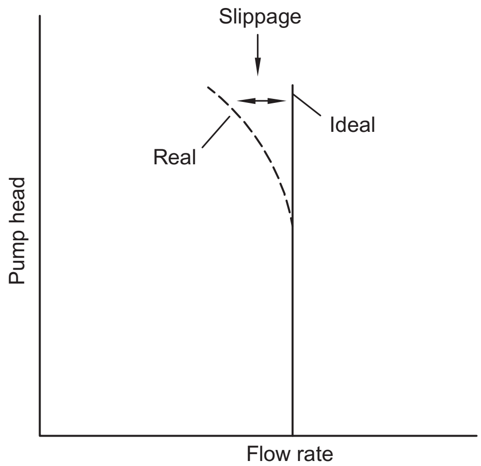


Fig. 4.9 Positive displacement reciprocating pump characteristic curve.

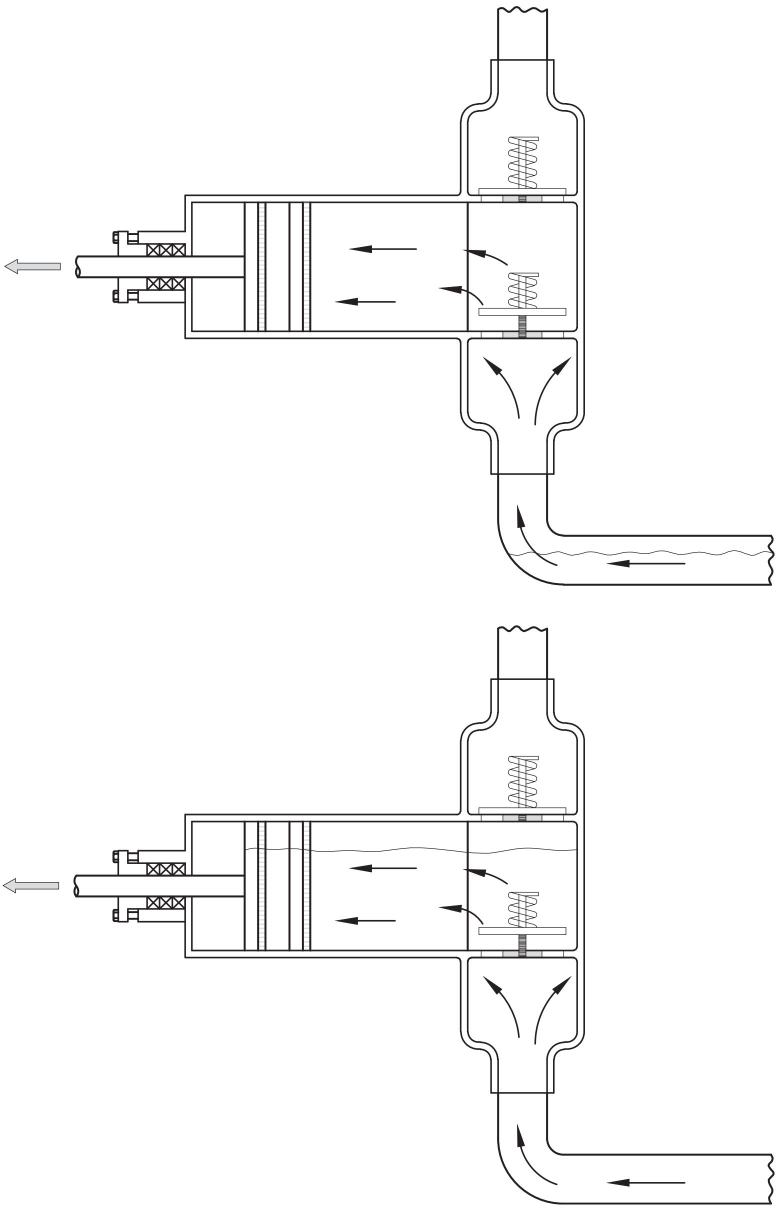


Fig. 4.10 Schematic showing "self-priming" of a reciprocating pump.

4.1.3.1.2 Limitations

Reciprocating pumps, because of the nature of their construction, have the following limitations when compared to centrifugal pumps:

- They have higher maintenance cost and lower availability because of the pulsating flow and large number of moving parts.

- They are poorer at handling liquids containing solids that tend to erode valves and seats.
- Because of the pulsating flow and pressure drop through the valves, they require larger suction pressures ($NPSH_R$) at the suction flange to avoid cavitation.
- They are heavier in weight and require more space.
- Pulsating flow requires special attention to suction and discharge piping design to avoid both acoustical pulsations and mechanical vibrations.

4.1.3.1.3 Vapor-locking

Vapor-locking is the condition where the cylinder becomes filled with high-pressure gas. As the piston moves forward, it compresses the gas into a smaller volume. The pressure of the gas in the cylinder may not rise as high as the discharge line pressure thus preventing the discharge valve from opening. This condition causes excess heat and packing failure (Fig. 4.11).

4.1.3.2 Diaphragm pump

Diaphragm pumps are a special type of reciprocating pump that uses the action of a diaphragm moving back and forth within a fixed chamber. Fig. 4.12 shows a typical diaphragm pump where the flexure of the diaphragm creates the pumping action. Diaphragm pumps are also known as controlled volume, proportioning, metering, and chemical injection.

The principle of operation is similar to that of the plunger/piston pump except that, instead of a cylinder, there is a flexible pulsating diaphragm. When gas pressure is applied against either diaphragm it forces liquid out. When the gas is relieved the diaphragm flexes under the pressure in the suction line and allows liquid to enter.

Diaphragm pumps can be fluid or mechanically driven at either constant or variable speeds. They are limited to small flow rates and moderate temperatures. They usually handle a small discharge volume (typically between 1 and 10 gpm) and high discharge

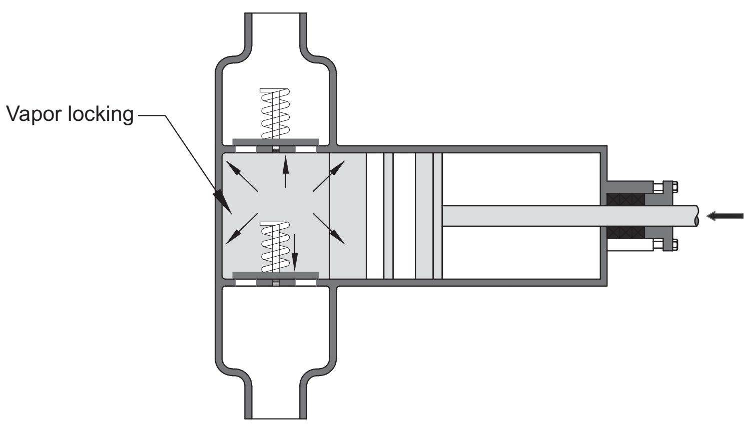


Fig. 4.11 Schematic illustrating the concept of "vapor-locking."

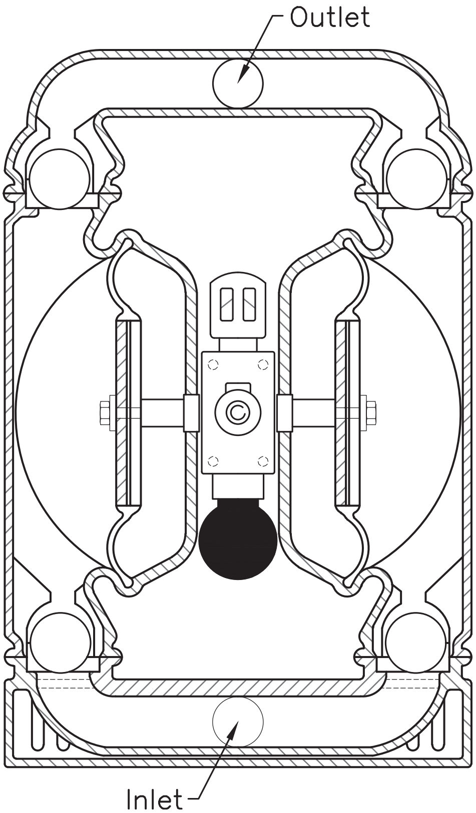


Fig. 4.12 Schematic diagram of a diaphragm pump.

pressure (up to 30,000 psig) (206,843 kPa). The volume must be infinitely controlled between limits and virtually independent of discharge pressure.

Diaphragm pumps are driven by fixed-speed electric motor through an integral reduction gear. Capacity control is achieved by adjusting the pump stroke. Air-driven are also commonly used in upstream operations. Proportioning mechanisms are usually integrated with the drive mechanism.

Diaphragm pumps provide an effective solution to leakage problems and, to a lesser extent, abrasive problems. There are two kinds of diaphragm pumps.

- Mechanical diaphragm drive (Fig. 4.13).
- Hydraulic diaphragm drive (Fig. 4.14).

Although mechanical drives are both simple and cheap, they have a short diaphragm life and are only suitable for very light duty (automobile fuel pumps) applications and will not be discussed further in this section.

Hydraulic drives may have a single or double diaphragm. A single diaphragm is the most common and suitable for most services (Fig. 4.14). However, a double diaphragm may be necessary for extremely toxic services.

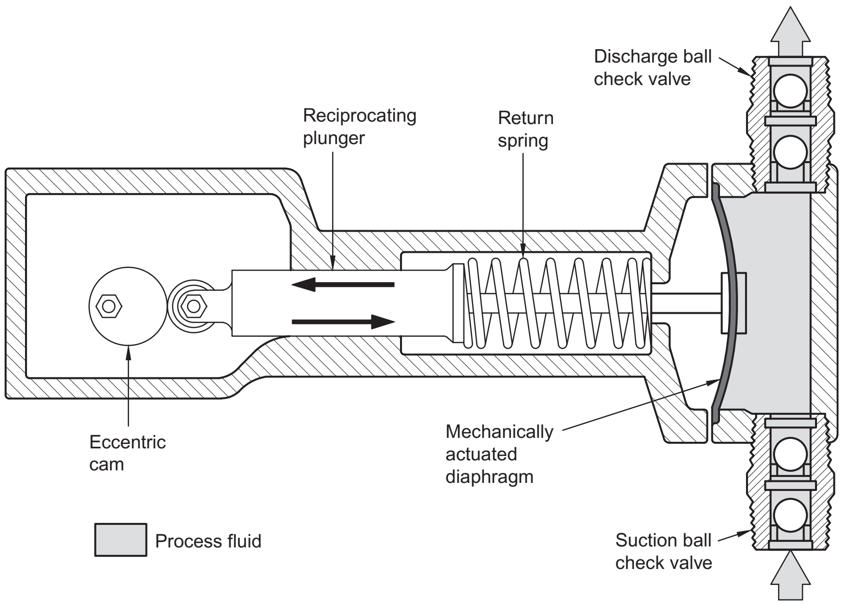


Fig. 4.13 Schematic diagram of a mechanical diaphragm driver.
Courtesy of Marcel Dekker, Inc.

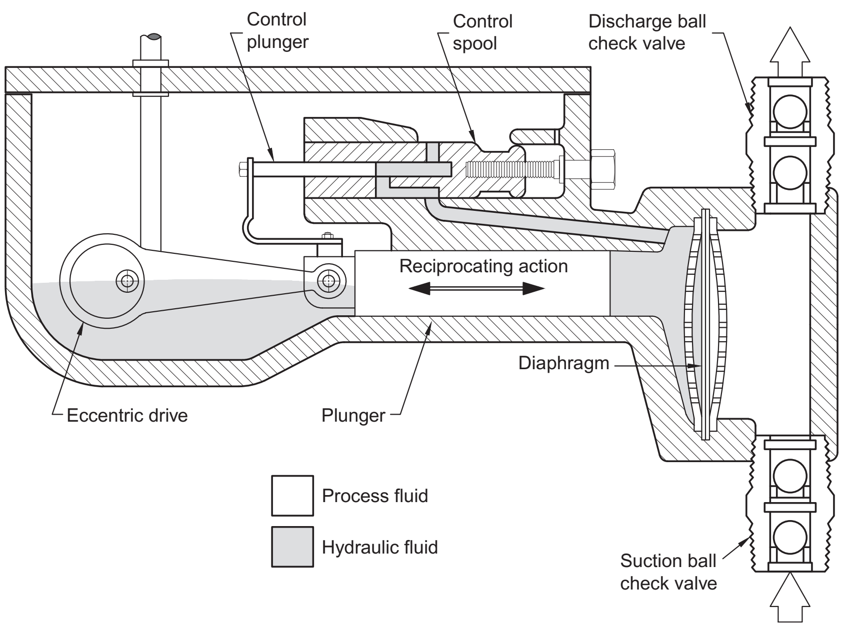


Fig. 4.14 Schematic diagram of a hydraulic diaphragm driver.
Courtesy of Marcel Dekker, Inc.

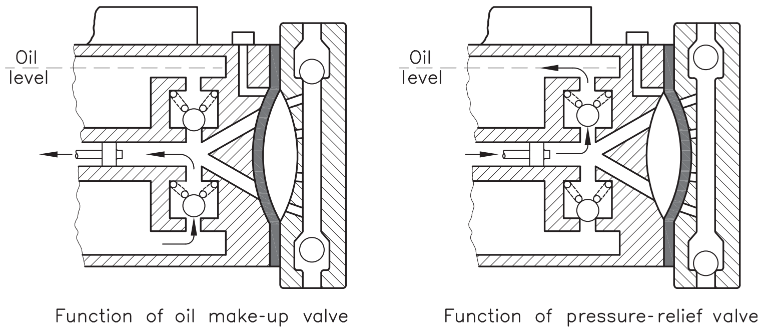


Fig. 4.15 Schematic diagram illustrating relief valve principles used in a diaphragm pump.

Diaphragms are usually made of Teflon or stainless steel, but elastomers or elastomer-coated steel diaphragms are also available. The double diaphragm provides positive isolation between the process fluid and the drive fluid (hydraulic oil). The diaphragm interspace may be designed with alarms to alert personnel to contamination by process fluid, for example, conductivity, which may indicate outer diaphragm failure.

To prevent diaphragm overstress, relief valves are incorporated into the drive system. Fig. 4.15 illustrates relief valve principles.

The hydraulic drive system looks similar to a packed plunger pump. However, it has a number of advantages over the plunger pump:

- The plunger works in an ideal fluid, that is, good lubricity, clean, and so on.
- The hydraulic drive uses relief valves to avoid diaphragm overstress. This feature is the equivalent of a discharge pressure shut off.
- It pumps corrosive and abrasive materials with much lower wear rates and better reliability than packed plunger pumps.
- Field repairs can be made quickly.

4.1.3.2.1 Advantages

Diaphragm pumps have the following advantages:

- Capable of pumping fluids that are viscous, erosive, corrosive, or contain large amounts of suspended solids.
- Long life and are inexpensive to repair since they have no stuffing box and have few moving parts.
- Can handle low flow rates inexpensively.
- Self-priming and can run periodically without any liquid.

4.1.3.2.2 Limitations

Diaphragm pumps have the following limitations:

- Limited to small flow rates (90 gpm) (21 m^2/h), moderate discharge pressures [1000 ft (305 m) of head], and moderate temperatures.
- Require frequent maintenance because they are reciprocating pumps.

- Diaphragm has a tendency to *fatigue* with time (diaphragm life is usually more than several hundred thousand but <2 to 3 million cycles).
- Diaphragm leakage can cause a hazard by mixing power gas with the process fluid.

Sometimes air or natural gas is used to power a diaphragm, which in turn drives a reciprocating pump. In this situation it is possible to handle large discharge pressures, but only if the flow rate is very small. Typically, this type of pump is used as sump pumps or used for chemical injection in field gathering and treating operations.

4.1.3.2.3 Application guidelines

Pumps may be arranged with hydraulic ends in parallel and/or series. Parallel operation helps achieve desired volumes. One major advantage of parallel operation is that the pulsating flow characteristic of a single unit can effectively "smooth" by carefully phasing the drive for each liquid end.

Series pumping is rare but required for high pressure [>1000 psi (68 bar)].

Fig. 4.16 summarizes diaphragm pump application guidelines.

4.2 Performance considerations

4.2.1 Pressure

Reciprocating pumps are constant volume pumps. Variations in discharge pressure do not affect flow rate. Since these pumps continue to deliver the same capacity, any attempt to throttle the discharge flow may overpressure the pump casing and/or discharge piping. Thus no reciprocating pump should ever be started or operated with the discharge block valve closed. Flow is regulated by speed. In rare cases that require a discharge throttle valve, an automatic bypass valve that is piped back to the suction source must be provided.

Fig. 4.17 is a schematic diagram illustrating how a force of 10,000 pounds (44,482,216 N) exerted on a piston with an area of 50 in^2 (323 cm^2) yields a discharge pressure of 200 psi (1379 kPa)

4.2.2 Capacity

The capacity (q) of a reciprocating pump is the total volume of fluid actually delivered per unit of time. The volume includes liquid and any dissolved or entrained gas at the stated operating conditions. Pump capacity is calculated from the following

$$q = D(1 - S) \tag{4.1}$$

Field units

$$q = dnm(1 - S) \tag{4.2a}$$

SI units

$$q = 3600\,dnm(1 - S) \tag{4.2b}$$

Metering pump application criteria		
Liquid end type	**Recommended for**	**Not recommended for**
Packed plunger	• Very high discharge pressure	• Corrosive fluids
	• Temperature over 250°F	• Abrasive fluids
	• Low vapor pressure fluids	• Applications whose trace contamination of pumpage with packing lubricants is not permitted
Single diaphragm (hydraulic drive)	• Corrosive fluids • Applications requiring high reliability	• High discharge pressure (>1500 psi) • Temperatures over 175°F (elastomer diaphragm) • Over 250°F (teflon diaphragm)
Double diaphragm	• Very corrosive or hazardous fluids	• High discharge pressure (>1000 psi)
	• Abrasive fluids	
	• Duties requiring guaranteed isolation from drive oil	
	• Applications requiring early warning of diaphragm failure	

(A)

Comparison of metering pumps		
Pump type	**Advantage(s)**	**Disadvantage(s)**
Mechanical diaphragm	• Least expensive • Can handle most fluids • Glandless	• Accuracy not as good as plunger and piston-diaphragm pumps • Diaphragm subject to fatigue failure • Limited to low pressure deliveries
Piston diaphragm	• High accuracy • Good diaphragm life • Can handle most fluids • Glandless • Readily rendered in double diaphragm form for fail-safe characteristics	• Diaphragm may be subject to wear if pumped fluids include particulate matter
Electromagnetic-driven diaphragm	• Particularly suitable for micro-metering with precise pulse operation • Glandless	• Limited capacity • More complex control system (needs digital signal input)

(B)

Fig. 4.16 (A) Diaphragm pump application guidelines and (B) comparison of diaphragm pumps.

where

q = pump capacity, gpm (m^3/h), delivered by all pistons, plungers, or diaphragms
D = pump displacement, gpm (m^3/h)
S = slip, fraction
d = displacement per pumping chamber, gal (m^3)
m = number of pistons, plungers, or diaphragms
n = pump speed, rpm (rps)

The displacement (D) of a reciprocating pump cylinder or chamber is the calculated volume displaced per chamber per single stroke of the piston, plunger, or diaphragm

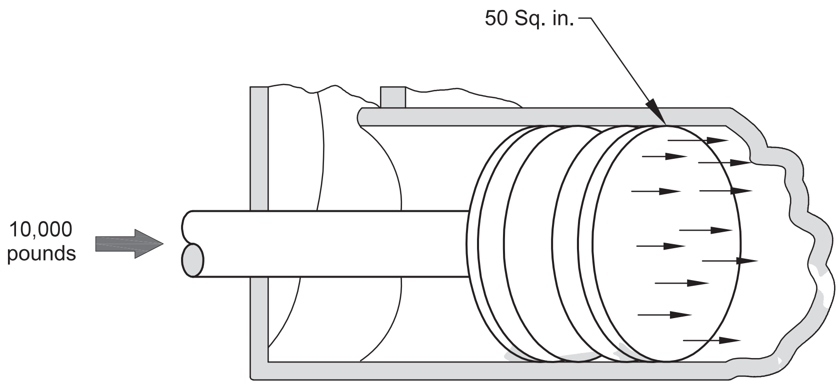


Fig. 4.17 Schematic diagram illustrating a force of 10,000 pounds (44,482,216 N) acting on a piston with area of 50 in^2 (323 cm^2) yields a discharge pressure of 200 psi (1379 kPa).

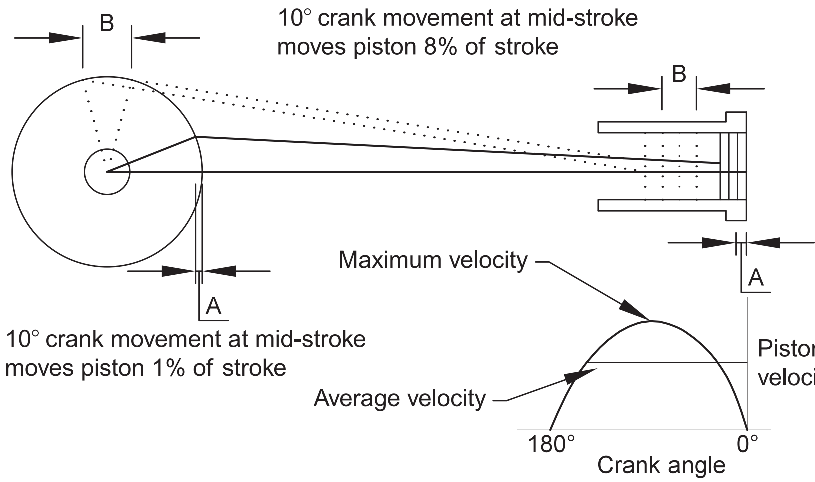


Fig. 4.18 Schematic diagram illustrating "cylinder displacement."

(Fig. 4.18), with no losses due to slip or fluid compressibility. Displacement of an entire pump is the total displacement of all the chambers when the pump is operating at a specified speed. Deductions for piston rod volume should be made on double-acting piston pumps when displacement is calculated.

Cylinder displacement for a *single-acting* cylinder is given by

Field units

$$D = \frac{ASNm}{231} \tag{4.3a}$$

SI units

$$D = \frac{ASNm}{277,808} \tag{4.3b}$$

Cylinder displacement for a *double-acting* cylinder is given by
Field units

$$D = \frac{(2A - a)SNm}{231} \tag{4.4a}$$

SI units

$$D = \frac{(2A - a)SNm}{277,808} \tag{4.4b}$$

where
D = cylinder displacement, gpm (m^3/h)
A = plunger or piston area, in^2 (mm^2)
a = piston rod cross-sectional area, in^2 (mm^2)
S = stroke length, in (mm)
N = speed, rpm (rps)
m = number of pistons or plungers

The stroke of a reciprocating pump is the number of complete pumping cycles of the plunger, piston, or diaphragm per minute (stokes/min).

Slip is the loss of capacity as a percentage of the cylinder displacement due to

- volumetric efficiency
- stuffing box losses
- valve losses

Volumetric efficiency (not to be confused with mechanical efficiency) for pistons is normally 95% to 97% and lower for plungers. Efficiency is also reduced when pumping a light hydrocarbon that has some degree of compressibility.

Pump capacity can be determined from the following:

$$Q = \frac{D(100 - Slip)\%}{100} \tag{4.5}$$

Table 4.1 presents typical values of slip for various viscosities.

4.2.3 Speed

Speed is the primary factor in the determination of the capacity of a reciprocating pump. The primary constraint in determining a pump's speed is the average velocity (V_{AVG}) of the reciprocating elements, which can be determined by

Table 4.1 Pump efficiency reductions due to high-viscosity fluids

Centistokes	Viscosity (SSU)	Slip (%)
215	1000	90
430	2000	80
865	4000	70
1300	6000	62
1725	8000	55
2160	10,000	50

$$V_{AVG} = N\left(\frac{S}{6}\right) \tag{4.6}$$

where

V_{AVG} = average velocity, ft/min (m/s)

Running at high speeds shortens packing life. Table 4.2, based on field tests and observations, lists recommended maximum continuous service speeds and velocities.

It is permissible to interpolate between the values given in Table 4.2. Higher velocity values may be used, but only for intermittent and cyclic duty. Lower speeds should be used when the NPSH margin is <2 ft.

4.2.4 Viscosity

In general, reciprocating pumps are capable of handling liquids of up to about 100 centistokes (475 SSU) without any reduction in speed and capacity, although the efficiency will be slightly lower than with water. At higher viscosities, efficiency is likely to fall still further because of increased resistance to flow through ports and valves. It is therefore usual to operate reciprocating pumps at reduced speed when handling high-viscosity fluids.

Table 4.2 Recommended maximum continuous service speeds and velocities

Stroke (S), in. (mm)	Speed (N), rpm (rps)	(V_{AVG}), fpm (m/s)
1 (25)	500 (30,000)	83 (25)
2 (51)	450 (27,000)	150 (45)
3 (76)	400 (24,000)	200 (61)
4 (102)	350 (21,000)	233 (71)
5 (127)	310 (18,600)	258 (79)
6 (152)	270 (16,200)	270 (82)
7 (178)	240 (14,400)	280 (85)
8 (203)	210 (12,600)	280 (85)

Table 4.3 Pump efficiency reductions due to high-viscosity fluids

Viscosity, SSU (centistokes)	% of Maximum speed
<300 (<65)	100
300–1000 (65–220)	90
1000–2000 (220–440)	80
2000–4000 (440–880)	70
4000–6000(880–1320)	60
6000–8000(1320–1760)	55
8000–10,000(1760–2200)	50

Courtesy of Hydraulic Institute.

High viscosity also reduces the slip parameter, S, reducing capacity as presented in Table 4.3. These values are illustrative only, as much depends on the size and detailed design of the pump.

Operating below maximum "rated" speed may be advantageous when:

- Pump is operated unattended
- Pump has no spares and no standby
- There is a high penalty for down time
- Unit is remotely located
- Unit maintenance is poor
- Long life is desired
- NPSH margin is low

Operating at the maximum "rated" speeds requires

- Clean, cool fluids
- Excellent piping layout with rigidly fixed piping
- Good operating environment and solid foundation
- Adequate NPSH margin
- Well-designed suction stabilizers and discharge dampeners
- Good maintenance

Whenever it becomes necessary to operate above the maximum "rated" speed, very close attention should be given to all design, operation, and maintenance details.

4.2.5 Net positive suction head ($NPSH_R$)

For a reciprocating pump, energy is required to push the suction valve from its seat and to overcome the friction losses and acceleration head. Since these losses can be significant (especially at low speeds), $NPSH_R$ is expressed in terms of pressure (psi) rather than head (feet).

$NPSH_R$ is a function of many factors including

- Liquid density
- Vapor pressure
- Viscosity

- Dissolved air/gas
- Pump speed
- Piston diameter
- Valve type and flow area
- Valve spring load and spring rate
- Liquid passage configuration
- Stuffing box leakage

The $NPSH_A$ is the total suction head at the pump suction flange less the absolute vapor pressure of the liquid being pumped. The NPSH margin is defined as the difference between $NPSH_A$ less $NPSH_R$. $NPSH_A$ must exceed $NPSH_R$. $NPSH_R$ is affected by changes in

- Plunger size
- Pump speed
- Flow rate
- Suction pressure

A minimum of 7-ft (2.1 m) of NPSH margin is recommended. Additional NPSH margin may be required when handling:

- high vapor pressure fluids
- fluids with dissolved air or gases, or
- fluids with suspended solids

If a reciprocating pumping system exhibits inadequate NPSH margin and cannot be redesigned, it may be necessary to install a suction stabilizer or stand pipe near the pump inlet, reduce the operating speed or increase the source tank liquid level or operating pressure.

Cavitation occurs in a pump when the pressure of the liquid is reduced to a value equal to or below its vapor pressure and small vapor bubbles or pockets begin to form. As these vapor bubbles are compressed to a higher pressure they rapidly collapse. The forces during the collapse are generally high enough to cause fatigue failure of pump valves and under severe conditions can cause serious pitting damage to the pump internals.

The accompanying valve damage and pressure pulsations in the piping system are the most easily recognized signs of cavitation. Cavitation also results in reduced capacity due to the vapor present in the pump. Vibration and mechanical damage to the pump and piping system can also occur as a result of operating in cavitation.

The only way to prevent the undesirable effects of cavitation is to ensure that the $NPSH_A$ in the system is greater than the $NPSH_R$ by the pump. $NPSH_A$ for a reciprocating pump application is calculated in the same manner as for a centrifugal pump, with the exception that, in determining the $NPSH_R$ for a reciprocating pump, some additional allowance must be made for the reciprocating actin of the pump. This additional requirement is termed "acceleration head," which is defined as the head required to accelerate the liquid column on each suction stroke so that there will be no separation due to liquid inertia in the pump or suction line.

If this minimum condition is not met, the pump will experience a fluid knock caused when the liquid separates and then overtakes the receding plunger. This knock occurs approximately two-thirds of the way through the suction stroke. If sufficient head is provided for the liquid to completely follow the motion of the receding face of the plunger, this knock will disappear. NPSH must always have a positive value and can be calculated by the following equations:

For suction lift (liquid supply level below pump centerline)

$$NPSH_A = H_{PA} - H_{VPA} - H_{VH} - H_{F1} - H_1 - H_A \tag{4.7}$$

For "flooded" suction head (liquid supply level above pump centerline)

$$NPSH_A = H_{PA} - H_{VPA} - H_{VH} - H_{F1} + H_1 - H_A \tag{4.8}$$

where

$NPSH_A$ = net positive suction head available, ft (m)
H_{PA} = absolute head on the surface liquid. This will be barometric pressure if suction is from an open or the pressure existing in a closed tank or vessel, ft (m)
H_{VPA} = absolute vapor pressure head at the temperature being pumped, ft (m)
H_{VH} = velocity head in the pump suction piping minus the velocity head in the suction supply, ft (m)
H_{F1} = friction head in the suction line, including entrance losses from the fluid storage vessel to the piping and friction losses through pipe, valves, and fittings, and so on, ft (m)
H_1 = static head that the liquid supply level is above or below the pump centerline, ft (m)
H_A = acceleration head, the head required to accelerate the liquid column on each stroke so that there will be no separation of this liquid in the pump or suction line, ft (m). Always subtracted from the $NPSH_A$

When pumping from a production vessel the liquid is normally in equilibrium with the gas in the vapor space and thus:

$$H_{PA} = H_{VPA} \text{ and } H_{PA} - H_{VPA} = 0 \tag{4.9}$$

The $NPSH_R$ is determined by the pump manufacturer and will depend on many factors, including losses in the suction valves of the pump, head needed to propel stationary fluid into motion at the average liquid velocity of the pump inlet port, and so on. The $NPSH_R$ of a reciprocating pump is determined with water at 60°F (15.6°C). The $NPSH_A$ is lowered with the pump at constant speed and throughout until there is a 3% reduction in throughput. The $NPSH_A$ at this point is the specified $NPSH_R$ for this speed. Tests are repeated at different speeds to draw a curve of $NPSH_R$ versus speed.

The $NPSH_A$ depends on the system layout and must always be equal to or greater than the $NPSH_R$. On an existing installation the $NPSH_A$ is the reading of a gauge at the suction flange converted to height of liquid absolute and corrected to the pump centerline elevation less the sum of the vapor pressure of the liquid in height absolute and the velocity head in height of liquid at the point of gauge attachment and the acceleration head.

Field experience and laboratory tests have confirmed that pumps handling gas-free hydrocarbon fluids and water at elevated temperatures will operate satisfactory with little or no cavitation at lower $NPSH_A$ than is required for cold water. Fig. 4.19 shows NPSH reductions that may be considered for hot water and gas-free pure hydrocarbon liquids.

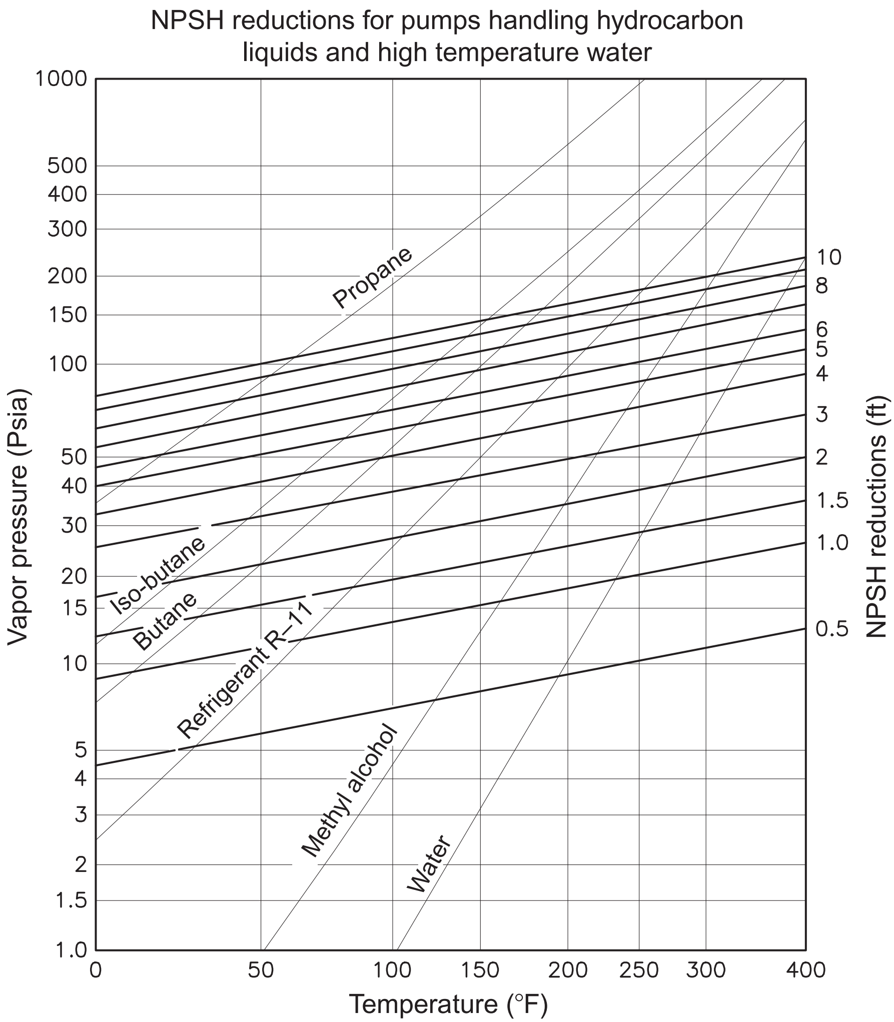


Fig. 4.19 NPSH reductions for pumps handling hydrocarbon liquids and high-temperature water.
Courtesy of Hydraulic Institute.

The use and application of Fig. 4.19 is subject to the following limitations:

(1) The NPSH reductions shown are based on laboratory test data at steady-state suction conditions and on the gas-free pure hydrocarbon liquids shown. Its application to other liquids must be considered experimental and is not recommended.

(2) No NPSH reduction should exceed 50% of the $NPSH_R$ for cold water or 10 ft (3 m), whichever is smaller.

(3) In the absence of test data demonstrating NPSH reductions >10 ft (3 m), the chart has been limited to that extent; extrapolation beyond that is not recommended.

(4) Vapor pressure for the liquid is true vapor pressure at bubble point and not Reid vapor pressure.
(5) Do not use the chart for liquids having entrained air or other noncondensable gases which may be released as the absolute pressure is lowered in the pump suction manifold, in which case additional NPSH may be required for satisfactory operation.
(6) In the use of the chart for high-temperature liquids, particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure which might require additional NPSH to provide a margin of safety, far exceeding the reduction otherwise permitted for steady-state operation.

Subject to the earlier limitations, which should be reviewed with the manufacturer, the procedure in using the chart is illustrated as follows:

Assume a pump with a specified $NPSH_R$ of 16 ft (5 m) as the design capacity is to handle pure propane at 55°F (12.8°C). From Fig. 4.19, propane at 55°F (12.8°C) has a vapor pressure of ~100 psia (689.5 kPa), and a reduction of 9.5 ft (2.9 m) could be considered. This reduction, however, is greater than one half the cold water $NPSH_R$. Thus the corrected value of the $NPSH_R$ is one half the cold water $NPSH_R$ or 8 ft (2.4 m). If this same pump has an application to handle propane 14°F (10°C), where its vapor pressure is 50 psia (345 kPa), a reduction of 6 ft (2 m) could be considered as shown in Fig. 4.19. This is less than one half of the cold water $NPSH_R$. The corrected value of $NPSH_R$ is, therefore, 16 ft less 6 ft, or 10 ft (5 m less 2 m, or 3 m) for this second case.

4.2.6 Acceleration head

Fluid in the piping has to accelerate and decelerate a number of times for each revolution of the crankshaft. Sufficient additional energy must be provided on the suction side of the pump to prevent cavitation. This energy takes away from the energy already existing at the pump suction. The head required to accelerate the fluid column is a function of the suction line, the average velocity in this line, the rotating speed, the type of pump, and the relative elasticity of the fluid. Thus the amount of $NPSH_A$, calculated previously, is reduced by an amount equal to

Field units

$$H_a = \frac{LVNC}{kg} \tag{4.10a}$$

SI units

$$H_A = \frac{60LVNC}{kg} \tag{4.10b}$$

where

H_a = acceleration head, ft (m)
L = actual length of suction line, ft (m)
V = velocity in suction line, ft/s (m/s)
N = speed, rpm (rps)
C = constant for the type of pump (from Table 4.4)
k = constant for relative fluid compressibility (from Table 4.5)
g = acceleration of gravity, 32.3 ft/s^2 (9.81 m/s^2)

Table 4.4 Constant "*C*" used for determining acceleration head

Pump type	"*C*" constant
Simplex, double-acting	0.200
Duplex, single-acting	0.200
Duplex, double-acting	0.115
Triplex	0.066
Quintuplex	0.040
Septuplex	0.028
Nonuplex	0.022

Table 4.5 Constant "*k*" used in determining acceleration head

Fluid(s)	"*k*"
Amine, glycol, water	1.5
Most hydrocarbons	2.0
Hot oils, ethane	2.5

The factor "C" depends upon the type of pump used as follows

The value of "*k*" can be *estimated* from the following:

The acceleration head, where L is ≤ 5 ft (1.5 m) is slightly conservative. When L exceeds 10 ft (3 m) the acceleration head is moderately conservative and becomes overly conservative when L exceeds 50 ft (15 m). The equation assumes piping is rigid and not elastic. The equation does not account for pressure valve velocities.

Example 4.1. Determination of acceleration head (field units)

Given:

A triplex reciprocating pump is required to inject 6000 barrels of salt water per day. The pump operates at 360 rpm. The suction line is 10 ft. of 4-in., Schedule 40 pipe.

Determine:

Calculate the head required to accelerate the fluid through the suction pipe (assume $g = 32.17\,\text{ft/s}^2$).

Solution:

(1) Determine the water displaced by the pump in gpm.

$$q = \frac{(6000)(42)}{(60)(24)}$$
$$= 175\,\text{gpm}$$

(2) Determine the average velocity in the suction piping.

$$v = \frac{(0.4085)(q)}{d^2}$$
$$= \frac{(0.4085)(175)}{4.026^2}$$
$$= 4.41\,fps$$

(3) Determine the acceleration head.

$$H_a = \frac{LVNC}{kg} = \frac{(10)(4.41)(360)(0.066)}{(1.5)(32.17)} = 21.7\,\text{ft}$$

Thus 21.7 ft must be subtracted from the value of calculated $NPSH_A$. The NPSH margin must be provided unless the piping system is accurately defined.

4.2.7 Power and efficiency

The hydraulic power is calculated in a similar fashion as centrifugal pumps. The differences are

- pressure rather than head
- zero suction pressure
- higher efficiencies

The hydraulic power that must be developed by the pump is given by the following equations:

Field units

$$HHP = \frac{(TDH)(\rho)(Q)}{550} \tag{4.11a}$$

SI units

$$HHP = \frac{(TDH)(\rho)(Q)}{367{,}000} \tag{4.11b}$$

where

HHP = hydraulic horsepower, hp (kW), (1 hp = 550 ft-lb/s)
TDH = pump head, ft (m)
ρ = density of liquid, lb/ft^3 (kg/m^3)
Q = flow rate, ft^3/s (m^3/s)

By making the appropriate unit conversions Eqs. (4.11a, 4.11b) may be expressed as:

Field units

$$HHP = \frac{(TDH)(SG)q}{3960} \tag{4.12a}$$

SI units

$$HHP = \frac{(TDH)(SG)q}{367.6} \tag{4.12b}$$

Field units

$$HHP = \frac{q\Delta p}{1714e} \tag{4.13a}$$

SI units

$$HHP = \frac{q(\Delta P)}{3600} \tag{4.13b}$$

Field units

$$HHP = \frac{Q(\Delta P)}{58,766} \tag{4.14a}$$

SI units

$$HHP = \frac{q(\Delta P)}{3600} \tag{4.14b}$$

where

q = capacity, gpm (m^3/h)
Q = capacity, bpd (m^3/h)
ΔP = maximum discharge pressure of pump, psig (kPa) (for driver sizing, pump MAWP is used)
SG = specific gravity relative to water

For design, a conservative assumption is that the driver must be sized for the MAWP of the pump case. This is done in the event of accidental closing of the pump discharge valve.

The input power to the shaft of the pump is called the brake horsepower and is given by

$$BHP = \frac{HHP}{E_m} \tag{4.15}$$

where

BHP = brake horsepower, hp (kW)
E_M = pump mechanical efficiency (%)

Tables 4.6 and 4.7 provide approximate values to be used for mechanical efficiency based on either pump speed or developed pressure.

4.3 Pump classification

There are two types or reciprocating pumps: power pumps and direct-acting pumps. The driver for a power pump has a rotating shaft such as a motor, engine, or turbine. Power pumps reciprocate the pumping element with a crank or camshaft.

Table 4.6 Effect of speed on mechanical energy at constant developed pressure

Percent of full speed	44	50	73	100
E_M (%)	93.3	92.5	92.5	92.5

Table 4.7 Effect of pressure on mechanical efficiency at constant speed

Percent of full speed developed pressure	20	40	60	80	100
E_M (%)	82	88	90.5	92	92.5

Direct-acting pumps are driven by pressure from a motive gas. Direct-acting pumps were originally known as steam pumps because steam was the motive fluid.

Reciprocating pumps are typically classified by

Type of drive

- direct-acting, gas driven (Figs. 4.5 and 4.7)
- crank driven (power pump) (Figs. 4.6 and 4.8)

Cylinder orientation

- horizontal
- vertical (used for higher horsepower requirements)

Liquid end arrangement

- plunger (outside packing)
- piston (inside piston rings and packing on the piston rod)
  - drives the pumping element through a crank or camshaft
  - power sources include *motors and engines*

Number of pistons or plungers

- simples
- duplex
- triplex
- quintuplex, etc.

Type of action

- single-acting (delivers on either forward or backward stroke, not both)
- double-acting (delivers on both forward and backward strokes)

Fig. 4.20 illustrates the classifications. Pumps using reciprocating motion, such as rotary or metering pumps, are not discussed here.

4.3.1 Single- and double-acting pumps

Single-acting pumps discharge on either the forward or return stroke of the piston or plunger; every cycle of the pump displaces only one volume of liquid. In double-acting pumps, liquid is discharged on both the forward and return stroke of the piston. Plunger pumps are only single-acting; piston pumps can be either single- or double-acting. Fig. 4.21 illustrates these pumps' actions.

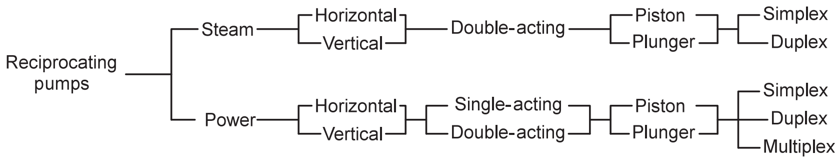


Fig. 4.20 Reciprocating pump types.

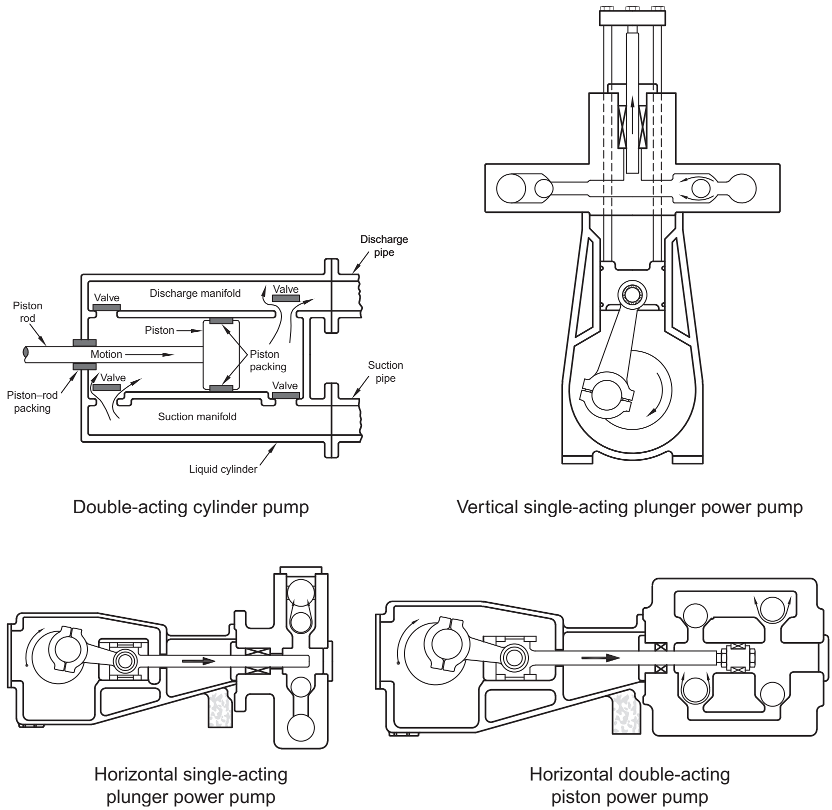


Fig. 4.21 Typical single- and double-acting pumps.
Courtesy of the Hydraulics Institute.

Simplex, duplex, and multiplex refer to the number of piston and rod assemblies in a pump. Simplex pumps have one piston and rod assembly; duplex pumps have two; multiplex pumps have three or more.

Unlike the centrifugal pump, which is a kinetic energy machine, the reciprocating pump does not require velocity to achieve pressure. This is one of the reciprocating pump's advantages, particularly for abrasive, slurry, and high-viscosity applications. High pressures can be obtained at low velocities, and fluid capacity varies directly with pump speed.

The discharge pressure of a reciprocating pump is only that required to force the desired volume of liquid through the discharge system. Within the constraints of pump construction, the maximum pressure developed for gas-driven pumps is limited only by the differential gas pressure available; for crank-driven pumps, the driver torque is the only limit.

The flow of liquid from a reciprocating pump pulsates, varying both in flow rate and pressure. As the piston or plunger moves back and forth in the cylinder, alternatively opening and closing the suction and discharge valves, a cyclic pulsation is set up in the suction and discharge lines of the pump. Fig. 4.22 shows the changes in flow rate

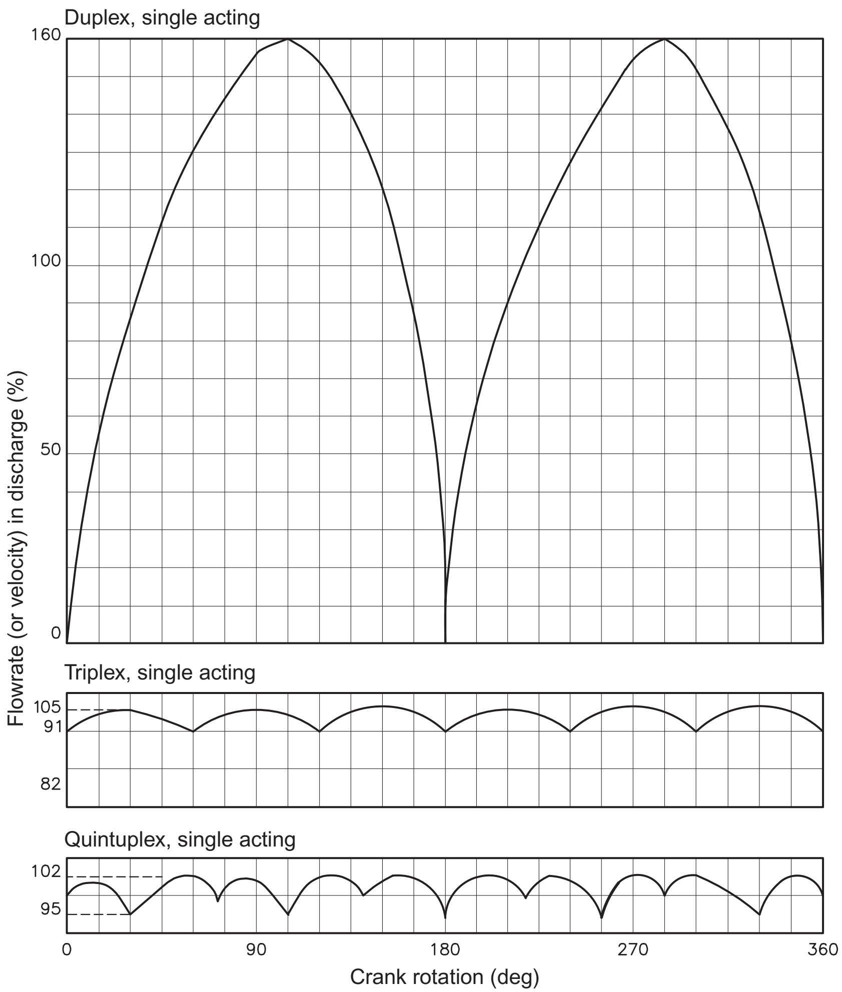


Fig. 4.22 Flow rate per stage.

as a function of crank angle for duplex, triplex, and quintuplex single-acting pumps. These changes become less severe as the number of stages increases.

4.3.2 Reciprocating pump selection charts

Pump selection charts is the equivalent to the centrifugal pump performance curve (Fig. 4.23A). Different manufacturers provide different forms. All charts consist of straight, capacity-speed lines because displacement is a direct function of speed. Speed is plotted against displacement for different piston/plunger diameters available in the pump. The maximum discharge pressures that each size piston/plunger could operate in order to fully stress the piston/plunger rod are listed across the top of the chart.

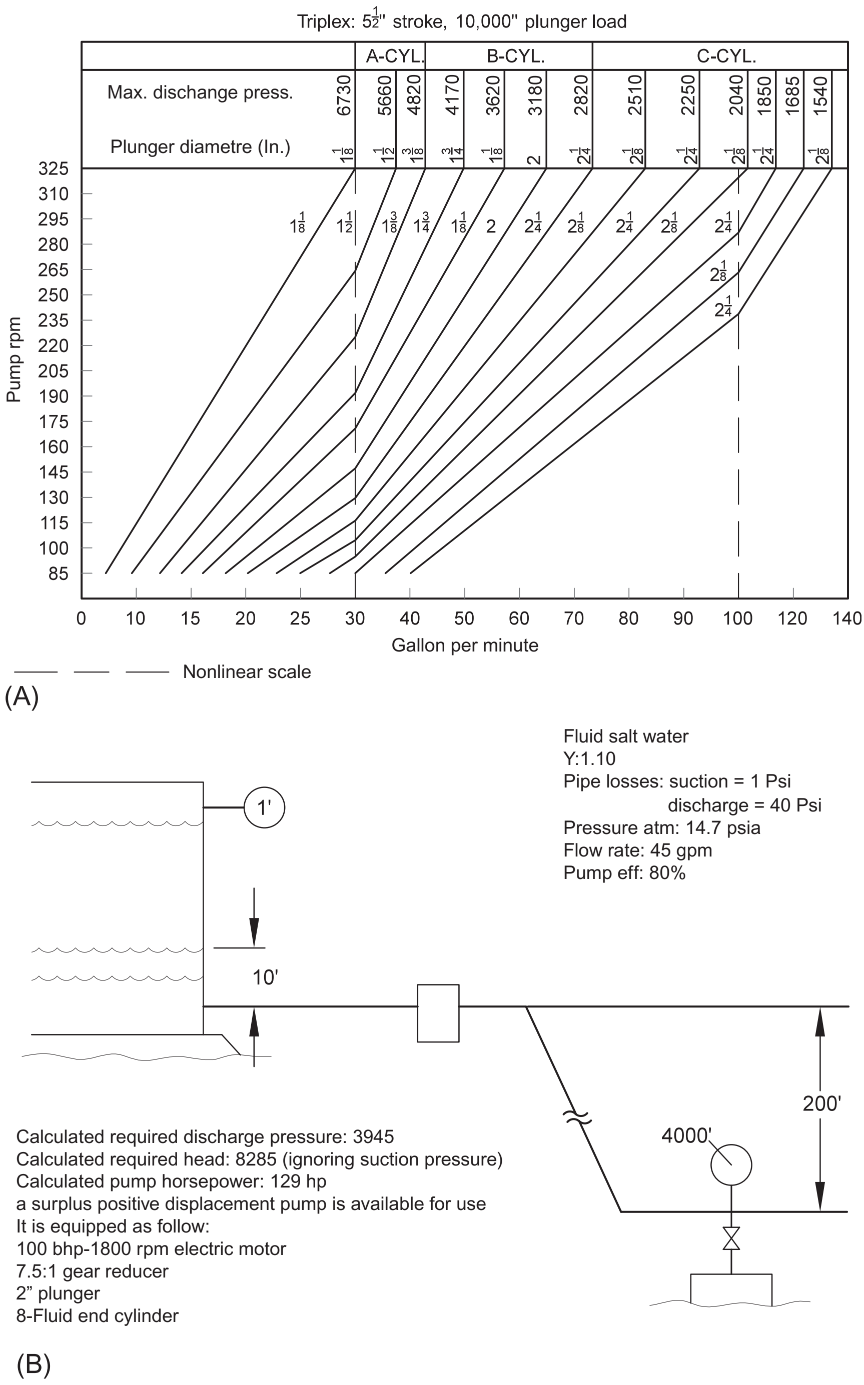


Fig. 4.23 (A) Typical reciprocating pump selection chart and (B) Example 4.2 problem schematic.

The plunger/piston load is the force transmitted to the power end by one piston/plunger for a single-acting pump and it can be calculated as

$$L = AP \tag{4.16}$$

where

L = plunger/piston load, Ibs (N)
A = area of face of plunger/piston, in^2 (cm^2)
P = discharge pressure, psig (kPa)

Example 4.2. Positive displacement pump selection (field units)

Given:

A positive displacement pump is required to waterflood a reservoir to increase oil production.

Determine:

Using Fig. 4.23A and B determine:

(a) The capacity and maximum discharge pressure of the pump.
(b) Does the pump, as currently configured, meet the system requirements? If not, what can be done?

Solution:

(1) Determine the surplus pump's operating speed.

$$\frac{1800}{7.5} = 240\,\text{rpm}$$

(2) Using Fig. 4.23A, determine the plunger's maximum capacity.
At 240 rpm, read 49 gpm. This exceeds the 45 gpm requirement.
(3) Determine if the pump meets the required discharge pressure of 3945 psi.
From Fig. 4.23A, the maximum plunger discharge pressure = 3180 psi. Thus the pump as configured *will not* provide the required injection pressure.
(4) Determine what can be done to meet the injection pressure requirement.
   a. Change the plunger size to obtain an injection pressure ≥3945 psi. Using Fig. 4.23A, a 1-3/4-in. diameter plunger has a 4170 psi MAWP.
   b. From Fig. 4.23A however, the 1-3/4-in. diameter plunger only produces 37 gpm at 240 rpm, and the flow rate requirement is 45 gpm. To obtain 45 gpm, the surplus pump's speed must be increased to 283 rpm.
(5) Determine the required horsepower for the new conditions in (4b) above.

$$BHP = \frac{(45\,\text{gpm})(4170\,\text{psi})}{(1714)(0.80)} = 137$$

Since 137 *BHP* motors are not standard, the next larger, standard motor is specified. Thus a new 150± *BHP* motor is needed. This is now a matter of project timing, costs, and economics whether to use the revamped surplus pump or to purchase a new packed unit.

4.4 Pump construction details

The mechanical components of a reciprocating pump vary according to design. Pumps are built in both horizontal (Fig. 4.24) and vertical (Fig. 4.25) construction. Horizontal pumps are available with capacities to 600 gpm (136 m^3/h) and discharge pressures up to 25,000 psi (172,000 kPa). Vertical pumps are available with capacities up to 5000 gpm (1140 m^3/h) and discharge pressures up to 10,000 psi (69,000 kPa).

The motor-driven or power input end of the pump is normally called the power end, and the pumping end is called the liquid end or fluid end. As shown in Figs. 4.26 and 4.27, the power end contains the following:

- frame
- crankshaft
- connecting rod
- crosshead
- pony rod
- bearings

As shown in Fig. 4.28, the liquid end contains the following:

- plunger
- piston
- stuffing box
- manifold
- cylinder

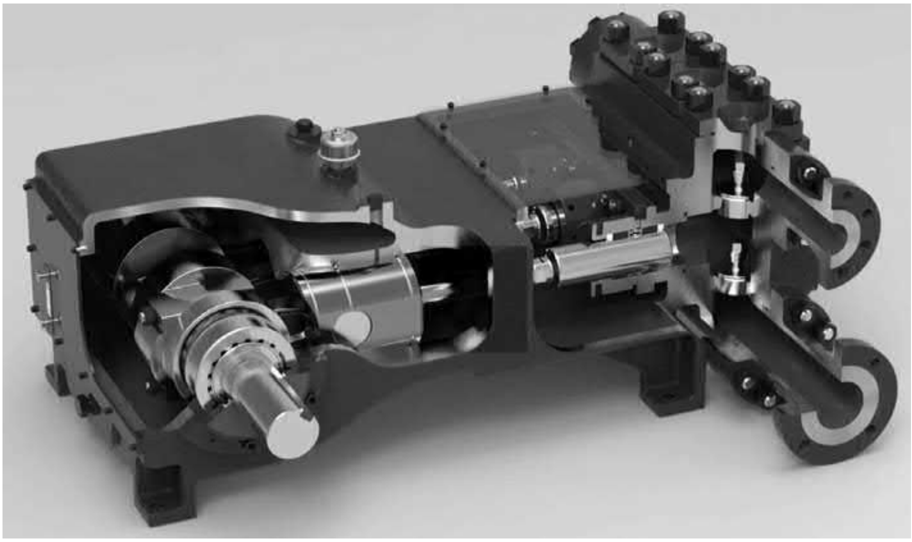


Fig. 4.24 Cutaway view of a typical horizontal pump.
Courtesy of GASO Pumps.

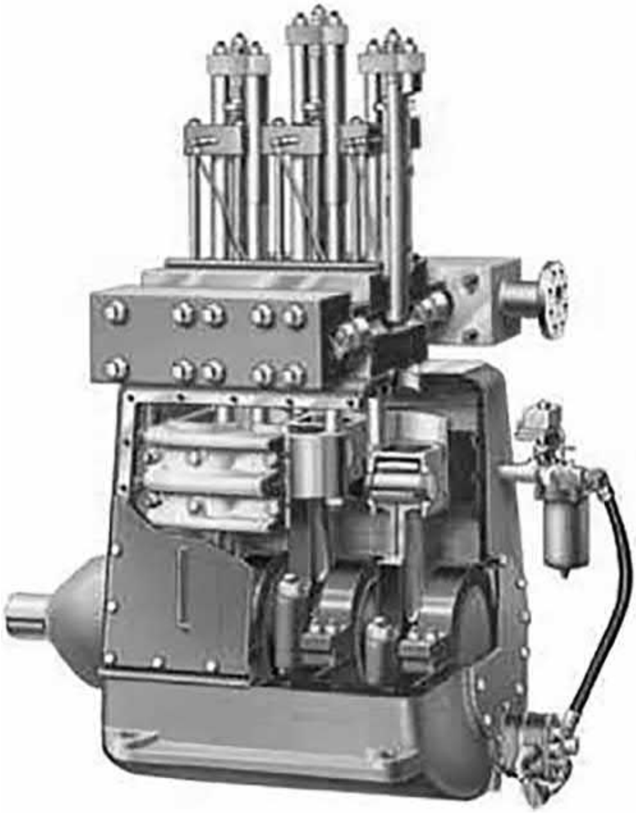


Fig. 4.25 Cutaway view of a typical vertical pump. Courtesy of Ingersoll-Rand Company.

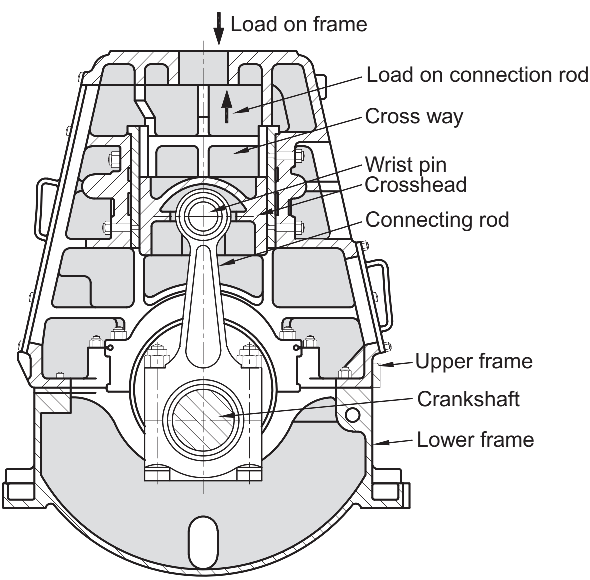


Fig. 4.26 Schematic diagram of the power end of a vertical pump. Courtesy of Ingersoll-Rand Company.

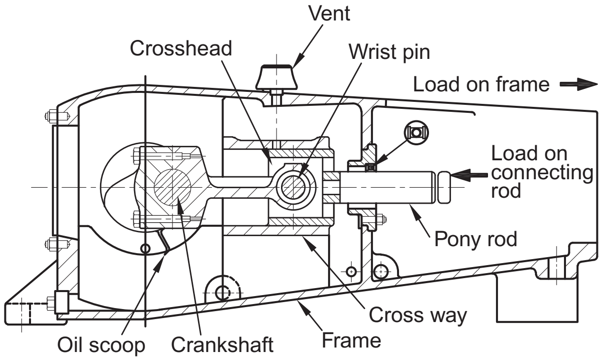


Fig. 4.27 Schematic diagram of the power end of a horizontal pump. Courtesy of Ingersoll Rand Company.

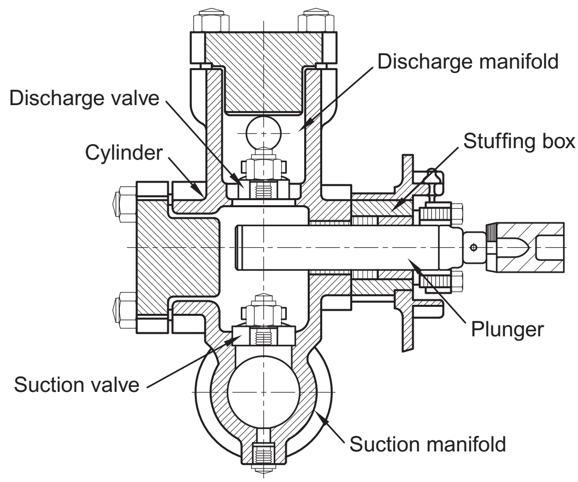


Fig. 4.28 Schematic diagram of the liquid end of a horizontal pump. Courtesy Ingersoll Rand Company.

4.4.1 Power end components

4.4.1.1 Frame

The frame absorbs the plunger load and torque. On vertical pumps with an outboard stuffing box (Fig. 4.29), the frame is in compression. Lubrication is achieved by using an integral rotary gear pump in a wet sump much the same as in an engine. To avoid contamination of the lube oil with dirt, water, or pumped fluid, the crankcase is completely isolated by providing wiper packing for the rods and usually a chamber to accumulate leakage.

With horizontal single-acting pumps, the frame is in tension (Fig. 4.27). Frames usually are close-grain cast iron. Slurry pump frames, designed for mobile service at various sites, usually are fabricated steel. The frame is vented to the atmosphere. When the working atmosphere is detrimental to the working parts in the frame (e.g., ammonia attack on bronze bearings), the frame may be purged continuously with nitrogen.

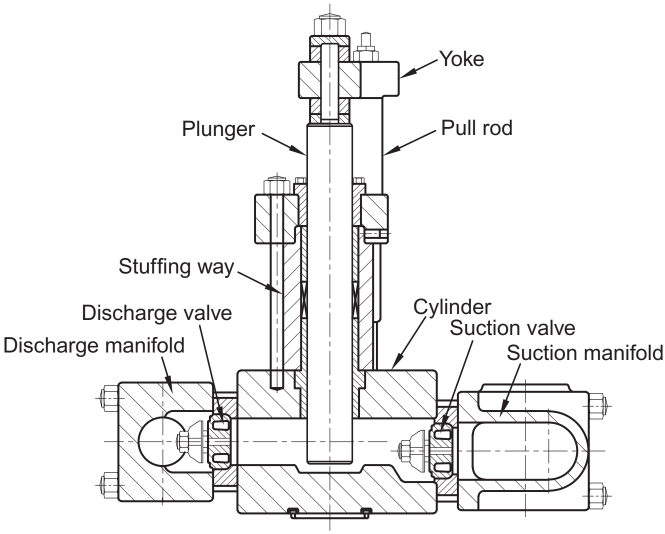


Fig. 4.29 Schematic diagram of the liquid end of a vertical pump.
Courtesy of Ingersoll Rand Company.

4.4.1.2 Crankshaft

The crankshaft (Figs. 4.30–4.32) varies in construction depending on design and power output. In horizontal pumps, the crankshafts are usually constructed of nodular iron or cast steel. Vertical pumps are constructed of forged steel or machined billet for high pressure. Since crankshafts operate at relatively low speeds and mass, counterweights are not used. Except in the case of duplex pumps, crankshafts are usually made with an odd number of throws to obtain the best pulsation characteristics. The firing order in a revolution depends upon the number of throws on the crankshaft, as presented in Table 4.8. Many designs incorporate rifle drilling between throws, thus furnishing lubrication to the crankpin bearings.

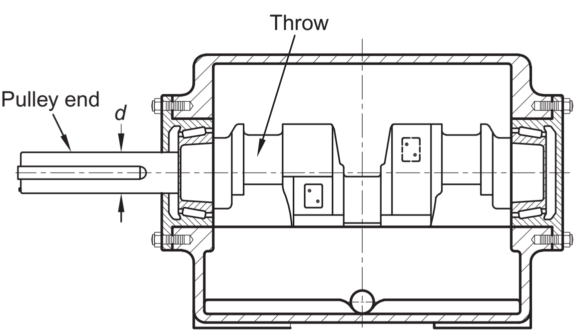


Fig. 4.30 Schematic diagram of a cast crankshaft.
Courtesy of Ingersoll Rand Company.

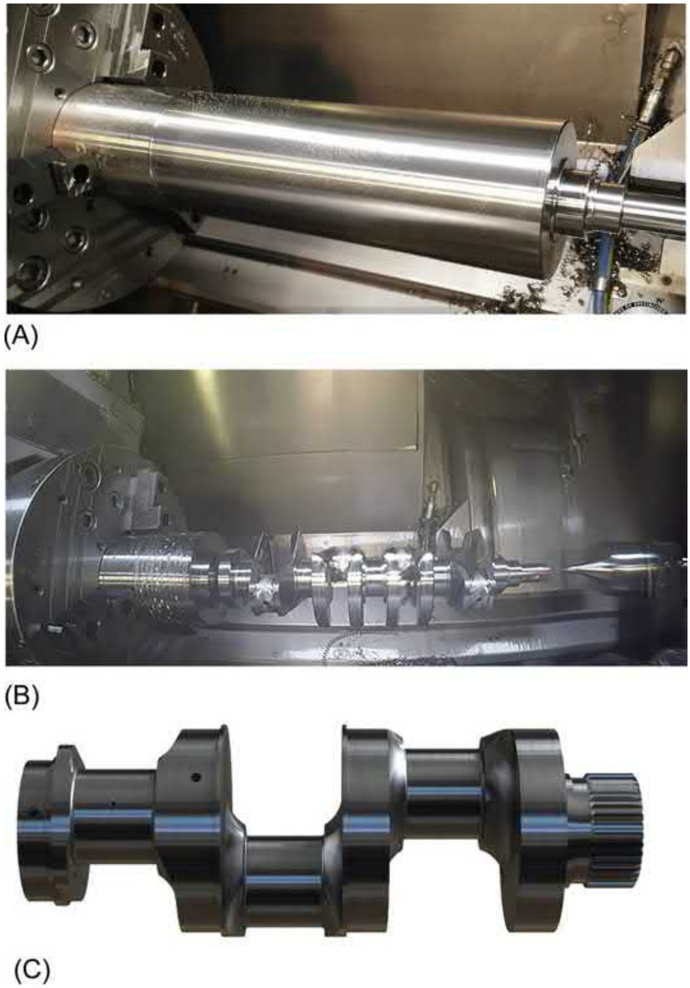


Fig. 4.31 Example of a crankshaft machined from a billet. Top: prior to machining; Center: billet during the machining process; and Bottom: final product.
Courtesy of Ingersoll Rand Company.

Fig. 4.32 Example of a cast crankshaft with an integral gear.
Courtesy of Continental EMSCO.

Table 4.8 Order of pressure buildup, or firing order

Throw from pulley end	No. of plungers or pistons	Pressure buildup order								
		1	2	3	4	5	6	7	8	9
Duplex	2	1	2							
Triplex	3	1	3	2						
Quintuplex	5	1	3	5	2	4				
Septuplex	7	1	4	7	3	6	2	5		
Nonuplex	9	1	5	9	4	8	3	7	2	6

4.4.1.3 Connecting rods and eccentric straps

The connecting rods (Fig. 4.33) transfer the rotating force of the crank pin to an oscillating force on the wrist pin. Connecting rods are split perpendicular to their centerlines at the crank pin end for assembly of the rod onto the crankshaft. The cap and rod are aligned with a close-tolerance bushing or body-bound bolts. The rods may be rifle drilled or they may have cast passages for transferring oil from the wrist pin to the crank pin. A connecting rod with a tension load is made of forged steel, cast steel, or fabricated steel. Rods with a compression loading are cast nodular steel or aluminum alloy.

The ratio of the distance between the centerlines of the wrist pin and of the crank pin bearings to half the length of the stroke is referred to as "*L*/*R*." The ratio directly affects the pressure pulsations, volumetric efficiency, size of pulsation dampener, speed of liquid separation, acceleration head, moment of inertia forces, and frame size. Low "*L*/*R*" results in high pulsations. High "*L*/*R*" reduces pulsations but may result in a large and uneconomical power frame. The common industrial "*L*/*R*" range is 4:1 to 6:1.

The eccentric strap (Fig. 4.34) has the same function as a connecting rod except that the former usually is not split. The eccentric strap is furnished with antifriction (roller or ball) bearings, whereas connecting rods are furnished with sleeve bearings. Eccentric straps are applied to mud and slurry pumps, which are started up against full load without requiring a bypass line.

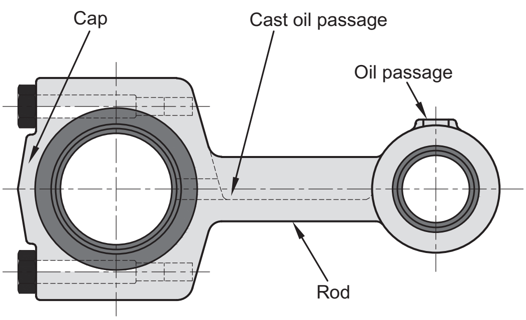


Fig. 4.33 Schematic diagram of a connecting rod.
Courtesy of Ingersoll Rand Company.

Fig. 4.34 Example of an eccentric strap.
Courtesy of Gardner Denver.

4.4.1.4 Wrist pin

The wrist pin is located in the crosshead and it transforms the oscillating motion of the connecting rod to reciprocating pumps. The maximum stress in the wrist pin from deflection is normally limited to 10,000 psi (69,000 kPa). The stress can be calculated from:

Field units

$$S = \frac{PL \times l}{8 \times (0.098)(d_w)^3} \tag{4.17a}$$

SI units

$$S = \frac{1275(PL \times l)}{(d_w)^3} \tag{4.17b}$$

where

S = stress, psi (kPa)
PL = plunger load, lb (kg)
l = length under load, in. (mm)
d_w = diameter of wrist pin, in. (mm)

Large pins are made hollow to reduce the oscillating mass and assembly weight. Depending on the design, pins can have a tight straight fit, a taper fit, or a loose fit in the crosshead. When needle or roller bearings are used for the wrist pin bearing, the wrist pin is used as the inner race.

Fig. 4.35 Example of a crosshead.
Courtesy of Gardner Denver.

4.4.1.5 Crosshead

The crosshead (Fig. 4.35) moves in a reciprocating motion and transfers the plunger load to the wrist pin. It is designed to absorb the side, or radial, load from the plunger as the crosshead moves linearly on the crossway. The side load is ~25% of the plunger load. For cast iron crossheads the allowable bearing load is normally 80–125 psi (551–862 kPa). Crossheads are grooved for oil lubrication with a normal bearing surface of 63 rms. Crossheads are piston type (full round) or partial contact type. The piston type should be open end or vented to prevent air compression at the end of the stroke.

On vertical pumps the pull rods go through the crosshead so that the crosshead is under compression when the load is applied (Fig. 4.26).

4.4.1.6 Crossways

The crossway (Fig. 4.36) is the surface on which the crosshead reciprocates. On horizontal pumps it is cast integral with the frame (Fig. 4.27). On large frames it is usually replaceable and is shimmed to effect proper running clearance (Figs. 4.26 and 4.36). The crossway finish is normally 63 rms.

4.4.1.7 Pony rod (intermediate rod)

The pony rod is an extension of the crosshead on horizontal pumps (Fig. 4.27). It is screwed or bolted to the crosshead, and it extends through the "A" seal on the frame and against the pony rod preventing oil from leaking out of the frame. A baffle is fixed onto the rod to keep leakage from coming in contact with the frame (Fig. 4.27).

4.4.1.8 Pull rod (tie rod)

On vertical pumps two pull rods go through the crosshead. The rods are secured by a shoulder and nut so that the cast iron crosshead is in compression when the load is applied (Fig. 4.26). The rods extend out of the top of the frame and fasten to a yoke. The plunger is attached to the middle of the yoke with an aligning feature for the plunger (Fig. 4.29).

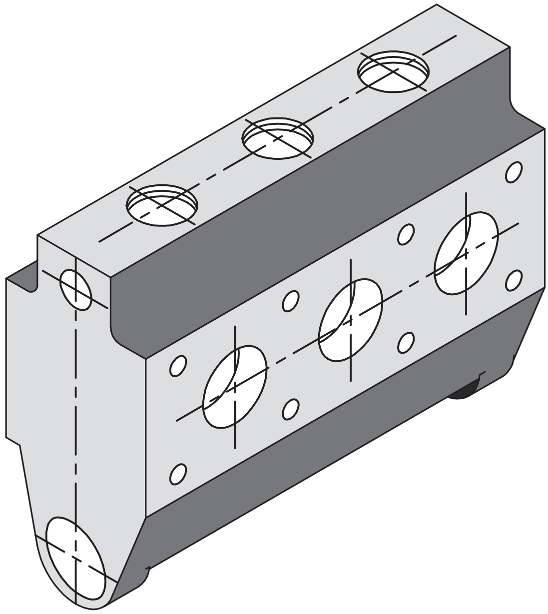


Fig. 4.36 Schematic diagram of a single cylinder.
Courtesy of Hydraulic Institute.

4.4.2 Fluid end components

The liquid end consists of the cylinder, plunger or piston, valves, stuffing box, manifolds, and cylinder head. Fig. 4.28 shows the liquid end of a horizontal pump, and Fig. 4.29 shows the same for a vertical pump.

4.4.2.1 Cylinder (working barrel)

The cylinder is the body in which the pressure is developed. It is continuously under stress. Cylinders on many horizontal pumps have the suction and discharge manifolds made integral with the cylinder. Vertical pumps usually have separate manifolds. A cylinder containing the passages for more than one plunger is referred to as a single cylinder (Fig. 4.36).

When the cylinder is used for only one plunger, it is called as individual cylinder (Fig. 4.37). Individual cylinders are used when developed stresses are high.

Cylinder allowable stresses range from 10,000 to 25,000 psi (69,000 to 172,000 kPa) depending on the material of the cylinder and the liquid being pumped. The allowable stress is a function of the fatigue stress of the material for the liquid pumped and the lifecycles required.

4.4.2.2 Plunger

The plunger transmits the force that develops the pressure. It is normally solid up to 5-in (127 mm) diameter. At large sizes, it may be made hollow to reduce weight. Small-diameter plungers used for 6000 psi (41,400 kPa) and higher should be

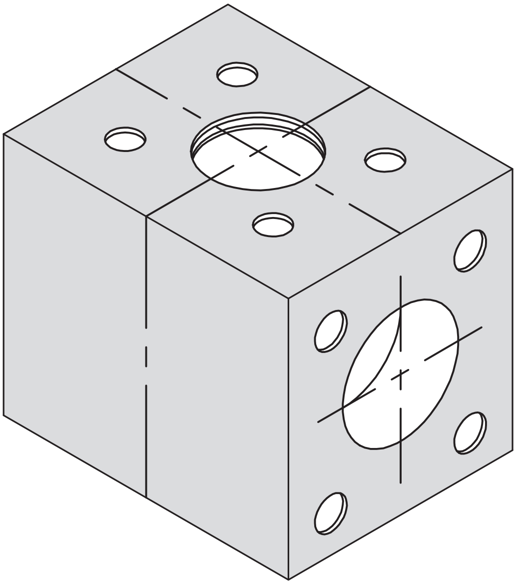


Fig. 4.37 Schematic diagram of an individual cylinder.
Courtesy of the Hydraulic Institute.

reviewed for possible buckling. Plunger speed ranges from 150 to 350 ft/min (46 to 107 m/s). The finish is normally 16 rms with a 30 to 58 Rockwell C hardness. Materials of construction are Colmonoy No. 6 on 1020, chrome plate on 1020, 440°C, 316, ceramic on 1020 with 93°C (200°F) limit, and solid ceramic. For $<$3000-psi (20,700 kPa) discharge pressure, ceramic is used for soft water, crude oil, mild acids, and mild alkalis. The problem with coatings on plungers is that at high pressures the liquid gets through the pores and underneath the coating. Pressure underneath the coating may cause it to flake off the plunger.

4.4.2.3 Pistons

Pistons are used in double-acting pumps and in some single-acting pumps for water pressures below 2000 psi (13,800 kPa); for higher pressures a plunger is usually used in single-acting pumps. Pistons are cast iron, bronze, or steel with reinforced elastomer faces (Fig. 4.38).

4.4.2.4 Cylinder liner

The cylinder wear liner (Figs. 4.38 and 4.39) is usually made of Ni-resist material. Its length is slightly longer than the stroke of the pump, allowing for an assembly entrance taper of the piston into the liner. On double-acting pumps, the liner has packing to prevent leakage from the high-pressure side to the low side of the cylinder. Because of the brittleness of the liner, the construction should be such that the liner is not compressed. The finish of the liner is normally 16 rms.

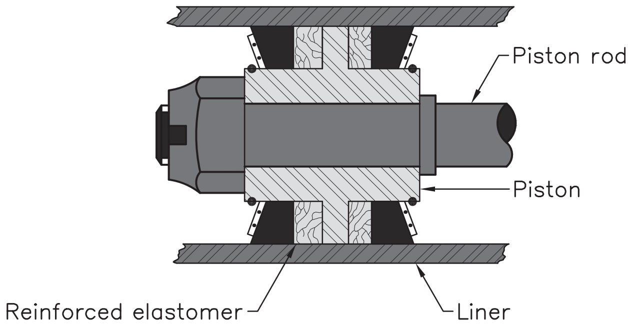
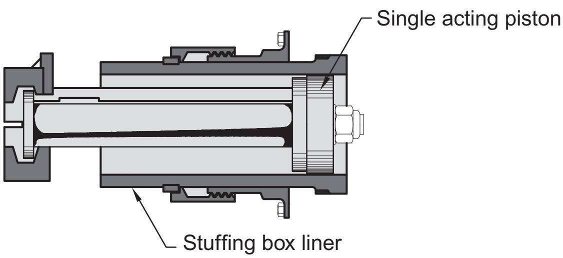


Fig. 4.38 Schematic diagram of a piston with an elastomer face.
Courtesy of FWI.

Fig. 4.39 Schematic diagram of a single-acting piston stuffing box.
Courtesy of Ingersoll Rand Company.



4.4.2.5 Manifolds

These are the chambers in which liquid is dispersed or collected for distribution before or after passing through the cylinder. On horizontal pumps, the suction and discharge manifold is usually made integral with the cylinder (Fig. 4.40).

Some horizontal pumps and some vertical pumps have only the discharge manifold integral with the cylinder (Fig. 4.41). On most vertical pumps, the suction and discharge manifolds are separate from the cylinder.

Suction manifolds are designed to eliminate air pockets from the flange to the valve entrance. Separate suction manifolds are cast iron or fabricated steel. Discharge manifolds are steel forgings or fabricated steel. The manifolds have a minimum deflection to prevent gasket shift when subjected to the plunger load.

4.4.2.6 Stuffing box

The function of the stuffing box is to seal the point where the plunger or piston enters the cylinder. The stuffing box of a pump consists of the box, lower and upper bushing, packing, and gland (Fig. 4.42). It should be removable for maintenance. The stuffing box bore is normally machined to a 63-rms finish to ensure packing sealing and life. A hoop stress calculation is used to determine the stuffing thickness, with an allowance of 10,000 to 20,000 psi (69,000 to 138,000 kPa).

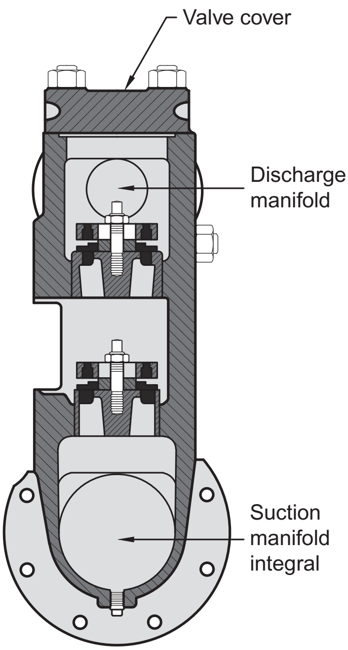


Fig. 4.40 Schematic diagram of an integral suction and discharge manifold.
Courtesy of Gardner Denver.

The bushings normally have a 63-rms finish with a clearance of ~0.001 to 0.002 in per in (0.001 to 0.002 mm per mm) of plunger diameter. The lower bushing is sometimes secured in an axial position to prevent the working of the packing. Bushings are made of bearing bronze, Ni-resist, or 316 stainless steel.

Stuffing box packing is either square cut (Figs. 4.43 and 4.44) or chevron (V-ring) (Fig. 4.45).

Some types use metal backup adapters. Unit packing consists of top and bottom adapters with a seal ring. A stuffing box can use three to five rings of packing or units, depending on the pressure and on the fluid being pumped. Packing or seal rings are usually made of reinforced asbestos, teflon, or neoprene.

Packing can be made self-adjusting by installing a spring between the bottom of the packing and the lower bushing or bottom of the cylinder. This arrangement eliminates overtightening and allows for uniform break-in of the packing. The packing is lubricated by injecting grease through a fitting, by gravity oil feed, or by an auxiliary lubricator driven through a takeoff on the crankshaft.

For chemical or slurry service a lower injection ring is used for flushing. This procedure prevents concentrated pumped fluid from impinging directly on the packing. The injection can be a continuous flush or can be synchronized to inject only on the suction stroke. Flush glands are employed where toxic vapor or flashing appears after the packing.

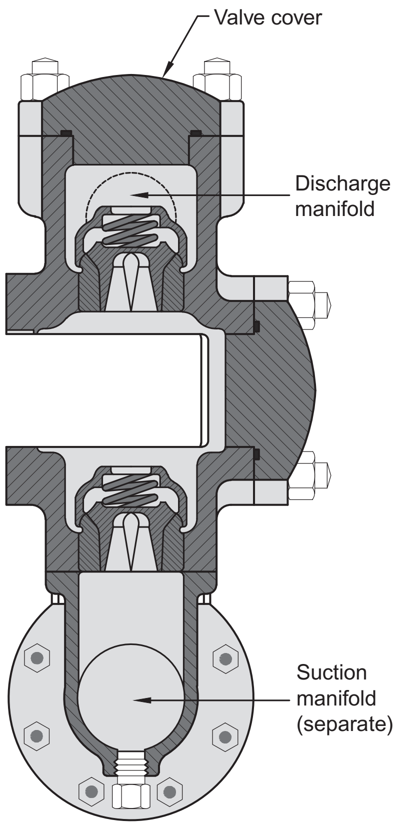


Fig. 4.41 Schematic diagram of a separate suction with integral discharge manifold.

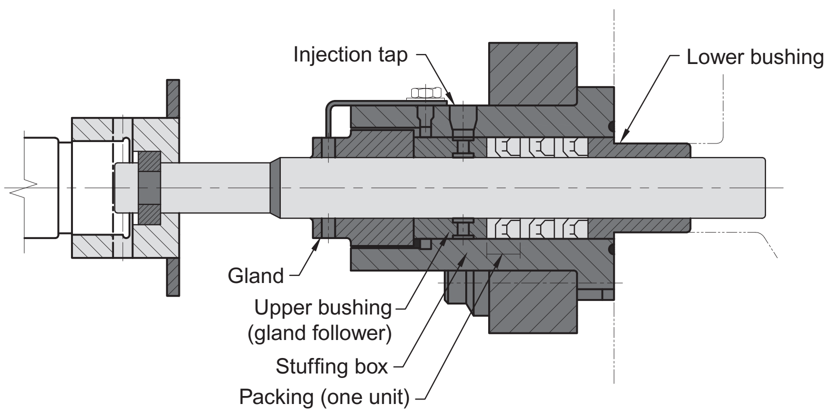


Fig. 4.42 Schematic diagram of a stuffing box.
Courtesy of Ingersoll Rand Company.

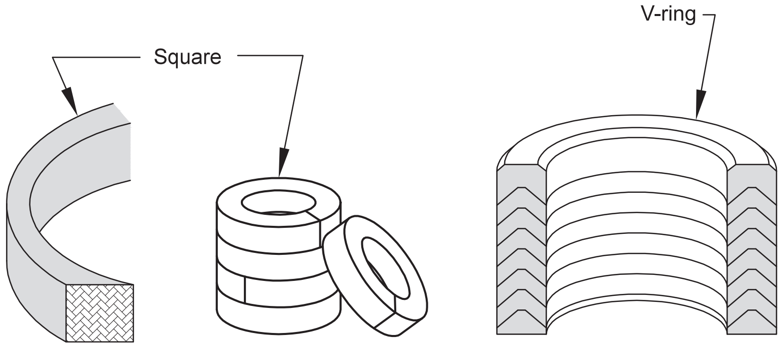


Fig. 4.43 Schematic diagram of square cut and chevron (V-ring) packing. Courtesy of Hydraulic Institute.

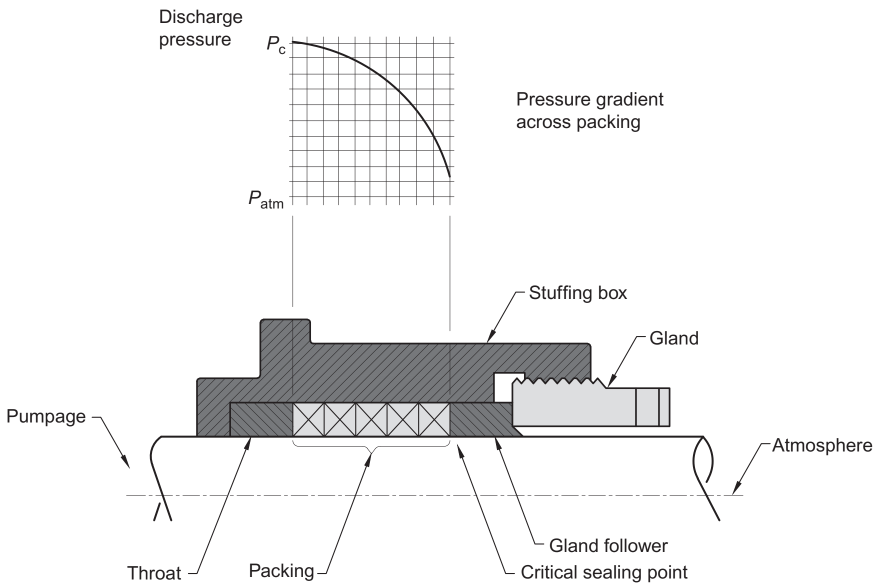


Fig. 4.44 Schematic diagram of the square cut packing arrangement. Courtesy of Hydraulic Institute.

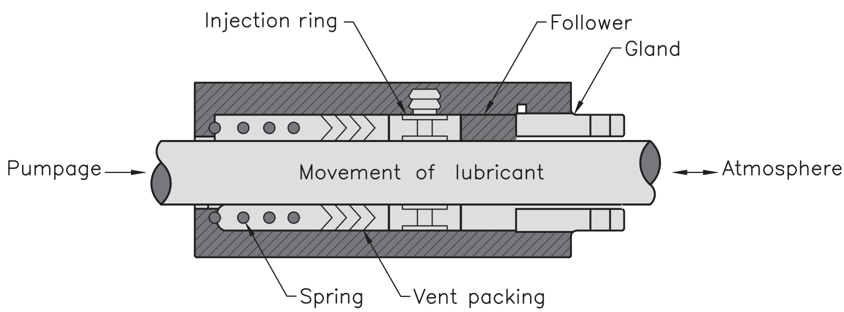


Fig. 4.45 Schematic diagram of the Chevron packing arrangement.

The stuffing box of a double-acting piston does not require an upper and lower bushing because the piston is guided by the cylinder liner. Single-acting pistons do not employ a stuffing box. Leakage of the piston goes into the frame extension to mix with the stuffing box's continuous circulating lubricant.

The advantages and limitations of square cut packing and chevron (V-ring) packing is as follows:

4.4.2.6.1 Square cut packing

The advantages of square cut packing are as follows:

- low cost
- smaller overall space required
- easy to repair

Limitations include

- requires continuous adjustment gland
- not good when solids are present in the fluid pumped unless an injection ring is installed

4.4.2.6.2 Chevron packing

Advantages of Chevron packing are as follows:

- Gland does not have to be adjusted as packing wears
- Spring provides a cavity for injection of clean seal fluids
- Less leakage than square cut

Limitations include

- Cost
- Spring cavity can reduce volumetric efficiency for high compressibility fluids as it is in communication with cylinder
- Vapors can accumulate in spring cavity

4.4.2.7 Valves

The valve assembly can be classified according to the manner in which it is installed in the pump (Fig. 4.46). In a caged assembly, the entire valve assembly fits into a straight bore machined into the pump. The seal between the fluid end and the valve assembly is maintained by a gasket. This is the most common design.

A tapered seat valve assembly fits into a taper machined into the fluid end. The fit between the fluid end and assembly is forced. This design is the original design offered by pump manufacturers and is best used for large flow rates. The various types of valves are shown in Fig. 4.47.

Plate, wing, and plug valves are normally limited to clean fluids because of the flow patterns through the open valve. Ball valves are normally applied in high pressure or abrasive service since the line contact at seating surfaces is better suited for abrasive service. The ball is free to rotate, eliminating side load. Plate and plug valves have

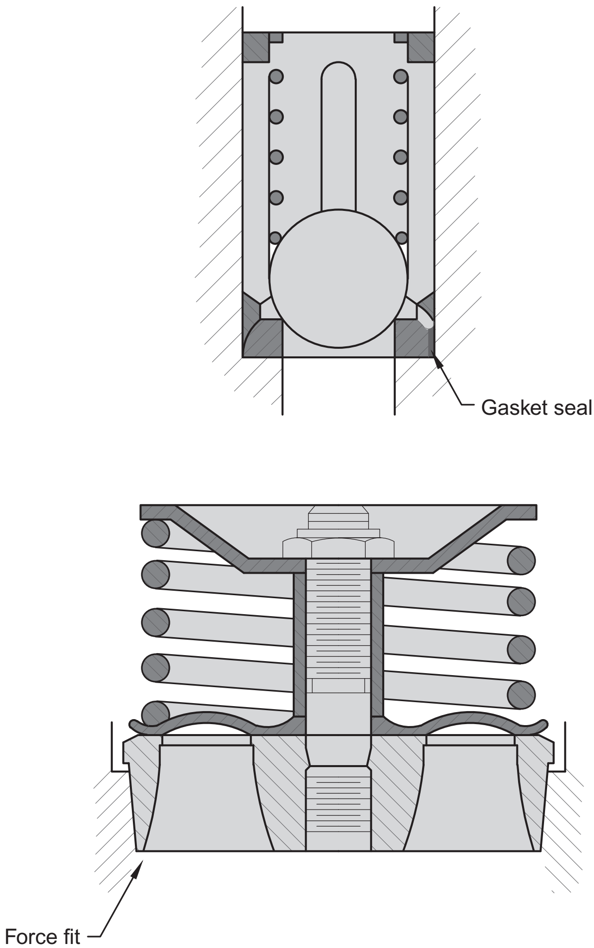


Fig. 4.46 Schematic diagram of the valve assembly classification.

smaller pressure losses than balls, but their flat valve surfaces are not suitable for abrasive surfaces. Wing valves are generally considered to be the best currently available pump valve. Pressure drop through the valves is low. Guided motion prevents flutter. Slurry valves are used in low-pressure service.

Some pumps use the same size suction and discharge valves for interchangeability. Some use larger suction valves than discharge valves for $NPSH_R$ reasons. Because of space considerations, valves are sometimes used in clusters on each side of the plunger to obtain the required total valve area. Table 4.9 presents seat and plate hardness for some valve materials.

Seats and plates made of 316 stainless steel are chrome plated or Colmonoy No. 6 plated to give them surface hardness. Seats and plates normally have a 32-rms finish.

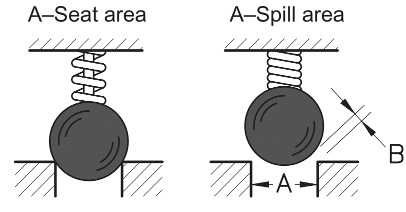
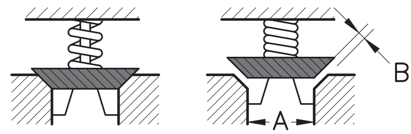
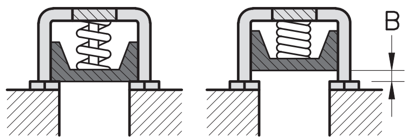
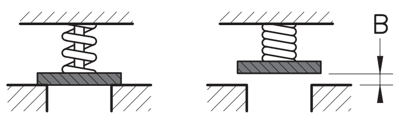
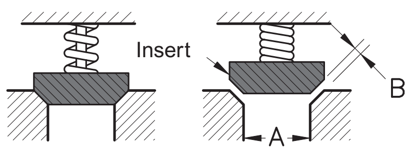
Type	Sketch	Pressure psi (bar)
Ball		30,000 (2069)
Wing		10,000 (690)
Plug		6000 (414)
Plate		5000 (345)
Slurry		2500 (172)

Fig. 4.47 Various types of valves and their pressure applications. Courtesy of Ingersoll Rand Company.

The number and size of valves are chosen so that the velocity through any valve, sometimes referred to as the "spill velocity," is controlled. Some representative velocities are given in Table 4.10. The spill velocity is given by

Field units

$$V = \frac{(\text{gpm through valve})(0.642)}{\text{Spill area of the valve, in}^2} \tag{4.18a}$$

Table 4.9 Recommended material hardness for valve plate and seat

Material	Plate	Seat
Rockwell C hardness		
329	30 to 35	38 to 43
440	44 to 48	52 to 56
17-4 PH	35 to 40	40 to 45
15-5 PH	35 to 40	40 to 45
Brinell hardness		
316	150 to 180	150 to 180

Table 4.10 Typical medium valve spill velocities

Valve	Spill velocity	
	m/s	(ft/s)
Clean liquid suction valve	0.9–2.4	(3–8)
Clean liquid discharge valve	1.8–6	(6–20)
Slurry suction and discharge valve	1.8–3.7	(6–12)

SI units

$$V = \frac{(\text{m}^3/\text{h through valve})(556.0)}{\text{spill area of the valve, mm}^2} \tag{4.18b}$$

where:

V = velocity, ft/s (m/s)

4.4.2.8 Bearings

Bearings are classified as either hydrodynamically lubricated (oil film sleeve type) or rolling contact (antifriction). With the hydrodynamically lubricated bearing the primary motion is a sliding of one surface over another. The surfaces of this bearing are separated by a relatively thin layer of lubricant, which prevents any metallic contact, except when starting or stopping under load. With the roller type, the primary motion is a rolling of one surface over another. Balls or rollers are used to support the applied load by actual metal-to-metal contact over a relatively small area.

The advantages of hydrodynamic lubricated bearings are as follows:

- The bearing life is not a function of speed or load. Very little wear occurs.
- Oil forces tend to hold the shaft in the center of the bearing, providing some degree of dampening.
- Journal bearings and flat plate thrust bearings have very low relative cost.

The limitations of hydrodynamic lubricated bearings are as follows:

- They require a continuous supply of cool, uncontaminated oil, at a constant fixed pressure.
- Journal bearings can tolerate no axial load, while flat plate and Kingsbury bearings can tolerate no radial load.
- They have a relatively high friction drag as opposed to rolling contact bearings and thus have higher horsepower losses.

The advantages of rolling contact bearings are as follows:

- A continuous lubrication system is generally not required. Occasional greasing of the bearing is all that is required.
- They offer very little resistance to movement; therefore power losses are smaller than with hydrodynamic lubricated bearings.
- They can be designed for both thrust and radial loads simultaneously.

The limitations of rolling contact bearings are as follows:

- They have a specific life for the design load.
- They have limited load/speed capability.
- Both hydrodynamic and antifriction rolling contact bearings are used in pumps. Some frames use all hydrodynamic, other use all rolling contact, and others use a combination of both.

4.4.2.8.1 Hydrodynamic bearings

When properly installed and lubricated, hydrodynamic bearings are considered to have infinite life. They are designed to operate within a certain speed range, and too high or too low a speed will upset the film lubrication. They cannot be operated satisfactorily below 40 rpm with the plunger fully loaded, using standard lubrication. Below this speed, film lubrication is inadequate. The finish on the hydrodynamic bearing is normally 16 rms. Clearances are ~0.001 in per in (0.001 mm per mm) of diameter of the bearing.

Wrist pin bearings have only oscillating motion. On single-acting pumps with low suction pressure, there is adequate reverse loading on the bearing to permit replenishment of the oil film. High suction pressure on horizontal pumps increases the reverse loading. On vertical pumps, high suction pressure can produce a condition of no reversal loading, and in this case higher oil pressure is required. Allowable projected area loading is 1200 to 1500 psi (8300 to 10,300 kPa) with bronze bearings.

The crank pin bearing is a rotating split bearing that has a better oil film than the wrist pin bearing. It is clamped between the connecting rod and cap. The bearing is bronze-backed Babbitt metal bearing. Allowable projected area loading is 1200 to 1600 psi (8300 to 11,000 kPa).

The main bearings absorb the plunger load and gear load. The total plunger varies during the revolution of the crankshaft. The triplex main bearings receive the greatest variations in loading because the crankshaft has the greatest relative span between bearings. Sleeve bearings are flanged to lock them in an axial position, and the flange absorbs residual axial thrust. On large-stroke vertical pumps, there is a main bearing between every connecting rod. Split bearings are bronze-backed Babbitt metal with an allowable projected area load of 750 psi (5200 kPa).

4.4.2.8.2 Rolling contact bearings

A pump with rolling contact bearings can be started under full plunger load without a bypass line. Rolling contact bearings allow the pump to operate continuously at low speeds with full plunge load. They are selected for a designated bearing life.

Wrist pin bearings are needle or roller bearings. The outer race is a tight fit into the strap, and the wrist pin is used as the inner race. This reduces the size of the bearing and strap. The wrist pin is held in the crosshead with a taper or keeper plate (Fig. 4.48).

Crank pin bearings are roller bearings. The outer race is mounted separately from the inner race; the inner race is mounted on the crankshaft and secured axially by a shoulder on the shaft and by a keeper plate, or by keeper plates on both sides of the race. The outer race is assembled in the strap in the same way as the inner race mounting. The strap is then slipped over the shaft and assembled to the inner race.

The main bearings used in conjunction with the hydrodynamic wrist bearing and the hydrodynamic crank pin bearings are usually tapered roller bearings, as in the case of a horizontal triplex plunger pump. The main bearings are ordinarily mounted directly into the frame (Fig. 4.49).

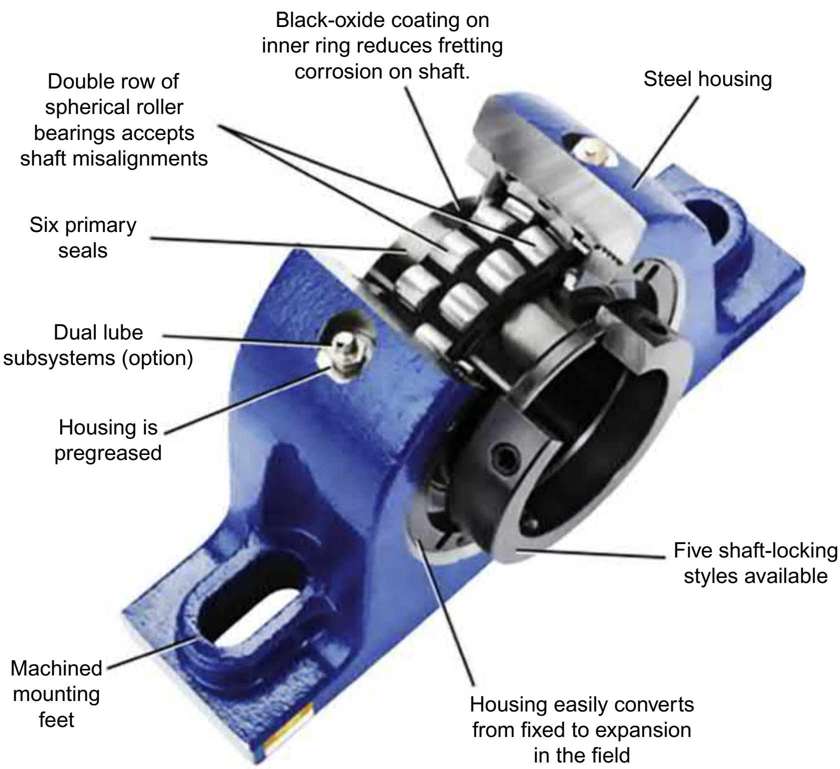


Fig. 4.48 Schematic diagram of a main bearing mounted in a bearing holder.

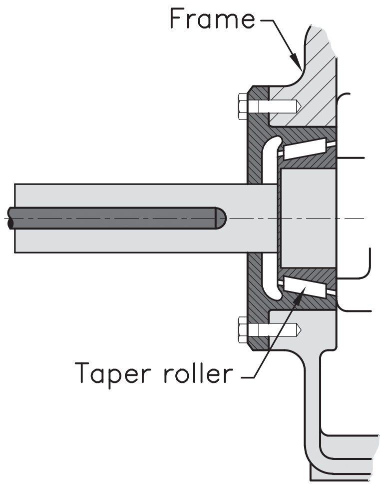


Fig. 4.49 Schematic diagram showing the main bearing mounted directly in the frame. Courtesy of Ingersoll Rand Company.

The main bearings used with full rolling contact bearing design are self-aligning spherical roller bearings. These bearings compensate for axial and radial movement of the crankshaft and are usually mounted in bearing holders, which in turn are mounted on the frame (Fig. 4.48). This type of design is used on mud and slurry pumps.

4.4.2.9 Lubrication system

The power end of large slow-speed positive displacement pumps operating at speeds below 160 to 165 rpm is normally furnished with pressurized (force-feed) lubrication systems, while smaller positive displacement pumps operating at speeds above 165 rpm are normally furnished with splash-type lubrication systems. Splash-type lubrication systems use checks on the crankshaft or oil scoops to throw oil by centrifugal force against the frame wall. This oil is then distributed by gravity to the crosshead, wrist pin, and crank pin. Pressurized or force-feed lubrication systems may use either a built-in direct drive lube oil pump or an external direct drive or motor-driven lube pump to lubricate power end components. Force-feed lubrication systems give better lubrication but are more expensive to purchase.

4.5 Reciprocating pump standards

4.5.1 Overview

Most reciprocating pumps used in oil field service are built to manufacturer's standards, although they have many features required by *API Standard 674 (Positive Displacement Pumps Reciprocating)*. The basic requirements of *API*

Standard 674 are summarized as follows to provide some guidance in making selections of various options:

- *Pressure ratings*: All pressure and temperature ratings encountered in oil field production operations.
- *Pump speed*: Speeds for single-acting plunger pumps are limited to the values presented in Table 4.11.
- *Cylinder design*: Requires ASME code Section VIII Division 1 for cylinders. Forged cylinders required for applications above 3000 psi (20,700 kPa).
- *Cylinder liner*: Required.
- *Plunger or rods*: Requires that all piston rods or plungers in contact with packing be hardened or coated to Rockwell C35.
- *Stuffing boxes*: Required to accept at least three packing rings.
- *Bearings*: Ball bearings must be capable of three years continuous operation at rated pumping conditions.
- *Materials*: Cast iron or ductile iron is not allowed for pressure-retaining parts handling flammable or toxic fluids.
- *Testing*: Pressure-retaining parts (including auxiliaries) are tested hydrostatically with liquid at a minimum of 1.1 times the maximum allowable working pressure.

4.5.2 Water injection pumps—NACE RP-04-75

The National Association of Corrosion Engineers has published their "Standard Recommended Practice for the selection of Metallic Materials to be used in All Phases of Water Handling for Injection into Oil Bearing Formations." This standard provides guidelines for selecting materials used in water handling equipment of all types, with one section devoted to pumps. This is often a useful reference when specifying pumps or reviewing manufacturer proposals for water handling and injection services.

4.5.3 General oilfield standards

Table 4.12 presents a comparison of some of the major requirements of API STD 674 compared to manufacturer's standards generally used in oil field service. It can be seen that the API standard is more stringent in design requirements and quality control and

Table 4.11 API Standard 674—Maximum allowable speed

Stroke length		RPM	Max allowable speed	
mm	**(in.)**	**rpm**	**m/s**	**(ft/min)**
50.8	(2)	450	0.76	(150)
76.2	(3)	400	1.02	(200)
101.6	(4)	350	1.18	(233)
127.0	(5)	310	1.31	(258)
152.4	(6)	270	1.37	(270)
177.8	(7)	240	1.42	(280)
203.2	(8)	210	1.42	(280)

Table 4.12 Comparison of API Standard 674 and normal oilfield specifications

	API Standard 674	**Normal oilfield specifications**
Pump casings rating	110% of the design discharge pressure	120% of the design discharge pressure
Pump speed	Speeds for single-acting plunger pumps are limited to the values presented in Table 4.11	85% of manufacturer's maximum continuous rpm rating
Cylinder design	Requires ASME Code Section VIII, Division 1 for cylinders. Forged cylinders required for applications above 3000 psi (20,700 kPa)	No requirements
Plunger or rods	Requires that all piston rods or plungers in contact with packing be hardened or coated to Rockwell C35	Hardened in the packing area
Stuffing boxes	Required to accept at least three packing rings	No requirements
Bearings	Ball bearings must be capable of three years continuous operation at rated pumping conditions	No requirements
Materials	Cast iron or ductile iron is not allowed for pressure-retaining parts handling flammable or toxic fluids	Forged steel fluid end
Testing	Pressure-retaining parts (including auxiliaries) are tested hydrostatically at a minimum of 11/2 times the maximum allowable working pressure	No requirements

is thus normally used only for critical services where reliability is important. Pumps for normal oil field applications are less expensive and much more readily available. Whenever service conditions allow considerable time, cost savings are possible by specifying manufacturer's standards.

4.5.4 Data sheets

API has a standard 4-page data sheet for pumps. An example copy may be found in API Standard 674.

4.6 Materials of construction

The choice of materials used for pump parts varies with operating conditions and liquids handled. Tables 4.13–4.15 from *API Standard 674* list combinations of metals most commonly used in pumps. Table 4.16 from *API Standard 610* may be used as a guide in selecting material classes for pumps. Table 4.17 and Table 4.18 from *NACE RP-0475* present pump materials for water service.

Table 4.13 Materials and material specifications for major component parts

Materials for reciprocating pump liquid end parts[a] Material class and material class abbreviations[b]				
API material class code Part	I-1 CI/CI	I-2 CI/BRZ	S-1 STL/CI	S-2 STL/HI ALLOY
Pressure-retaining parts				
Cylinder	Cast iron	Cast iron	Carbon steel	Carbon steel
Cylinder head, valve cover	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Stuffing box	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Gland	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Bolting	Carbon steel	Carbon steel	AISI 4140 steel	AISI 4140 steel
Nonpressure-retaining parts				
Lantern ring, throat bushing, packing follower	Cast iron, 12% chrome, or any stainless steel	Cast iron or bronze	Cast iron, 12% chrome, or any stainless steel	12% chrome, or any stainless steel
Valve and seat	Stainless steel	Bronze	Carbon steel or any stainless steel	12% chrome, or any stainless steel
Valve spring	Stainless steel	Bronze or stainless steel	Stainless steel	Inconel or stainless steel
Piston[c]	Cast iron	Cast iron	Cast iron	Ni-resist
Piston rod[c]	12% chrome	12% chrome	12% chrome	12% chrome
Piston ring[c]	Cast iron or nonmetallic	Nonmetallic	Cast iron or nonmetallic	Special cast iron or Ni-resist
Piston ring expander[c]	Cast iron or stainless steel	Stainless steel	Cast iron or stainless steel	Stainless steel
Plunger (power pumps)[c]	12% chrome, hard surfaced	12% chrome, hard surfaced	12% chrome, hard surfaced	12% chrome, hard surfaced
Plunger (direct-acting pumps)[c]	12% chrome, hardened	12% chrome, hardened	12% chrome, hardened	12% chrome, hardened
Cylinder liner	Cast iron	Bronze	Cast iron	Ni-resist or 12% chrome

[a]This table is to be used as a guide.

[b]The abbreviation above the diagonal line indicated cylinder material: the abbreviation below the diagonal line indicates fitting material. Abbreviations are as follows: *CI*, cast iron; *BRZ*, bronze; *STL*, steel; *HI ALLOY*, high-temperature alloy.

[c]Applies where applicable.

Table 4.14 Materials and material specifications for major component parts

Materials for direct-acting reciprocating pump gas end parts[a] **Material class and material class abbreviations**[b]				
API material class code Part	I-1 CI/CI	I-2 CI/BRZ	S-1 STL/CI	S-2 STL/HI ALLOY
Pressure-retaining parts				
Cylinder	Cast iron	High-temperature cast iron	Carbon steel	Carbon steel
Head, chest, cover	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Stuffing box	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Gland	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Head, chest, box and gland bolting	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel
Nonpressure-retaining parts				
Cylinder liner	None	None	Cast iron	High-temperature cast iron
Box bushing[c]	Cast iron	Cast iron	Cast iron	Cast iron
Piston	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Piston rod	12% chrome	12% chrome, hardened	12% chrome	12% chrome, hardened
Piston ring	Cast iron	Cast iron	Cast iron	Special cast iron
Main valve	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Valve rings[c]	Cast iron	Cast iron	Cast iron	Special cast iron
Auxiliary valve[c]	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Valve rod	12% chrome	12% chrome, hardened	12% chrome	12% chrome, hardened

[a]This table is to be used as a guide.
[b]The abbreviation above the diagonal line indicates cylinder material: the abbreviation below the diagonal line indicates fitting material. Abbreviations are as follows: *CI*, cast iron; *BRZ*, bronze; *STL*, steel; *HI ALLOY*, high-temperature alloy.
[c]Applies where applicable.

Table 4.15 Materials for reciprocating pump parts[a]

Material	Castings	Forgings	Bar stock	Bolts and studs
Cast iron	ASTM A 48 or A 278	–	–	–
Nodular iron	ASTM A 395 or A 536	–	–	–
High-temperature cast iron	ASTM A 278, Class 40 or 60, stress relief, annealed	–	–	–
Carbon steel	ASTM A 216, Grade WCA or WCB, or ASTM A 352	ASTM A 105 or ASTM A 576	ASTM A 108 or ASTM A 575	–
5% chrome steel	ASTM A 217, Grade C5	ASTM A 182, Grade F5	–	–
12% chrome steel	ASTM A 296, Grade CA6NM, or CA15	ASTM A 182, Grade F6	ASTM A 276, Type 410, or ASTM A 582, Type 416	ASTM A 193, Grade B6
18-8 stainless steel	ASTM A 296, Grade CF20	ASTM A 182, Grade F304	ASTM A 276, Type 304	ASTM A 193, Grade B8
316 stainless steel	ASTM A 296, Grade CF8M	ASTM A 182, Grade F316	ASTM A 276, Type 316	ASTM A 193, Grade B8M
AISI 4140 steel	–	–	ASTM A 322, Type 4140	ASTM A 193, Grade B7
Bronze	ASTM B 584	–	ASTM B 139	ASTM B 5124
Material	Typical description			
Ni-resist	Type 1, 2, or 3 as recommended by International Nickel Co. for service conditions			
Hardenable stainless steel	ARMCO 17.7 PH, ARMCO 17.4 PH, US Steel Stainless W, Allegheny Ludlum AM 350 and AM 355, nitronic 50 or 60			
Colmonoy	Sprayed or fused deposit of 5.08 mm (0.020 in) or gas weld deposit of 7.94 mm (1/32 in) minimum thickness of Wall-Colmonoy AWS, Class R, Ni-Cr-C Metso 16c			

[a]This table is to be used as a guide.

Table 4.16 Material classes for centrifugal pumps

Caution: This table is intended as a general guide. It should not be used without a knowledgeable review of the specific services involved.

Service	On-plot process plant	Off-plot transfer and loading	Temperature range (F)	Pressure range (psig)	Material class	Note reference number
Fresh water, condensate, cooling tower water	X	X	<212	All	1-1 or 1-2	
Boiling water and process water	X	X	<250	All	1-1 or 1-2	7
	X	X	250–350	All	S-5	7
	X	X	>350	All	D-6	7
Boiler feed water						
Cast horizontally split	X	X	>200	All	C-6	
Forged barrel	X	X	>200	All	S-6	
Boiler circulator	X	X	>200	All	C-6	
Foul water, reflux drum water, water draw, and hydrocarbons containing these waters, including reflux streams	X	X	<350	All	S-3	2
Propane, butane, liquefied petroleum gas, and ammonia (NH3); diesel oil; gasoline; naphtha; kerosene, gas oils; light, medium, and heavy lube oils; fuel oil residuum; crude oil; asphalt; synthetic crude buttons	X	X	<450	All	S-1	
	X	X	<450	All	S-1	
	X	X	450–700	All	S-6	2.5
	X	X	>700	All	C-6	2
Noncorrosive hydrocarbons, for example, catalytic reformate, desulfurized oils	X	X	450–700	All	S-4	5
Xylene, toluene, acetone, benzene, furfural, MEK	X	X	<450	All	S-1	

Sodium carbonate, doctor solution	X	X	<350	All	1-Jan	
Caustic (sodium hydroxide) concentration of 20% or less	X	X	<140	All	S-1	8
	X		140–200	All	S-3	8
	X		>200	All		6
MEA, DEA, TEA-stock solutions	X	X	<250	All	S-1	
DEA, TEA-lean solutions	X	X	<250	All	S-1	
MEA-lean solution (CO_2 only)	X	X	175–300	All	S-9	
MEA-lean solution (CO_2 and H_2S)	X	X	175–300	All		9
MEA, DEA, TEA-rich solutions	X	X	<175	All	S-1	
Sulfuric acid concentration						
85%	X	X	<100	All	S-1	2
85%–15%	X	X	<100	All	A-8	2
15%–1%	X	X	<100	All	A-8	2
1%	X	X	<450	All	A-8	2
Hydrofluoric acid concentration of over 96%	X	X	<100	All	S-9	2

Notes:

The materials for pump parts for each material class are given in API 618.

The corrosiveness of foul waters, hydrocarbons over 450°F, acids, and acid sludges may vary widely. A materials recommendation should be obtained for each service. The material class indicated before will be satisfactory for many of these services but must be verified.

Cast iron cases, where recommended for chemical services, are for nonhazardous locations only. Use steel cases (S-1 or 1-1) for pumps in services located near process plants or in any location where released vapor from a failure could create a hazardous situation or where pumps could be subjected to hydraulic shock, for example, in loading service.

Obtain separate materials recommendations for services not clearly identified by the service descriptions listed in this table.

If product corrosivity is low, Class S-4 materials may be used for services at 451 to 700°F. Obtain a separate materials recommendation in each instance.

Use Alloy 20 or Monel pump material and double mechanical seals with a pressurized seal oil system.

Consider oxygen content and buffering of water in the selection of material.

All welds should be stress relieved.

Use Class A-7 materials except for a carbon steel case.

Mechanical seal materials for streams containing chlorides, all springs, and other metal parts should be Alloy 20 or better. Buna-N and neoprene should not be used in any service containing aromatics. Viton should not be used in services containing aromatics above 200°F.

Table 4.17 Injection pumps (plunger)

	Environment			
	Aerated		Nonaerated	
	Without H2S	**With H2S**	**Without H2S**	**With H2S**
Fluid end	Ti 400M, K500, AlBr6ca,w, A205, In6255	Ti NiAl6w, 3167,8, 316L7,8, A205, In6255, K5005	Ti 400M, K500, AlBr6ca,w, A205, In6255	Ti AlBr6, NiAlw, 400M5, K5005, A205, In6255
Plunger	NM6 HardNi1,6	NM6 HardNi1,6	NM6 HardNi1,6	NM6 HardNi1,6
Valve	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-48, 15-7MO
Valve seat	400M, 17-4, NiAlca, 3163,7	400M, 17-4, NiAlca, 3163,7	400M, 17-4, NiAlca, 3163,7	400M, 17-48, NiAlca, 3163,7
Valve spring	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M
Stuffing box	AlBrca, 3164	AlBrca, 3164	AlBrca, 3164	AlBrca, 3164

Notes

1. For many services can be on carbon steel base. For extreme corrosion, should be austenitic stainless steel.
2. Ball valve.
3. For ball valves with hard insert.
4. Other materials can be furnished to match fluid ends.
5. Limited experience—unrated.
6. Widely used materials.
7. Subject to chloride cracking and subject to differential aeration.
8. Subject to stress cracking under some conditions.

Table 4.18 Material code for use with Table 4.17

Code	Description of material (ASTM designations unless otherwise noted)	Code	Description of material (ASTM designations unless otherwise noted)
Fe1	Carbon or low-alloy steel	AlBrw	Aluminum bronze, B150, alloy 614 (9C sic) (wrought)
Fec1	Carbon or low-alloy steel, coated or lined internally	NiAlca	Nickel aluminum bronze, B148, alloy 955 (9D) (cast)
Fe2c1	Carbon or low-alloy steel, coated in or out		
Cl1	Gray cast iron, A48, Class 30	NiAlw	Nickel aluminum bronze, B150, alloy 630 (9D sic) (wrought)
Clc1	Cast iron, coated internally	NiVBr	Ni-Vee3 bronze, Type D

Table 4.18 Continued

Code	Description of material (ASTM designations unless otherwise noted)	Code	Description of material (ASTM designations unless otherwise noted)
Cl2c1	Cast iron, coated in and out	A20	Alloy/20 (wrought) or A296, Grade CN-7M (cast)
D11	Ductile (nodular) iron, A536, various grades		
NiRei	Ni-resist iron, A436, Type 1	400M	Monel3 400 (wrought) or Monel3 410 (cast)
NiRei 2A	Ni-resist iron, A436, Type 2A	K500	Monel3 K-500 (wrought) or Monel3 505 (cast)
NiRe D2	Ductile Ni-resist3 iron, A439, Type D-2	Inall	Inconel3 metal alloy (all listed grades)
410	AISI 410 stainless steel (wrought) or A296, Grade CA-15 stainless steel (cast)	In600	Inconel3 metal alloy 600
303	AISI 303 stainless steel (wrought)	In625	Inconel3 metal alloy 625
304	AISI 304 stainless steel (wrought) annealed or A296, Grade CF-8 (cast) annealed	In718	Inconel3 metal alloy 718
		In750	Inconel3 metal alloy X-750
316	AISI 316 stainless steel (wrought) annealed or A296, Grade CF-8M (cast) annealed	C1	Carbon or graphite
		Ti1	Titanium or titanium alloys
316L	AISI 316L stainless steel (wrought) annealed or A296, Grade CF-3M (cast) annealed	WC1	Tungsten carbide
		HaC	Hastelloy4 "C," B334 and B336 (wrought) or A494 (cast)
17-4	17-4 PH stainless steel (wrought) or ACI Grade CB-7Cu (cast)	Hard Ni1	Ni-Cr-B-Si alloys (such as Colmonoy) 2,5
15-7Mo	15-7 Mo PH stainless steel	Hard Co1	Cobalt-base alloys (such as stellite) 2,4
BrgBr	High-leaded tin bronze, B144, alloy 943 (70-5-25)	Cr/1	Chromium plating
63 Br	Zinc-less bronze, SAE 63 (Copper Alloy 927)	Ni/1	Nickel plating
AlBrca	Aluminum bronze, B148, alloy 954 (9c) salt water anneal (cast)	NM1	Nonmetallic

Notes

1. Material group having various analysis—not listed in Table 4.17.
2. Various forms—sprayed and fused coatings, weld overlays, cast, wrought.
3. Reg. T.M. The International Nickel Company.
4. Reg. T.M. Cabot Corporation.
5. Reg. T.M. Wall-Colmonoy Corporation.

4.7 Installation considerations

4.7.1 Overview

If positive displacement pumps are properly installed and operated, satisfactory performance can be realized for a long time. These pumps are manufactured in a variety of designs for many different services. Each manufacturer's instructions should be carefully followed for specific machine or application requirement. The following discussion relates to general installation guidelines for positive displacement reciprocating pumps.

4.7.2 Foundations and alignment

4.7.2.1 Background

Most pump foundations are constructed of reinforced concrete. Pump and driver are bolted to a cast iron or steel baseplate that is secured to the concrete foundation via anchor bolts. Small pumps need a foundation large enough to accommodate the baseplate assembly. On the other hand, large pumps require a foundation that is 3 to 4 times the weight of the pump and driver.

4.7.2.2 Anchor bolt-sleeve installation

Each bolt is fitted with a washer and passed through a pipe sleeve that has a diameter 3 to 4 times greater than the bolt. The bolt-sleeve unit is set into the concrete at the predetermined base hole positions. The flexibility in the sleeve-washer unit allows minor adjustments to be made in the bolt position prior to final tightening, even after the concrete foundation has set.

4.7.2.3 Metal shim adjustments

Metal shims are used to position the pump on the foundation. Adjustments are made until the pump shaft and port flanges are completely leveled. Alignment between the pump and driver is then adjusted before connecting the pump to the suction and discharge lines. The latter should have previously been aligned during the initial positioning of the baseplate.

4.7.2.4 Grouting

Once the piping has been securely bolted, the entire pump assembly should be rechecked for flexure due to pipe strain. If the drive train alignment has not been changed by bolting the piping, the space between the baseplate and concrete foundations is filled with grouting. Grouting should be sufficiently fluid so as to fill all the available space under the baseplate.

4.7.2.5 Operating temperature considerations

It is essential the alignment between the piping, pump, and driver do not change. Ideally, alignments should be made at the operating temperature of the pumping system thus eliminating any alignment changes due to thermal expansion.

4.7.3 Process piping considerations

4.7.3.1 General piping considerations

Next to the selection of operating speeds, proper piping design is the most important consideration in pump installation design. Poor piping is often the result of inattention to details which can lead to

- more than average down time
- higher maintenance costs
- loss of operating personnel confidence

Fig. 4.50 illustrates some of the following *recommended installation* guidelines for reciprocating pumps.

4.7.3.2 Suction piping

It is critical that the suction piping be sized to assure the $NPSH_A$ exceeds the $NPSH_R$ by the pump. Often, this can be arranged by elevating the suction tank or by providing a low-head centrifugal charge pump to feed the reciprocating pump. If the $NPSH_A$ is too low, valve breakage and pump maintenance costs will be excessive.

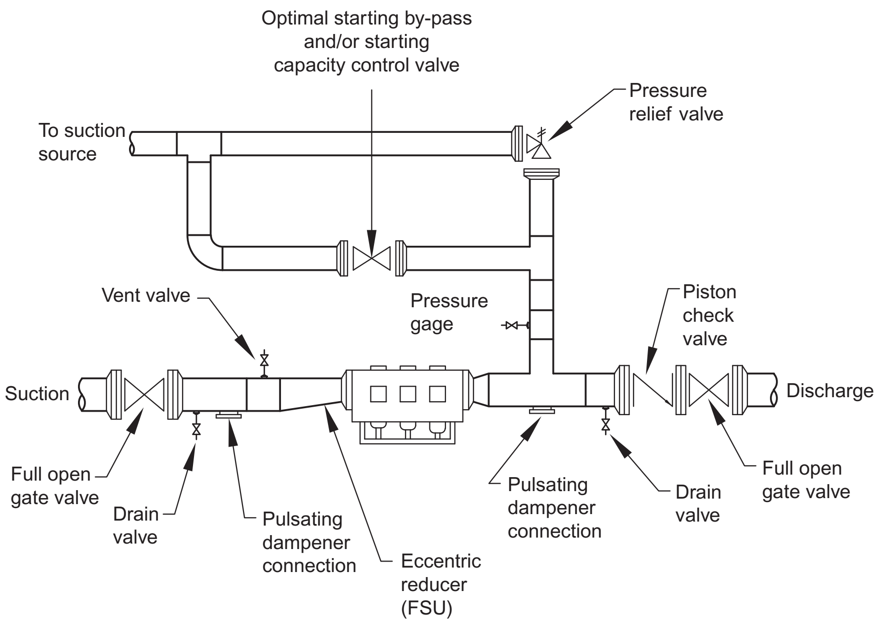


Fig. 4.50 Typical reciprocating pump installation.

The pump inlet size has no bearing on the required suction piping size. The available suction head and the suction requirement and losses must be considered and piping sized on this basis, regardless of the pump size.

Both the suction and discharge piping should be

- short
- direct
- free of bends, if possible
- minimum number of elbows and fittings
- avoid piping high points where vapors can become trapped
- elevation and plan changes should be laid out using 45° ells rather than 90° ells (if 90° ells are used, they should be long radius type)
- suction pipe diameter changes should be made with eccentric reducers, with flat side up (FSU) to eliminate gas pockets
- at least one nominal pipe size larger than the pump suction
- to allow pump isolation, a full opening block valve should be installed in the suction piping

The suction piping should include a strainer and a pulsation dampener, if required. The suction strainer should not be installed unless regular maintenance can be assured. A fluid starved condition, resulting from a plugged strainer, can cause more damage to the pump than solids ingestion.

The suction supply vessel outlet should be slightly higher than the pump inlet so that gases accumulating in the system may flow back to the vessel rather than through the pump. The supply vessel should have sufficient retention time for the evolution of "free" gas. The suction and bypass lines should enter the supply vessel below the minimum fluid level. A vortex breaker should be installed on the outlet of the supply vessel.

Table 4.19 lists some suggested maximum flow velocities for sizing suction and discharge piping for reciprocating pumps. The piping should be large enough so that the velocity limits are not exceeded. A low flow velocity for the suction piping is particularly important. Some companies use a maximum velocity of 1 ft/s (0.3 m/s) regardless of pump speed.

When two or more pumps are installed in parallel, each pump's suction line between the tank and the pump should be piped separately (rather than in common) to preclude mutually reinforced pulsations. In most cases, however, this procedure is not practical, and the suction lines are often manifold together. If manifold together, the lines should be sized so that the velocity in the common feed line is approximately equal to the velocities in the lateral lines feeding the individual pumps. This avoids abrupt velocity changes and minimizes acceleration head effects. (The acceleration head requirement for multiple pumps on a common suction line is not the sum of

Table 4.19 Maximum suction and discharge pipe velocities for reciprocating pumps

Pump speed, rpm	Suction velocity, ft/s (m/s)	Discharge velocity, ft/s (m/s)
<250	2 (0.6)	6(1.8)
250–330	1.5 (0.46)	4.5 (1.37)
>330	1 (0.3)	3 (0.9)

single pump requirements; it increases approximately by the square of the number of pumps. For example, 3 pumps operating on a common suction line require approximately nine times the acceleration head (H_A) of a single pump.)

4.7.3.3 Discharge piping

Just as fluid flows to a reciprocating pump in a pulsating flow pattern, it is discharged in the same manner. It has been shown that these pressure surges travel through the fluid in a straight line and are reflected back toward the source by restrictions or bends in the system. Also, when two or more pumps are discharging into a common header, these pressure surges may be amplified, causing damage to the pumps and piping. When designing a discharge piping system for reciprocating pumps:

- Avoid sharp bends, reducers, valves with less than full opening, and so on, near the pump; these may reflect pressure surges back toward the pump.
- Manifolds in which two or more pumps are tied into a common system should be located as far from the pumps as practical to allow dampening of surges. For most applications, 100 to 150 ft. (30 to 45 m) is adequate.
- Discharge piping should be securely anchored as near the pump as practical to prevent system vibrations from acting directly on the pump.
- Concentric reducers may be used, but they should be placed as near to the pump as practical.
- Pressure safety valves (PSVs) should be installed in the discharge piping near the pump and certainly upstream of the first block valve. Discharge/outlet piping should have a high enough pressure rating for potential future needs. Consideration must also be given to discharge PSVs flange ratings. PSVs must be set high enough to avoid inadvertent discharges.
- Directional piping changes should be made with 90° long radius ells.
- Pipe diameters should be based on the maximum velocities recommended in Table 4.19. Common practice is to size discharge pipe one nominal pipe size larger than the pump discharge connection.
- To facilitate priming and starting the following should be installed: Recycle (bypass) piped back to suction vessel, check valve, and block valve.
- If a pulsation dampener is not included in the initial installation, a flanged connection should be provided should pulsation attenuation be required in the future.

4.7.3.4 Piping hook-up considerations

Fig. 4.51 shows an example hook-up for two reciprocating pumps operating in parallel. Since the pump can be accidently started when the discharge block valve is closed, a PSV is installed in the discharge line to keep the pump from overpressuring the pipe and flanges. The PSV should be installed downstream of the pulsation dampener. It is also possible to leave the suction valve closed while the discharge valve is opened. Discharge fluid could leak through the discharge FSV and pump valves, pressuring up the suction piping, which is rated for ASME 150 Class. Thus, in this installation, a PSV was installed in the pump suction piping. Some companies believe a suction PSV is unnecessary because of the low probability of multiple failures.

An appendage dampener and cone strainer are installed in the suction line. An inline bladder/desurger and FSV are installed in the discharge line. The FSV protects against leakage from discharge when the pump is not running. It is preferable that this

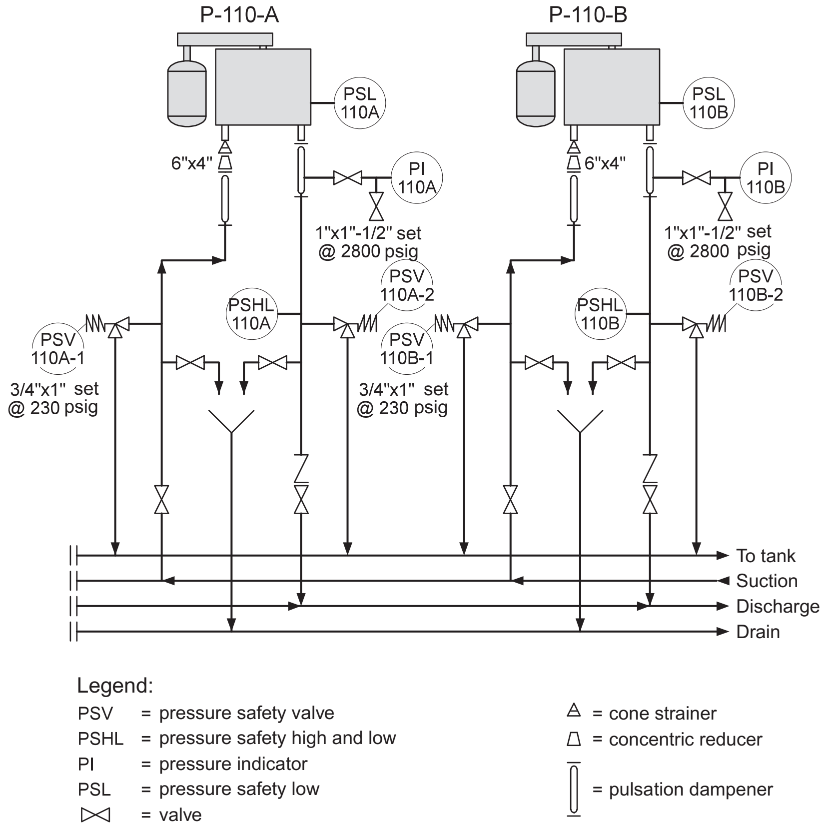


Fig. 4.51 Mechanical flow diagram of two reciprocating pumps in parallel.

be a piston FSV to keep it from chattering due to pressure pulsations. Nevertheless, swing FSVs are used successfully in some installations that follow the design practices for minimizing pulsations. Drain valves are provided so that the pump can be maintained easily, and an oil PSL is provided to shut-in the motor.

API RP 14C requires a PSH be installed on the discharge so that the pump will shutdown before the discharge PSV opens. It also requires that a PSL sensor on the discharge be installed to shutdown the pump in case of a large leak in the discharge piping. These two functions are carried out by one device.

4.7.4 Pulsation considerations

4.7.4.1 General considerations

Flow from a reciprocating pump is not uniform but pulsating. The oscillating motion of the plungers creates disturbances (pulsations) that travel at the speed of sound from pump cylinder to the piping system. Pulsations are a function of the pump piston/plunger velocity and the pump's internal valves, which generate liquid pulsations at integral multiples of the pump operating speed. The pressure pulsation can also excite both mechanical resonances and piping acoustic liquid resonances. Figs. 4.52 and 4.53 show the suction and discharge pressure variations versus time for a water injection station with 5 quintuplex pumps operating in parallel.

4.7.4.2 Adverse effects

The possible adverse effects of reciprocating pump pressure pulsations are as follows:

- Suction pulsations can cause the pressure level to drop instantaneously below fluid vapor pressures, resulting in cavitation. Cavitation can contribute to failure of pump parts such as valves, crossheads, rods, and so on. Severe cavitation can cause high piping vibrations and failures of vents, drains, and gauge lines.
- Pulsation forces can be so high that normal pipe clamps and supports may be ineffective in controlling the vibrations.
- Pulsations generated by the flow modulation from the pump plungers can be amplified by the acoustical resonances of the piping system, which can result in pump fluid end failures and piping failures due to the shaking caused by pressure pulsations.

4.7.4.3 Pulsation dampeners

In many installations when the piping system is sized in accordance with the recommendations in the process piping discussed in Section 4.7.3, there may be no need to

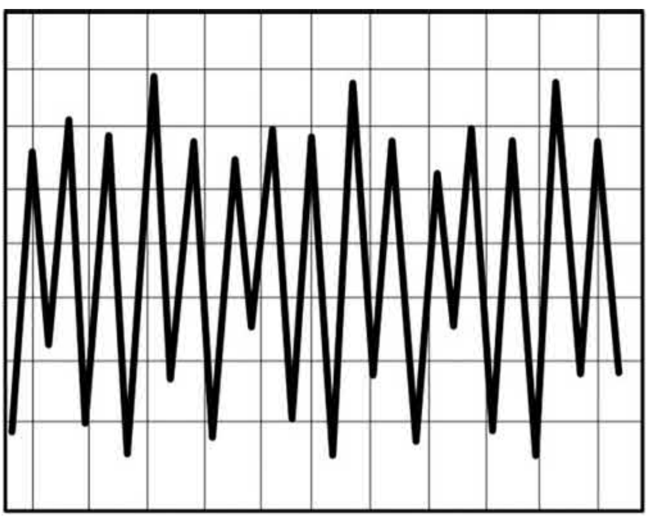


Fig. 4.52 Suction pressure variation versus time for a water injection station with 5 quintuplex pumps operating in parallel (mean suction pressure 516 psig).

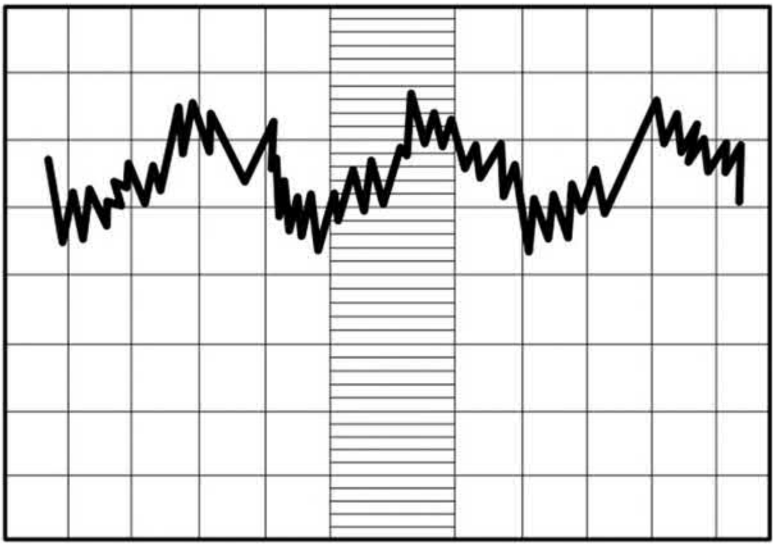


Fig. 4.53 Discharge pressure variation versus time for a water injection station with 5 quintuplex pumps operating in parallel (mean discharge pressure = 1450 psig).

install dampeners to reduce liquid pulsations. However, in most instances, it is possible to reduce pulsations further and thus reduce pump maintenance costs and piping vibration through the use of pulsation dampeners. Dampeners are recommended for all major multipump installations, unless computer analog studies indicate that they are not needed. Often it is cheaper to install the dampeners than to perform the detailed engineering studies to prove that they are not needed. For simple piping layouts and low to moderate pump speeds, pulsation dampeners are used to attenuate the effects of pulsating flows. Fig. 4.54 shows the discharge pressure variations "before" and "after" the installation of a pulsation dampener. Fig. 4.55 shows the suction and discharge pressure variations "before" and "after" the installation of a pulsation dampener.

Pulsation dampeners are normally installed on both the suction and discharge of the pump. Dampeners can be

- Liquid filled (Fig. 4.56)
- Gas cushioned (Fig. 4.57)
- Tuned-acoustical filter (Figs. 4.58 and 4.59)

4.7.4.3.1 Liquid-filled dampeners

A liquid-filled dampener is a large surge vessel located close to the pump. It uses the compressibility of the liquid to absorb pressure pulsations, and thus it works better on gaseous liquids (hydrocarbons, rich glycol) than on relatively gas-free liquids (water). The volume of the vessel is normally recommended to be ten times the pump

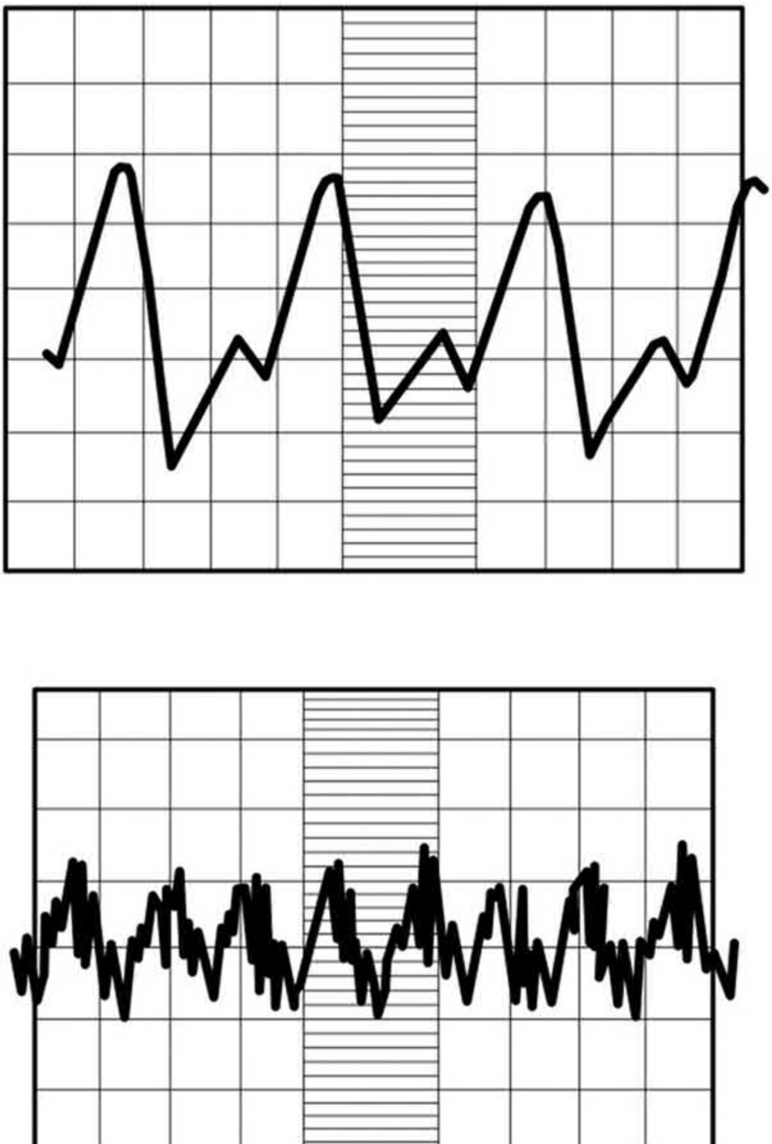


Fig. 4.54 Discharge pressure variations. Top: before installation of a pulsation dampener (pressure fluctuation = 48 psi) and Bottom: after the installation of a pulsation dampener (pressure fluctuation = 12 psi).

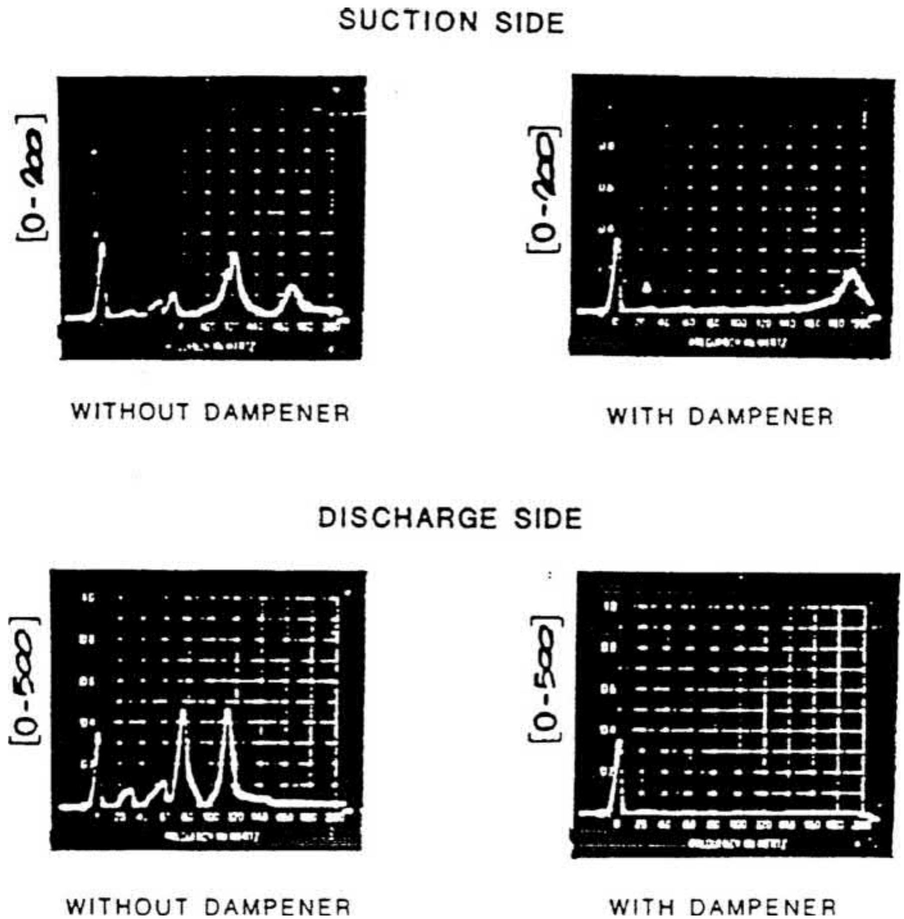


Fig. 4.55 Analog study showing the suction and discharge pressure variations "before" and "after" the installation of a pulsation dampener.

displacement (ft^3/min). Thus, for single-acting pumps, volume can be calculated from the following:

Field units

$$\text{Vol} = 1.337q \tag{4.19a}$$

SI units

$$\text{Vol} = 0.167q \tag{4.19b}$$

where

Vol = volume of surge tank, ft^2 (m^3)

Q = flow rate, gpm (m^3/h)

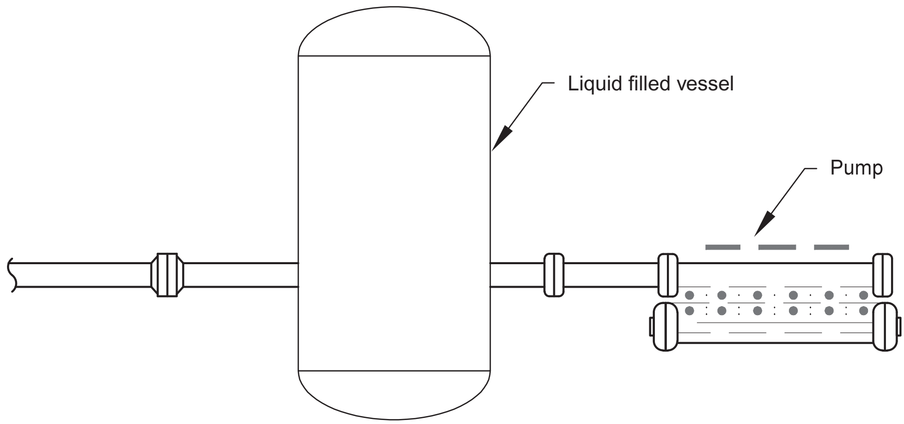


Fig. 4.56 Schematic diagram of liquid-filled dampener.

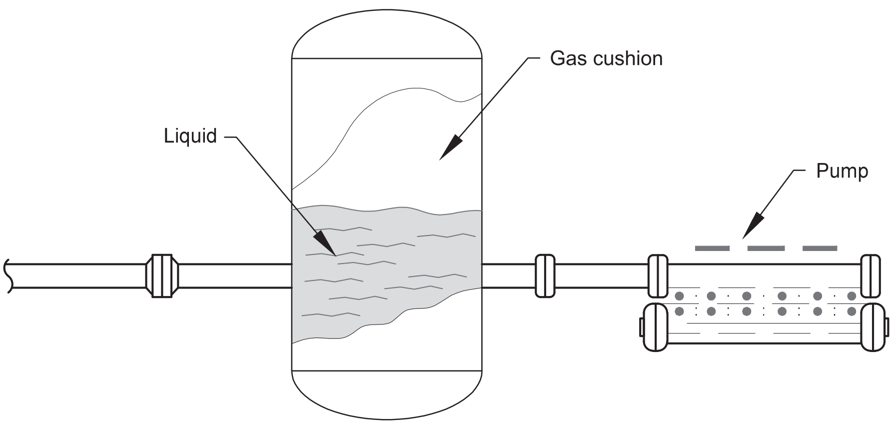


Fig. 4.57 Schematic diagram of gas-cushioned dampener.

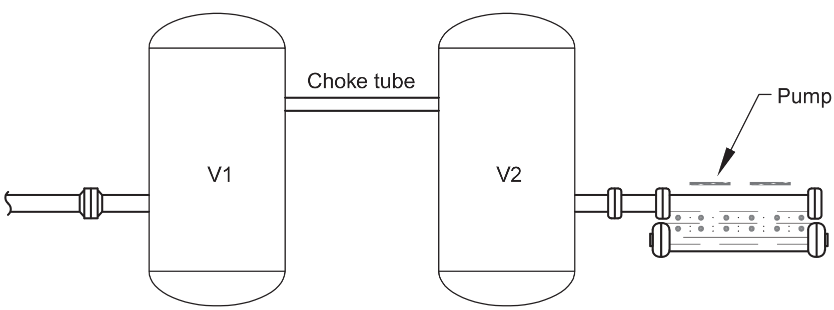


Fig. 4.58 Schematic diagram of "Type 1" tuned-acoustical filter.

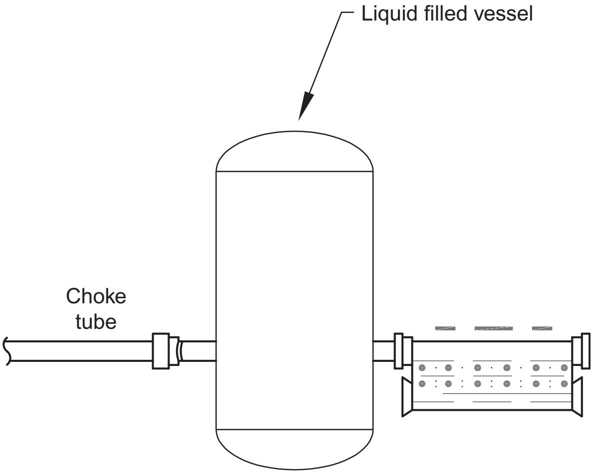


Fig. 4.59 Schematic diagram of "Type 2" tuned-acoustical filter.

Advantages

- Maintenance free
- Insignificant pressure drop

Disadvantages

- Takes up a large amount of space
- Heavy when filled with water
- Expensive because they are rated to the MAWP of the piping system

Fig. 4.60 shows an installation for a quintuplex pump with liquid surge section and discharge dampeners.

4.7.4.3.2 Gas-cushioned dampeners

Fig. 4.61 shows typical gas-cushioned dampeners. The simplest type is a vertical surge bottle partially filled with pumped fluid and partially filled with gas. The highly compressible gas over time absorbs the pressure pulses. The gas in the vapor space is normally natural gas, which over a period of time can dissolve in the liquid. Thus a level gauge is installed so that the interface position can be observed and either more gas added or excess gas vented to maintain the required gas volume. Air should be avoided because it will increase the likelihood of corrosion, scale, and bacteria in water and increases the potential for an explosion in hydrocarbon service.

The required gas volume is given by

Field units

$$(\mathrm{Vol})_g = \frac{KSd^2P}{748(\Delta P)} \tag{4.20a}$$

Fig. 4.60 Example of a quintuplex pump installation showing a liquid surge suction and discharge dampener.

SI units

$$(\text{Vol})_g = \frac{KSd^2P}{4.328 \times 10^8 (\Delta P)} \tag{4.20b}$$

where

$(\text{Vol})_g$ = required gas volume, ft^3 (m^3)
K = pump constant (from Table 4.20)
S = pump stroke, in. (mm)
d = pump piston or plunger diameter, in. (mm)
P = average fluid pressure, psi (kPa)
ΔP = allowable pressure fluctuation, psi (kPa)
= Fig. 4.62

The allowable pressure pulsation amplitude is somewhat arbitrary. Fig. 4.62 can be used as an estimate if other information is not readily available.

Gas-cushioned dampeners are much smaller than liquid-filled dampeners but require monitoring of the interface. They are impractical in locations where the discharge pressure varies widely and the gas volume expands and contracts in response.

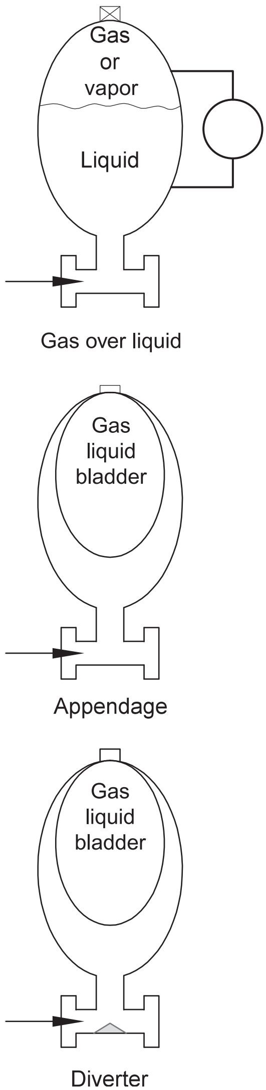


Fig. 4.61 Schematic diagram of typical gas-cushioned pulsation dampeners.

They require a vertical vessel otherwise gas will escape chamber. Thus they require a system for recharging.

Gas-cushioned dampeners can employ a pressured bladder to keep the gas from being absorbed in the liquid. They are small, inexpensive, and can be mounted in a horizontal vessel. They can have a configuration such as that shown in Fig. 4.61, or the bladder can be in the shape of the cylinder, as in the in-line bladder of

Table 4.20 Pump constant "K" for gas cushion design

Pump type	Action	K
Simplex	Single	0.67
Simplex	Double	0.55
Duplex	Single	0.55
Duplex	Double	0.196
Triplex	Single	0.098
Triplex	Double	0.196
Quintuplex	Single	0.360
Quintuplex	Double	0.060
Setuplex	Single	0.017
Setuplex	Double	0.034

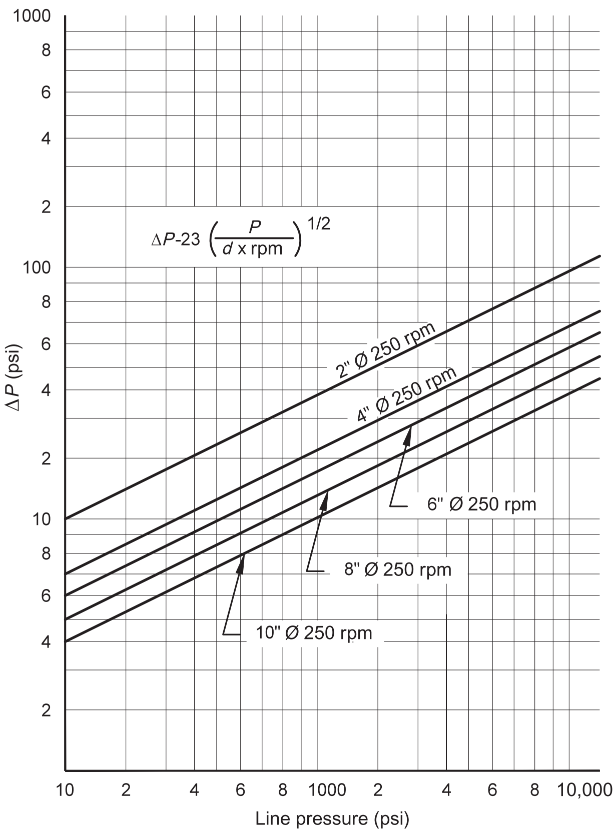


Fig. 4.62 Allowable pressure fluctuation versus frequency.

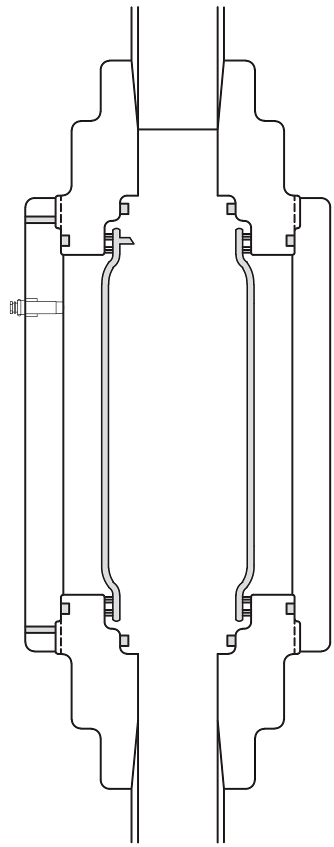


Fig. 4.63 Schematic diagram of an in-line "desurger" or dampener.

Fig. 4.63. The use of diverters in appendage-type dampeners or in-line configurations aids in attenuating high-frequency pulsations. The size of the gas volume depends upon the bladder properties and configuration of the design. For approximating purposes, Eqs. (4.18a, 4.18b) can be rewritten as follows:

Field units

$$(\mathrm{Vol})_{\mathrm{g}} = \frac{KAd^2P^2}{748(\Delta P)(P_c)} \tag{4.21a}$$

SI units

$$(\mathrm{Vol})_{\mathrm{g}} = \frac{KSd^2P^2}{4328 \times 10^8(\Delta P)(\Delta P_c)} \tag{4.21b}$$

where:

P_c = bladder precharge pressure, psi (kPa)

The bladder precharge pressure is normally set at 60% to 70% of average fluid pressure. Bladder-type dampeners are normally an economical solution and are in common use.

Bladder-type dampeners have the following limitations:

- Bladder is limited to applications below 300°F (149°C).
- Impractical at locations where pressures vary widely.
- Elastomer eventually will wear out and will need to be replaced.

4.7.4.3.3 Tuned-acoustical dampeners

Tuned-acoustical dampeners are formed when two liquid-filled vessels are connected by a short section of small diameter pipe called a choke tube. This system can be designed to have a specific resonant frequency. Pressure pulsations at frequencies above this level are attenuated considerably.

Type 1 is formed when two liquid-filled dampeners are connected by a short section of small diameter pipe called a "choke tube." System is designed, using a straightforward equation, to have a specific resonant frequency. Pressure pulsations at frequencies above this level are attenuated considerably.

Type 2 is formed by connecting the pump to a liquid-filled dampener. The choke tube is then installed between the dampener and the piping system. Acoustical filters are best done on computer analogs.

Acoustical dampeners have the following advantages:

- Can be used in high-temperature situations.
- Are essentially maintenance free.
- Temperature is not a limiting factor.
- Low maintenance costs.
- Filter can be tuned to suppress all frequencies low and high, that the pump will generate.

Acoustical dampeners have the following disadvantages:

- High relative cost.
- Requires a lot of space.
- Choke tube requires a high-pressure drop, which is a limitation in low NPSH applications (not used on suction lines in which NPSH may be a problem).

Fig. 4.64 shows a pump installation with an appendage bladder dampener on the suction and a tuned-acoustical filter on the discharge. The pressure vessel contains two sections, one above the other, with a choke tube connecting them internally. An extremely efficient dampener can be made by connecting two gas-cushioned dampeners in series with a short run of pipe that acts as a choke tube. The gas-cushioned dampeners attenuate low-frequency pulsations, and the choke tube arrangement alleviates high-frequency pulsations. Most installations do not require this complexity. The design of acoustical filters is best done on computer analogs and is beyond the scope of this text.

4.7.4.3.4 Performance curve comparison

Fig. 4.65 shows the gain factors for a tuned Helmholtz filter and a liquid-filled surge vessel under the same conditions. The filter gain factor is determined from Eq. (4.22):

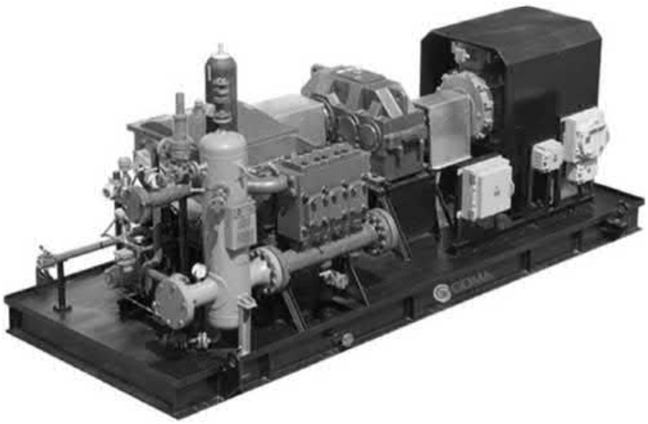


Fig. 4.64 Photograph of a pump with an appendage bladder dampener on the suction and a tuned-acoustical filter on the discharge.

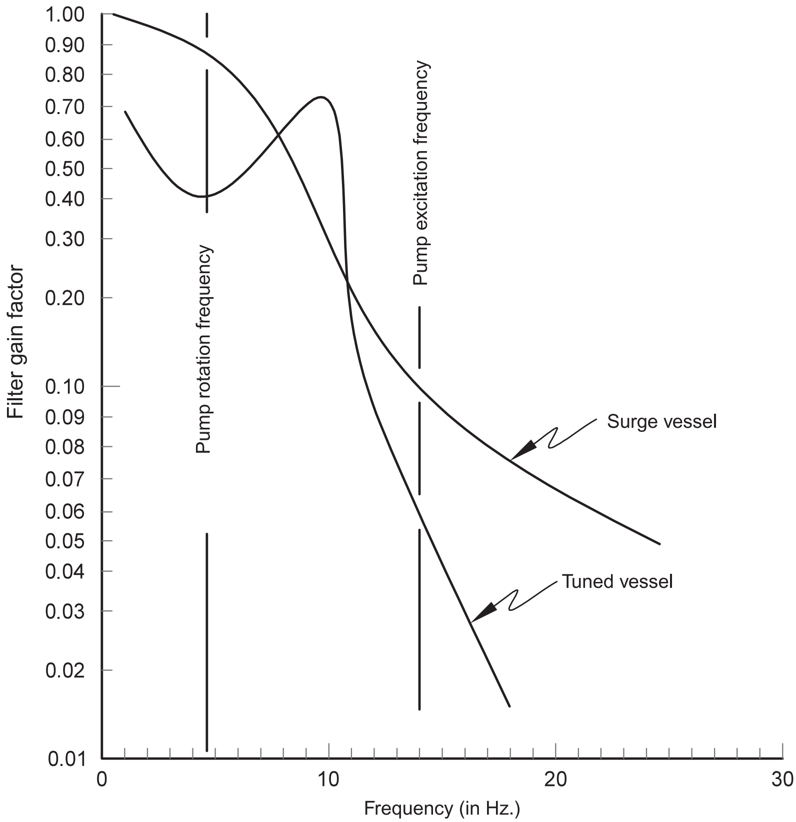


Fig. 4.65 Pump excitation frequency versus pulsation level.

$$(\text{Filter gain factor}) = \frac{(\text{Pressure pulse e filter inlet})}{(\text{Pressure pulse e filter outlet})} \tag{4.22}$$

Fig. 4.65 shows that at a pump excitation frequency (x-axis) of 14 Hz, attenuation through the tuned filter reduces the pulsation level at the filter discharge to 5% of the original discharge level. The surge vessel reduces the pulsation level to 10% of the original discharge level.

4.7.5 Pipe vibrations

To minimize pipe vibrations it is necessary to design pipe runs so that the "acoustic length" of the pipe run does not create a standing wave that adds to the pressure pulsations in the system. The acoustic length is the total overall length from end point to end point, including all elbows, bends, and straight pipe runs. Typical pipe runs with respect to acoustic length are considered to be

- Pipe length from suction tanks to the pump suction.
- Long pipe sections between pump and pulsation dampener.
- Pipe section between pump and manifold.

Piping should be designed so that piping runs with

- "*similar*" ends should not equal 0.5λ, λ, 1.5$\lambda\lambda$, 2λ... where λ is the acoustic wavelength
- "*dissimilar*" ends should not equal 0.25λ, 0.75λ, 1.25λ, 1.75λ... .

The ends of pipe runs are considered similar if both are either open or closed from an acoustic standpoint. They are dissimilar if one is open and one is closed. Examples of end classifications are as follows:

- If pipe size is dramatically reduced, it tends to act as a closed end.
- Orifice plates act as closed ends.
- Short-length flow nozzles act as closed ends.
- Abrupt pipe diameter enlargements act as open ends.
- Pipe tees presenting an increase in flow area (such as a tee with three equal legs) act as open ends.
- Pipe size changes that could occur smoothly over a pipe length corresponding to several pipe diameters do not act as a termination.

The acoustic wavelength (λ) is determined from the following:
Field units

$$\lambda = \frac{720\left(\frac{gB_L}{P}\right)^{\frac{1}{2}}}{(N)(n)}$$

$$\lambda = \frac{720\left(\frac{gB_L}{\rho}\right)^{\frac{1}{2}}}{(N)(m)} \tag{4.23a}$$

SI units

$$\lambda = \frac{10\left(\frac{gB_L}{\rho}\right)^{\frac{1}{2}}}{N(m)} \tag{4.23b}$$

where

B_L = bulk modules of the fluid, psi (kPa)
 = Fig. 4.66
g = gravitational constant, 32.2 ft/s^2 (9.81 m/s^2)
ρ = fluid density, Ib/ft^2 (kg/m^3)
N = pump speed, rpm (rps)
m = number of plungers

It is also desirable to assume that the natural frequency of all pipe spans is higher than the calculated pump pulsation frequency to minimize mechanical pipe vibrations. The pump pulsation frequency is given by the following:
Field units

$$f_P = \frac{Nm}{60} \tag{4.24a}$$

SI units

$$f_P = Nm \tag{4.24b}$$

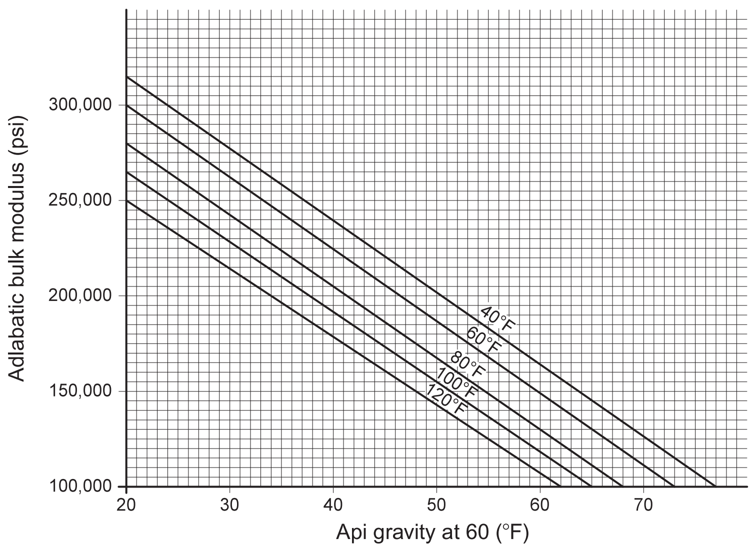


Fig. 4.66 Adiabatic bulk modules of crude oil versus API gravity.
Courtesy of API.

where

f_p = pump pulsation frequency, cycles/s
N = pump speed, rpm (rps)

The natural frequency of pipe spans can be estimated from Fig. 4.67. Normally, mechanical pipe vibrations will not be a problem if:

- Pipe support spacing is kept short.
- Pipe is securely tied down.
- Support spans are not uniform in length.
- Fluid pulsations have been adequately dampened.

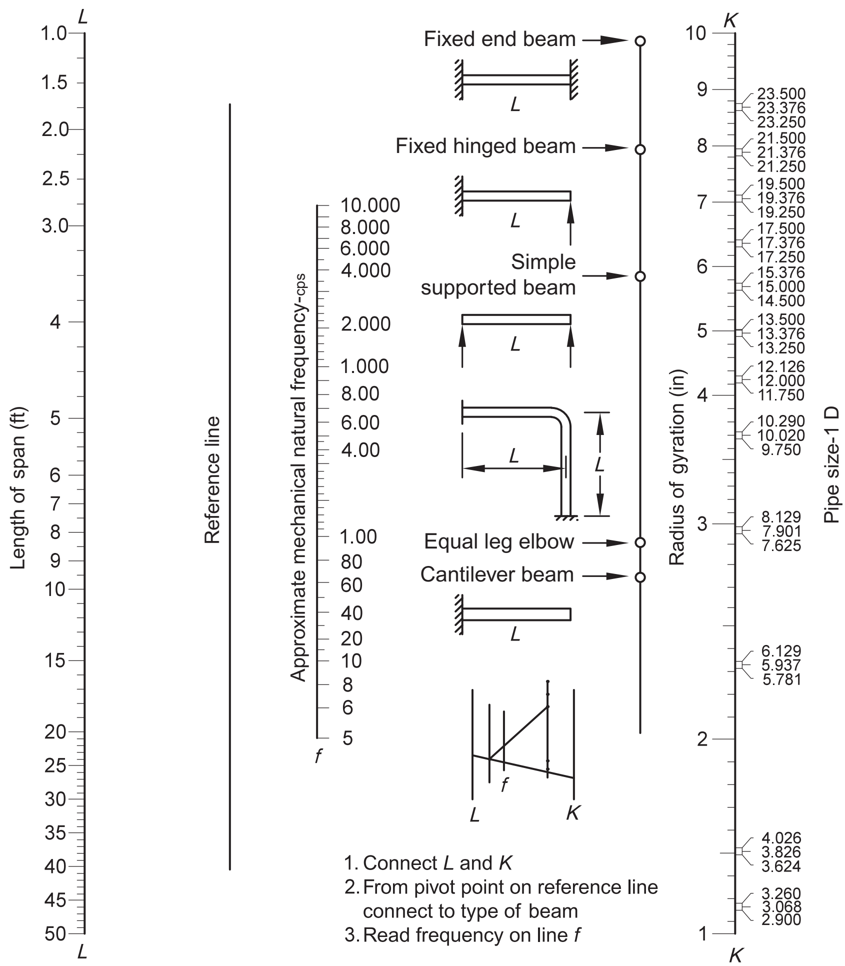


Fig. 4.67 Approximate mechanical natural frequency of pipe spans.

In summary, each pumping system must be individually evaluated to determine the specific dampening requirements. The slower a pump operates and the more cylinders it has, the chances of operation without dampeners or only with small dampeners are good. Double-acting piston pumps will generally result in smaller dampeners than single piston and plunger pumps.

4.7.6 Discharge PSV considerations

The purpose of the PSV on the pump discharge is to

- Protect the pump from excessive pressures.
- Protect the piping from excessive pressures.
- Protect the driver from excessive torque and horsepower.

The causes of excess pressure are clogged injection wellbore or accidental closure/failure of the pump discharge block valve. The PSV must be located between the pump and the first discharge block valve. The PSV should be as close to the pump as possible. The discharge of the PSV should be piped back to the source tank which avoids overheating the pumped fluid due to continued recirculation. The PSV and piping may have to be heat traced in high viscosity and in low-temperature services.

The PSV should be designed for intermittent safety use only and not be used to control pressure or flow. Most companies recommend that they be located in the piping even though some pumps have built-in relief valves. Recirculation flow with a built-in valve goes directly back to the suction side of the pump.

4.8 Reciprocating pump selection criteria

When selecting a reciprocating pump for oilfield applications it is necessary to determine if the proposed service calls for features required by API Standard 674 or if pumps built to manufacturer standards and modified for proposed oilfield usage are satisfactory.

There is normally little problem in choosing between the two basic types of pumps, direct-acting gas-driven pumps and crank-driven power pumps. Gas-driven pumps, once the workhorse of the industry, are generally limited to utility functions by the availability of compressed gas such as steam, air, or field gas. Power pumps, which are motor, turbine, or engine driven, are available in a wide spectrum of capacities and heads.

4.8.1 General design features

Items to consider when selecting a reciprocating pump are as follows:

1. The pump should be sized so that it is capable of delivering the required flow at the required discharge pressure at 85% of its maximum continuous rpm.
2. The pump fluid end should be a steel forging. The maximum allowable working pressure of the fluid end should exceed the design discharge pressure by 20%.
3. Pump valves should be removable and replaceable without disturbing the suction and discharge piping. Valve assemblies should be caged.

4. Pumps with a pressure lubrication system should be provided with a low oil pressure shutdown switch.
5. For pumped fluids that do not have lubrication qualities (e.g., seawater, fresh water), plungers should be lubricated. All lubrication tubing should be 316 SS.
6. All rolling contact bearings within the pump should be designed for a 3-year life.
7. Lube oil day tank should be provided for pump lube oil and plunger lube oil. Tanks should be sized for 30 days continuous operation. An automatic level instrument should be provided to maintain constant oil level. Day tanks should be equipped with level gauges.
8. Pump, driver, and all other appurtenances should be skid mounted. Piping connections should terminate at skid edge.
9. Skid material should be ASTM A36. Skid should be provided with a minimum of 4 leveling screws.
10. Unless specified otherwise, drain pans will not be provided by manufacturers. Where appropriate they should be specified.
11. Distance piece cover should be provided.
12. Piping connections of 1-1/4, 2-1/2, 3-1/2, 5, and 7 in. (31.75, 63.6, 88.9, 127, 177.8 mm) should be used.
13. Surfaces of rods or plungers in contact with packing shall be hardened or coated with hardened material.
14. All electrical connections should terminate at skid edge in a weatherproof, explosion proof, junction box.

4.8.1.1 Preparation for shipment

All openings should be plugged. Flanged openings should be covered with plywood or plastic protective covers. Instruments that can be damaged during shipment should be removed, boxed, and shipped separately.

4.8.1.2 Painting

Painting should be done in accordance with customer standards.

4.8.1.3 Documentation

Manufacturer should provide one (1) set of reproducible, good quality approval drawings that show skid outline and location of all customer connections. Drawing should be provided no later than two (2) weeks after receipt of order. Manufacturer should revise drawing to incorporate any comments. Operating and maintenance manuals should be provided.

4.8.1.4 Driver options

A designer should have some idea of the type of driver he desires and the type of power transmission device to be used to transfer power to the pump.

4.8.1.4.1 Choices of driver

- *Electric motor*: Usually <200 hp (149 kW), except for special situations.
- *Engine*: Normal for most oilfield applications over 200 hp (149 kW), but sometimes used down to 50 hp (37 kW).

- *Air/gas/hydraulic motor*: These usually require some transmissions, torque converters, or flexible couplings are available.

4.8.1.4.2 Choice of transmission

- Direct coupled: A number of different types of gear transmissions, torque converters, or flexible couplings are available.
- Belt driven: One of the most common types of power transmission in the oilfield, allows increased machinery life under load fluctuations.

4.8.2 Selecting a reciprocating pump

The following steps may be used to select a reciprocating pump.

1. Determine process duty.
2. Calculate liquid properties, if necessary.
3. Determine pipe pressure losses.
4. Calculate the suction head (same as for centrifugal pump).
5. Calculate the discharge head (same as for centrifugal pump)
6. Calculate the total head (same as for centrifugal pump)
7. Convert total head to pressure rise.
8. Calculate the $NPSH_A$. The procedure for calculating the $NPSH_A$ for a reciprocating pump is similar to that for a centrifugal pump except that acceleration head is included. Acceleration head is the force required to accelerate the fluid in the suction line.
9. Calculate brake horsepower.
10. Select particular pump. Using the pump manufacturers' literature and catalogs, select the pump for the conditions obtained in the calculation. If possible, avoid selecting the largest piston or plunger size for the pump case. Also avoid pumps which would have to operate continuously at maximum allowable speed.
11. Consult pump vendor. Discuss pump selection with the vendor for further recommendations and as a check of the selection procedure.
12. Prepare pump data sheet and specification.

Table 4.21 summarizes the advantages and disadvantages or positive displacement reciprocating pumps.

4.9 Reciprocating pump types

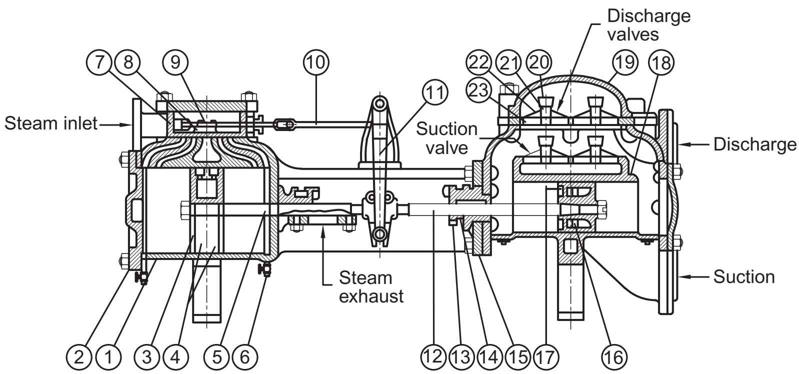
4.9.1 Reciprocating, piston, duplex, direct-acting gas-driven (process gas, air, steam) (Fig. 4.68)

Typical service	Relief drum pump-out Low-pressure boiler feed Water Sludge Sump pump Transfer

Pressure range	0 to 700 psig (0 to 4826 kPa)
Capacity range	0 to 500 gpm
Speed range	30 to 60 rpm (with piston speeds usually between 50 and 100 rpm)
Max allowable temperature	350°F (177°C)
Standard materials	Normal duplex, double-acting, simplex available. Normally cast iron and liquid ends with steel or bronze rods and trim
Specification	API 674
Typical control method	Speed control by throttling drive gas (steam, air, process gas), usually manual
Advantages	Self-priming Will operate at very low speeds High efficiency Minimizes liquid emulsification Handles viscous fluids No electrical power is required Suitable for unattended remote installation
Limitations	Pump speed is affected by system pressure Subject to vapor lock with low $NPSH_A$ Will stall with too-high system backpressure Pulsating flow can affect sensitive instrumentation downstream

Table 4.21 Summary of advantages and disadvantages of reciprocating pumps

	Advantages	Disadvantages	Use
Piston	Accurate flow rate Adjustable flow rate High efficiency	Jerky flow rate High price Sensitive to particles Limited chemical resistance	Pure, slightly corrosive nonhazardous liquids P: 100 bars V: 20 m^3/h
Diaphragm	Accurate flow rate Adjustable flow rate High efficiency Less sensitive to particles Very good chemical resistance	Jerky flow rate High price Limited operating temperature	Slurry, corrosive, hazardous liquids P: 100 bars V: 20 m^3/h



Part no.	Name of part	Part no.	Name of part	Part no.	Name of part
1	Steam cylinder	10	Valve rod	19	Valve cover
2	Steam cylinder head	11	Valve rod lever	20	Valve stem
3	Steam piston	12	Liquid end piston rod	21	Valve spring
4	Steam piston ring	13	Gland	22	Valve
5	Steam end piston rod	14	Gland bushing	23	Valve seat
6	Condensate drain	15	Stuffing box		
7	Steam chest	16	Liquid piston		
8	Slide valve	17	Liquid piston rings		
9	Valve rod adjustment nut	18	Cylinder liner		

Fig. 4.68 Reciprocating pump-piston, duplex, direct-acting gas driven (process gas, air, steam).

4.9.2 Reciprocating pump, plunger power pump (Fig. 4.69)

Typical service	High pressure/low flow Gathering systems/pipelines Waterflood Well workover Mud pumps
Pressure range	500 to 6000 psi(3447 to 41,368 kPa)
Capacity range	10 to 600 gpm
Speed range	0 to 450 rpm
Max allowable temperature	400°F (204°C)
Standard materials	Available in duplex through nonuplex SS, although triplex most common Vertical configurations available up to 200 hp (149 kW) Crank driven with motor, turbine with gearbox, or engine drivers Steel liquid end Cast iron and steel power end Self-contained lubrication system
Specification	API 674 Variable speed or flow bypass

Typical control method	
Advantages	Higher pressures available than with piston pumps (up to 30,000 psi) Self-priming Constant delivery at high efficiency over wide pressure range Minimum fluid emulsification Handles viscous stocks Can run dry for a limited time
Limitations	Pulsing flow Low capacity High first cost and maintenance cost Low tolerance for abrasives Subject to vapor lock at low suction pressure with high vapor pressure fluids

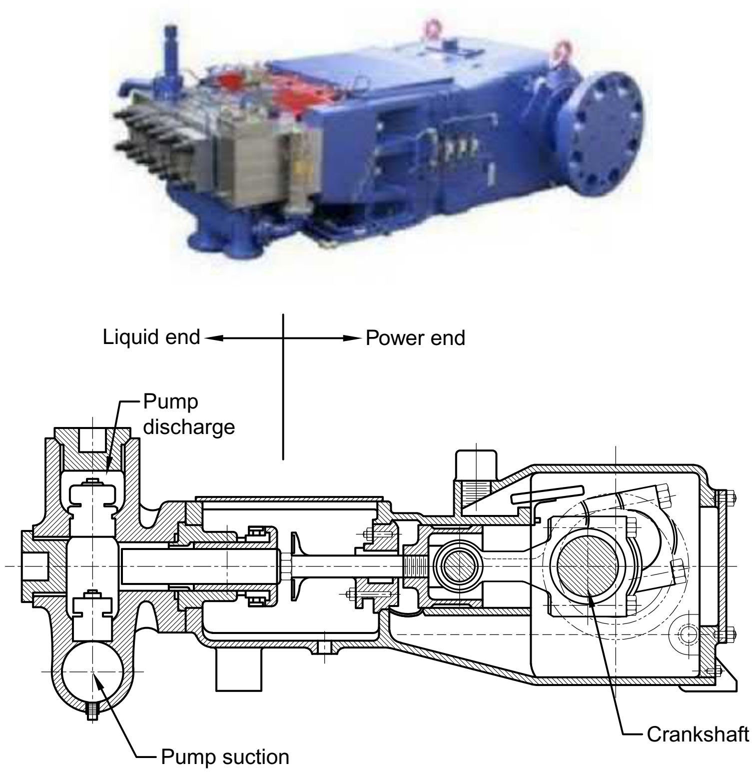


Fig. 4.69 Reciprocating plunger power pump. Top: assembled and Bottom: sectional. Courtesy of Worthington a Division of Ingersoll Dress Company.

4.10 Start-up considerations

4.10.1 Pump priming considerations

All pumps and associated piping must be purged of all vapors that may have accumulated since hydrostatic testing or while the unit was down. Purging a system is accomplished by priming to replace the vapors with liquids.

Procedure for a *positive static head* is as follows:

(1) Open the discharge vent valve.
(2) Open the discharge bypass valve.
(3) Partially open the suction block valve to allow fluid from the suction to fill the pump.
(4) Close the bypass valve and discharge vent valve when the fluid exits the vent valve.

Procedure for priming a pump with *static lift:*

(1) Open the discharge and suction block valves provided there is fluid in the discharge line.
(2) Pump fluid into the suction line and pump casing from an alternate source using an auxiliary pump.

4.10.2 Start-up guidelines

4.10.2.1 Initial start-up planning

Initial start-up planning should always be thoroughly planned and reviewed especially for critical services such as pumping hydrocarbons. All spare parts recommended by the manufacturer and deemed necessary by operating personnel should be on hand in case of a malfunction.

The operating manual should be reviewed with operating personnel. To help solve start-up problems, the manufacturer's representative should be present. All piping should be thoroughly cleaned before start-up and filled with fluid. For viscous or cold services, it may be necessary to provide heat tracing on the suction line.

4.10.2.2 Starting reciprocating pumps

The general start-up procedure is as follows:

(1) Suction and discharge block valves must be fully open. Failure to do this may result in severe damage to the pump.
(2) Discharge bypass valve can be turned back to a point where the pump is delivering the desired flow rate. For engine-driven pumps, the flow rate can be adjusted by changing the engine speed.
(3) The pump should run for several minutes under close supervision ensuring the desired fluid is being pumped, pumping system is not vibrating, and excessive heat buildup does not occur.
(4) After reaching operating temperatures, it is generally necessary to adjust the rod/plunger packing. Some leakage is necessary for lubrication and cooling. Trying to stop all leakage will cause heat buildup and premature pump failure.

4.11 Operations and maintenance considerations

4.11.1 Overview

Positive displacement pumps are very dependable machines. Extended pump life depends upon proper operation and good maintenance practices. Pump surveillance should be a daily routine for all people involved including engineers, operators, and maintenance personnel. This will ensure minimum down time, guard against the removal of a pump that is operating properly, and reduce expenses.

4.11.2 Troubleshooting

If a pump has no output or is producing less flow than expected, it is best to shut it down until the cause of the problem can be located. General troubleshooting procedure involves:

(1) Verify that liquid is reaching the pump.
- Check for an adequate supply of liquid at the suction source.
- Make certain that block valves on the suction line are fully open.
- Open bleed valves to bleed off trapped vapor.

(2) Check that downstream flowline is open.
- Check that all valves are open.

(3) If conditions seem normal to this point, briefly activate the pump and check for normal shaft rotation.
- If the shaft is not moving, or does not move at the correct speed, disconnect the pump from its driver and operate the driver by itself.
- If the driver operates properly, then the problem is in the pump.
- If not, ask a mechanic or electrician to inspect the driver.

(4) With the driver disconnected, rotate the pump shaft and listen for any abnormal noise (a stethoscope can be helpful).
- Remove the crankcase cover, rotate the crankshaft, and look for damage or misalignment.
- Check the crankshaft for bent or broken parts.

(5) Check packing for signs of scorching or poor lubrication.

(6) Check valves for damage.

4.11.3 Crankcase inspections and maintenance

The following inspections can be routinely performed without taking the pump out of service. Even so, the pump must be blocked in and depressurized and the pump driver must be locked out before the crankcase is opened.

(1) Crankcase oil should be sampled and inspected for brass or steel particles; these indicate abnormal bearing wear.
- Also check for water in the oil.

(2) Check cranks and connecting rods for discoloration caused by heat or misalignment.

(3) Turn the crankshaft manually (it may be necessary to first disconnect the driver).
- Listen for binding or abnormal bearing noise.

(4) Measure crankpin bearing clearances.
  - Four to six thousandths of an inch is a normal clearance. A higher reading could indicate bearing or crankpin wear.
  - Clearances can be adjusted by adding or removing some of the shims that separate the two halves of the bearing.

(5) Check crankshaft end play (i.e., the extent the shaft can move along its own length). This is done with a dial indicator and a pry bar.
  - Check the reading against manufacturer's specifications.
  - If endplay is excessive, shims must be removed from the crankshaft end bearing housing.
  - Shims of the same thickness must be removed from both housing until end play is within specifications.

(6) Check the clearance between the top of each crosshead and the crosshead guide with a feeler gauge.

If the above inspections indicate that bearings need to be changed, excessive clearance exists or a problem has been detected that is not immediately apparent, then the bearings, crankshaft, and crossheads will have to be removed for additional examination. In most cases, this requires a trip to the shop.

4.11.4 Liquid end inspection and maintenance

Inspecting the liquid end of the pump involves inspecting the valves, packing, and plunger (or piston). These inspections cannot be performed unless the pump has been

- isolated
- depressurized
- driver has been locked out

Valve chambers should be opened carefully since pressure still may be trapped inside even when the pump has been isolated and depressurized.

Inspection procedures include

(1) All valves should be removed and disassembled.
  - Worn or broken parts must be replaced.
  - Seating surfaces must be smooth.
  - Valve cavities must be cleaned before valves are reinstalled.

(2) The plunger must be removed and inspected for wear, especially at the point where it comes in contact with the packing.
  - The diameter of the plunger should be checked at several points with an outside micrometer.
  - If the plunger is badly worn or damaged, it must be replaced.

(3) New packing rings should be installed whenever the liquid end of a pump is overhauled.
  - If the pump is equipped with a force-feed lubricator, a lantern ring must be installed between two sets of packing rings immediately below the lubrication point on the packing box. Otherwise, lubricant will not be distributed to the packing and the packing will quickly wear out.

(4) The packing box bore should be inspected before new rings are installed. If the bore is worn, consider installing a stainless steel sleeve.

Example 4.3A. Reciprocating pump selection (field units)

Given:

An onshore water injection station has a pumping requirement of 6500 BWPD at a discharge pressure of 2150 psig. The discharge pressure requirement will remain relatively constant throughout the life of the field. As the reservoir fills up the injection flow rate will decrease to ~3000 BPD. The water is obtained from a source water well with a specific gravity of 1.0 and a temperature of 80°F. Fig. 4.70 illustrates the system that is to be considered for this problem. Table 4.22 is a sample Reciprocating Pump Data Sheet presenting the information required to select a pump.

In discussions with field operating personnel the following design criteria have been agreed:

(1) Pumps will be driven by a constant speed electric motor.
(2) Reciprocating pumps will be used.
(3) Bypassing of excess flow is acceptable as long as bypass rate can be minimized.
(4) Pump fluid end is to be a forging.
(5) Standby capacity is not required.

A Quintuplex pump is the First Choice with a triplex as an alternate.

Determine:

(1) Flow rate
(2) Discharge head
(3) Number of pumps
(4) Following mechanical features:
  (a) Valve assembly type
  (b) Type of wrist pin bearing
  (c) Type of packing
  (d) The pump material class is S-2, specify material for the following item: (1) fluid end and (2) plungers.

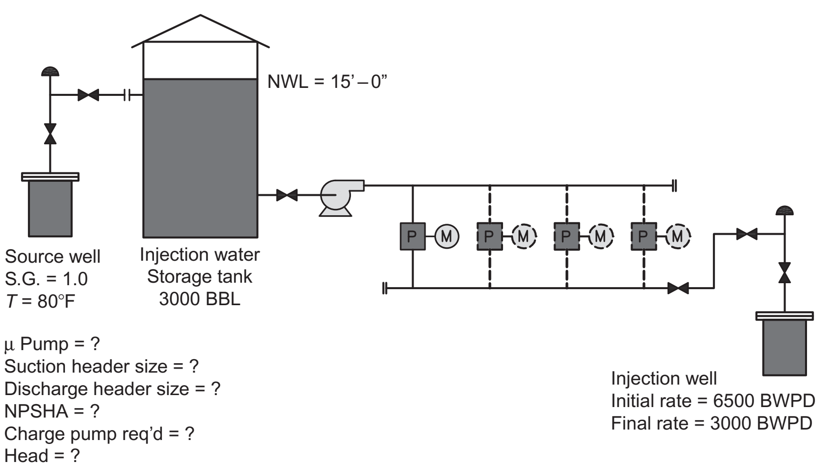


Fig. 4.70 Schematic flow diagram of a pump system.

Table 4.22 Example data sheet for Example 4.3A

A. Information required by pump manufacturer for establishing basic pump size

Flow (total flow for facility and flow for each pump)
Head
NPSH available
Liquid pumped
Pumping temperature
Pumped fluid = corrosive/erosive

B. Specify construction grade

API 674_______
Other_______

C. Specify major construction features

1. Materials of construction
   Fluid end_______
   Plunger_______
   Valve_______
   Valve spring_______
   Packing gland_______
   Cylinder liner_______
2. Valve assembly type
   Caged_______
   Tapered seat_______
3. Bearing type
   Manufacturer's standard_______
   Rolling contact_______

*Unless specific pump application dictates otherwise, bearing type should be Manufacturer's standard

4. Packing type
   Square cut_______
   V-ring_______
5. Lubrication type
   Splash_______
   Pressure_______
   Plunger lubrication_______
6. Pressure rating of fluid end 120% of design discharge pressure_______
7. Pump baseplate
   Common baseplate, pump skid and driver_______
   Drain pan_______
   Skid material_______
   Skid leveling screws_______

D. Specify testing

Performance (witness/nonwitness)_______
Hydrostatic (witness/nonwitness)_______
NPSH (witness/nonwitness_______

E. Driver type

Electric motor_______
Gas engine_______
Other_______

Solution:

(1) The flow rate (final and initial)

$$\text{Initial flow rate} = 6500\,\text{BWPD} \times \frac{.0291\,\text{gpm}}{\text{BWPD}}$$
$$= 189\,\text{gpm}$$

$$\text{Final flow rate} = 3000\,\text{BWPD} \times \frac{0291\,\text{gpm}}{\text{BWPD}}$$
$$= 88\,\text{gpm}$$

(2) Discharge head

$\text{Head} = \frac{2.31}{SpGr} \times \Delta P = \frac{2.31(2150)}{1.0} = 4966\,\text{ft}$

Calculate the $NPSH_A$ from the following:

ΔP = [(SG)(TDH)]/(2.31)
= [(1)(15)]/(2.31)
= 6 psi

The preliminary data indicates that since there is very little $NPSH_A$, a small charge pump will likely be required.

After pump manufacturers offerings have been reviewed by the engineer, some decisions will have to be made regarding:

- NPSH of charge pump
- $NPSH_A$ for pumps selected
- Size of suction and discharge headers
- Losses due to friction

An estimate for electric power load per pump can be made from the following (assume the charge pump has a discharge pressure of 244 psi):

HHP = [(189)(0.5)(2150 − 6 − 244)]/(1714)
= 104.8 hp

Assume a 90% mechanical efficiency:

BHP = (104.8)/(0.90)
= 116.4 hp

Discharge head

Head = [(231)ΔP]/(SG)
= [(2.31)(2150)]/(1.0)
= 4966 ft

(3) Number of pumps

Option I: Two pumps each with 100% capacity (189 gpm)

This pump system will definitely be the most reliable as only one pump will need to be operated and 100% standby is provided. However, the installed cost will be high and the amount of flow bypassed in the later years' operation will be ~50% resulting in wasted horsepower.

Option II: Two pumps each with 50% capacity (94.5 gpm)

This pump system does not provide standby capacity. However, it does have the following advantages:

(a) Lower installed cost
(b) Less bypassed water in later years' operation

Option III: Three pumps each with 50% capacity (94.5 gpm)

This will cost more than Option II, but has the advantage of providing standby capacity.

Option IV: Three pumps each with 33% capacity (63 gpm)

This selection will have a higher maintenance, installation, and operating costs than Option I and II. There is less loss of capacity when one pump is down, as compared to the two—50% pump system. In later years' operation more water will have to be bypassed.

Option V: Four pumps each with 33% capacity (63 gpm)

This is identical to Option IV except standby capacity is provided.

Selection

Option II is selected since standby capacity at the initial throughput rate is really not necessary.

(4) Mechanical features
(a) Valve assembly type

Caged valve should be specified as it is currently the most common available, and does not have to be forced into fluid end, which brings on the possibility of damaging.

(b) Type of wrist pin bearing

Due to the low suction pressure journal bearings are acceptable for this application.

(c) Type of packing

Either type is suitable for this application. Since fluid will contain some sand V-ring packing should be provided or square cut packing with a lantern ring.

(d) Pump materials

The material selection of the pump parts should be in accordance with NACE RP 0475. For this problem, assume that the water is aerated and that no H_2S is present. Table 4.23 presents a completed data sheet for this problem. It is worth noting that some thought should be given to the pressure rating of the piping and pump fluid end. The 2150 psi discharge pressure requirement and subsequent 900 ASME pressure rating is close to the maximum design working pressure (DWP) of valves, flanges, and fittings, leaving no margin for PSH and PSV setting. The reservoir engineer should be interviewed closely to determine the accuracy and need for this surface injection pressure. If still required, a change in fluid end to a higher pressure rating would probably be recommended.

This example illustrates that all pumping situations present engineer with several choices, alternatives, and unanswered questions. In the real world decisions must be made based on individual experience, on the engineering principles presented here, and on sound engineering judgment.

Table 4.23 Completed reciprocating pump data sheet for Example 4.3A

A. Information required by pump manufacturer for establishing basic pump size	
Flow	
Total flow for facility	189 gpm
Flow for each pump	94.5 gpm (2 required)
Head	15 ft
NPSH available ($NPSH_A$)	6 psi
Liquid pumped	Salt water—$SG = 1.0$
Pumping temperature	80°F
Pumped fluid = corrosive/erosive	Mildy corrosive
Pressure rating of fluid end	900 ANSI
B. Specify construction grade	
API 674	
Other	NACE NR04-75/fluid parts
C. Specify major construction features	
1. Material of construction	
Fluid end	Cast aluminum bronze
Plunger	Ceramic
Valve	Mfg std—note in response
Valve spring	Mfg std—note in response
Valve cage	Mfg std—note in response
Packing gland	Mfg std—note in response
Cylinder liner	Mfg std—note in response
2. Valve assembly type	
Manufacturer's standard	
Caged	Yes
Tapered seat	
3. Bearing type	
Manufacturer's standard	Yes
Rolling contact	
Hydrodynamic	
4. Packing type	
Manufacturer's standard	
Square cut	Acceptable w/lantern ring
V-ring	Preferred
5. Lubrication type	
Manufacturer's Standard	Yes
Splash	
Pressure	Preferred
6. Pump baseplate	
Common baseplate, pump skid, and driver	Skid mounted
Drain pan	None required

Continued

Table 4.23 Continued

Skid material	A-36 CS
D. Required testing	
Performance (witness/nonwitness) Hydrostatic (witness/nonwitness) $NPSH_R$ (witness/nonwitness)	Witness Witness Nonwitness
E. Driver type	
Electric motor Gas engine Other	TEFC electric motor, 440 VAC 65% reduced voltage motor starter required
F. Power transmission	
Direct coupled Belt driver	 Yes

Example 4.3B. Reciprocating pump selection (SI units)

Given:

An onshore water injection station has a pumping requirement of 43 m^3/h at a discharge pressure of 14,800 kPa. The discharge pressure requirement will remain relatively constant throughout the life of the field. As the reservoir fills up the injection flow rate will decrease to ~20 m^3/h. The water is obtained from a source water well with a specific gravity of 1.0 and a temperature of 27°C. Fig. 4.70 illustrates the system that is to be considered for this problem. Table 4.22 is a sample Reciprocating Pump Data Sheet presenting the information required to select a pump.

In discussions with field operating personnel the following design criteria have been agreed:

1. Pumps will be driven by a constant speed electric motor.
2. Reciprocating pumps will be used.
3. Bypassing of excess flow is acceptable as long as bypass rate can be minimized.
4. Pump fluid end is to be a forging.
5. Standby capacity is not required.

A Quintuplex pump is the First Choice with a triplex as an alternate.

Determine:

1. Flow rate
2. Discharge head
3. Number of pumps
4. Following mechanical features:
   - **(a)** Valve assembly type
   - **(b)** Type of wrist pin bearing

(c) Type of packing
(d) The pump material class is S-2, specify material for the following item: (1) fluid end and (2) plungers.

Solution:

1. The flow rate (final and initial)
Initial flow rate = 43 m^3/h
Final flow rate = 20 m^3/h

Calculate the $NPSH_A$ from the following:

$$\Delta P = \frac{(SG)(\text{Head in in})}{0.102}$$
$$\Delta P = \frac{(1)(4.6)}{0.102}$$
$$\Delta P = 45\,\text{kPa}$$

The preliminary data indicates that since there is very little $NPSH_A$, a small charge pump will likely be required.

After pump manufacturers offerings have been reviewed by the engineer, some decisions will have to be made regarding:

- NPSH of charge pump
- $NPSH_A$ for pumps selected
- Size of suction and discharge headers
- Losses due to friction

An estimate for electric power load per pump can be made from the following (assume the charge pump has a discharge pressure of 244 psi):

$$\begin{aligned} HHP &= [(43)(0.5)(14{,}800 - 45 - 1680)]/(3600) \\ &= 78 \text{ kW} \end{aligned}$$

Assume a 90% mechanical efficiency:

$$\begin{aligned} BHP &= (78)/(0.90) \\ &= 86.8 \text{ kW} \end{aligned}$$

2. Discharge head

$$\text{Head} = \frac{0.102}{SG} \times \Delta P = \frac{0.102(14,800)}{1.0} = 1510\text{m}$$

3. Number of pumps

Option I: Two pumps each with 100% capacity (189 gpm)

This pump system will definitely be the most reliable as only one pump will need to be operated and 100% standby is provided. However, the installed cost will be high and the amount of flow bypassed in the later years' operation will be ~50% resulting in wasted horsepower.

Option II: Two pumps each with 50% capacity (94.5 gpm)

This pump system does not provide standby capacity. However, it does have the following advantages:

(1) Lower installed cost.
(2) Less bypassed water in later years' operation.

Option III: Three pumps each with 50% capacity (94.5 gpm)

This will cost more than Option II, but has the advantage of providing standby capacity.

Option IV: Three pumps each with 33% capacity (63 gpm)

This selection will have a higher maintenance, installation, and operating costs than Option I and II. There is less loss of capacity when one pump is down, as compared to the two—50% pump system. In later years' operation more water will have to be bypassed.

Option V: Four pumps each with 33% capacity (63 gpm)

This is identical to Option IV except standby capacity is provided.

Selection

Option II is selected since standby capacity at the initial throughput rate is really not necessary.

4. Mechanical features
(a) Valve assembly type

Caged valve should be specified as it is currently the most common available, and does not have to be forced into fluid end, which brings on the possibility of damaging.

(b) Type of wrist pin bearing

Due to the low suction pressure journal bearings are acceptable for this application.

(c) Type of packing

Either type is suitable for this application. Since fluid will contain some sand V-ring packing should be provided or square cut packing with a lantern ring.

(d) Pump materials

The material selection of the pump parts should be in accordance with NACE RP 0475. For this problem, assume that the water is aerated and that no H_2S is present. Table 4.24 presents a completed data sheet for this problem. It is worth noting that some thought should be given to the pressure rating of the piping and pump fluid end. The 14,800 kPa discharge pressure requirement and subsequent 900 ASME pressure rating is close to the maximum Design Working Pressure (DWP) of valves, flanges, and fittings, leaving no margin for PSH and PSV setting. The reservoir engineer should be interviewed closely to determine the accuracy and need for this surface injection pressure. If still required, a change in fluid end to a higher pressure rating would probably be recommended.

This example illustrates that all pumping situations present engineer with several choices, alternatives, and unanswered questions. In the real world decisions must be made based on individual experience, on the engineering principles presented here, and on sound engineering judgment.

Table 4.24 Example data sheet for Example 4.3B

A. Information required by pump manufacturer for establishing basic pump size	
Flow	
Total flow for facility	
Flow for each pump	
Head	
NPSH available ($NPSH_A$)	
Liquid pumped	
Pumping temperature	
Pumped fluid = corrosive/erosive	
Pressure rating of fluid end	
B. Specify construction grade	
API 674	
Other	
C. Specify major construction features	
1. Material of Construction	
Fluid end	
Plunger	
Valve	
Valve spring	
Valve cage	
Packing gland	
Cylinder liner	
2. Valve assembly type	
Manufacturer's standard	
Caged	
Tapered seat	
3. Bearing type	
Manufacturer's standard	
Rolling contact	
Hydrodynamic	
4. Packing type	
Manufacturer's standard	
Square cut	
V-ring	
5. Lubrication type	
Manufacturer's standard	
Splash	
Pressure	
6. Pump baseplate	
Common baseplate, pump skid, and driver	
Drain pan	
Skid material	

Continued

Table 4.24 Continued

D. Required testing	
Performance (witness/nonwitness) Hydrostatic (witness/nonwitness) $NPSH_R$ (witness/nonwitness)	
E. Driver type	
Electric motor Gas engine Other	
F. Power transmission	
Direct coupled Belt driver	

4.12 Exercises

1. The most commonly used positive displacement pumps in production operations are______________________________________and ______________________________________
2. Using Fig. 4.70, identify the components of a reciprocating plunger pump.

a. __________________ h. __________________
b. __________________ i. __________________
c. __________________ j. __________________
d. __________________ k. __________________
e. __________________ l. __________________
f. __________________ m. __________________
g. __________________ n. __________________

3. The primary disadvantage of a reciprocating pump is
   a. Lower efficiencies
   b. High $NPSH_R$
   c. Pulsating flow
   d. Cannot be used for water service
   e. Low pressure output
4. In reciprocating pumps, the fluid is displaced by a ________________ or a ____________________________________.
5. The displacement of fluid by a piston or plunger pump is ____

Questions 6–9

A positive displacement pump is required to transfer 6000 barrels per day of crude oil from a storage facility to a refinery. The pump is to be operated 80% of the time. The discharge pressure required is 975 psig. Using the figure on the following page.

6. What type pump would be selected?
  - **a.** Type A
  - **b.** Type B
  - **c.** Type C
  - **d.** Type D
  - **e.** Type E

7. What is the maximum discharge pressure?
  - **a.** 800 psig
  - **b.** 1025 psig
  - **c.** 2700 psig
  - **d.** 670 psig
  - **e.** 3200 psig

8. What size plungers would be used?
  - **a.** 3 x 4-3/8
  - **b.** 2x5-1/2
  - **c.** 1-1/2x2-1/2
  - **d.** 3-1/4x5-1/2
  - **e.** 2-1/4x4-3/8

9. What is the maximum rpm for this type pump?
  - **a.** 550
  - **b.** 450
  - **c.** 380
  - **d.** Any of the above
  - **e.** None of the above

		Displacement			
Pump type	**Size**	**Maximum rpm**	**Barrel, per day**	**Gallon. per minute**	**Maximum discharge pressure psig**
A	¾ x 2 ½	550	270	7 9	5660
	3/8 x 2 ½	550	367	10 7	4160
	1 x 2 ½	550	480	14	3190
	1 1/8 x 2 ½	550	607	17.7	2520
	1 ¼ x 2 ½	550	750	21 9	2040
	1 3/8 x 2 ½	550	910	26.5	1680
	1 ½ x 2 ½	550	1080	31 6	1420
B	1 x 4 3/8	450	690	20 1	6050
	1 1/8 x 4 3/8	450	870	25 4	4780
	1 ¼ x 4 3/8	450	1076	31 4	3870
	1 3/8 x 4 3/8	450	1300	38 0	3200
	1 ½ x 4 3/8	450	1550	45 2	2700
C	1 ¼ x 5 ½	380	1140	33 3	6930
	1 1/8 x 5 ½	380	1380	40 3	5720

Continued

Continued

Pump type	Size	Displacement			Maximum discharge pressure psig
		Maximum rpm	Barrel, per day	Gallon. per minute	
	1 ½ x 5 ½	380	1645	48 0	4810
	1 3/8 x 5 ½	380	1930	56 3	4100
	1 ¾ x 5 ½	380	2240	65 3	3530
	1 2/8 x 5 ½	380	2570	74.9	3080
D	1 1/8 x 4 3/8	450	1300	38	3200
	1 ½ x 4 3/8	450	1550	45.2	2700
	1 3/8 x 4 3/8	450	1820	53	2300
	1 ¾ x 4 3/8	450	2110	61.5	1970
	2 x 4 3/8	450	2750	80.3	1510
	2 ¼ x 4 3/8	450	3450	101.5	1200
	2 ½ x 4 3/8	450	4300	125.5	970
	2 ¾ x 4 3/8	450	5200	151.8	800
	3 x 4 3/8	450	6190	180.7	670
E	1 ¾ x 5 1/2	380	2240	65.3	3530
	2 x 5 ½	380	2920	85.3	2700
	2 ¼ x 5 ½	380	3700	108	2140
	2 ½ x 5 ½	380	4570	133	1730
	2 ¾ x 5 ½	380	5530	161	1430
	3 x 5 ½	380	6580	192	1200
	3 ¼ x 5 ½	380	7690	224	1025
	3 ½ x 5 ½	380	8950	261	885

10. To account for additional energy that must be provided on the suction side of a reciprocating pump the $NPSH_A$ is reduced by the ____

a. Pulsations per minute
b. C factor
c. Acceleration head
d. Plunger capacity × 80%
e. Packing efficiency

11. Using Fig. 4.71, identify the components of a reciprocating pump installation

a. ____________________ **f.** ____________________
b. ____________________ **g.** ____________________
c. ____________________ **h.** ____________________
d. ____________________ **i.** ____________________
e. ____________________ **j.** ____________________

12. Name two of the most commonly used valves in a positive displacement pump are _______________ and ____________________ valves.

13. Plunger pumps use ____________________ packing, while piston pumps use ____________________ packing.

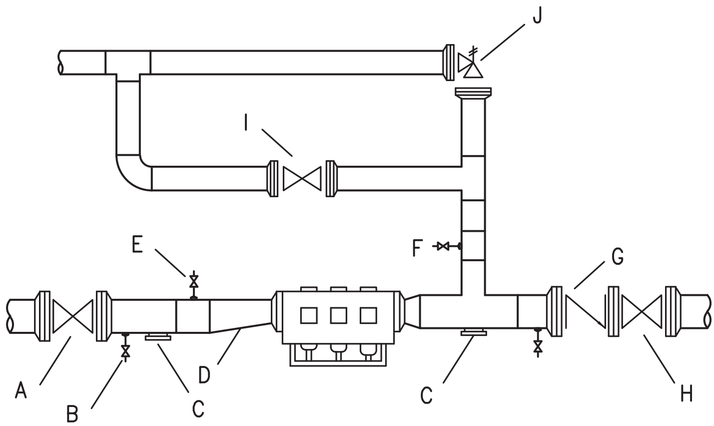


Fig. 4.71 Schematic for Problem 11.

14. To smooth the discharge flow from a reciprocating pump ______________________________ are most often used.
   a. Control valves
   b. Pulsation dampeners
   c. Acoustical filters
   d. All of the above
   e. B and C only

15. A discharge check valve for a reciprocating pump should be a ________________________________ type.
   a. Globe
   b. Piston
   c. Plunger
   d. Swing
   e. Water

16. What is the head required to accelerate 8000 barrels of crude oil per day from a storage tank to a pipeline for the following conditions:
   Pump: Reciprocating single-acting duplex, 300 rpm
   Suction piping: 6-in., Schedule 40, ID = 6.065 in., 5 feet long
   a. 3 ft
   b. 8 ft
   c. 12 ft
   d. 17 ft
   e. 29 ft

17. Liquid carbon dioxide at −10°F is to be pumped into a reservoir in a miscible flood EOR project. The injection pump is a quintuplex pump with 1.5-in. diameter plungers and a 5-in. stroke. Pump speed is 282 rpm and the pump capacity is 46 gpm. The suction line is 50-ft long and has a 4.026-in. diameter. Calculate the acceleration head.

18. It is desired to move 10,000 BPD of a 57° API condensate from atmospheric storage tanks to a pipeline operating at 500 psia. The distance from the storage area to the main pipeline is 10 miles. What size pipe would you recommend for the lateral line? What size pump would you specify (power and number of stages)? The pump must have a minimum efficiency of 60%. It is recommended that the pump speed be 330 rpm. The condensate viscosity is 0.9 cp.

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Rotary pumps

5

Nomenclature

A	plunger or piston cross-sectional area, mm^2 (in^2)
a	piston rod cross-sectional area, mm^2 (in^2)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
D	pump displacement, m^3/hr (gpm)
d	displacement per pumping chamber, m^3 (gal)
d'	pump piston or plunger diameter, mm (in)
d_w	diameter of wrist pin, mm (in)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/sec
g	acceleration due to gravity, 9.81 m/sec^2 (32.2 ft/sec^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
H_f	pipe friction loss, m (ft)
H_p	total head required for pump, m (ft)
HHP	hydraulic horsepower, kW (hp)
K	a factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in)
m	number of pistons, plungers, or diaphragms
NPSH	net positive suction head, m (ft)
NPSHA	net positive suction head available, m (ft)
NPSHR	net positive suction head required, m (ft)
n	stroke rate or crank revolutions per min, rps (rpm)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
ΔP	static pressure, kPa (psi)
Q	flow rate, m^3/hr (ft^3/sec)
Q'	flow rate, m^3/hr (BPD)
q	flow rate, m^3/hr (gpm)
S	stress, kPa (psi)
S'	valve slip, percent
s	stroke length, mm (in)
SG	specific gravity of liquid relative to water
V	velocity in suction line, m/sec (ft/sec)

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00005-3

Vol	volume of surge tank, m^3 (ft^3)
$(Vol)_g$	required gas volume, m^3 (ft^3)
Z	elevation above or below pump centerline datum, m (ft)
ρ	density of fluid, kg/m^3 (lb/ft^3)

5.1 Engineering principles

5.1.1 Background

Rotary pumps are positive displacement pumps, but unlike reciprocating pumps, have relatively steady, nonpulsating flow. Rotation of the rotor(s) within the casing traps pockets of liquid at suction conditions elevates the fluid pressure and then pushes the fluid out the discharge. Rotary pumps are normally limited to services where the fluid viscosity is very high or the flow rate is too low to be handled economically by other pumps. The most common applications are as follows:

- Supplying fuel oil to burners
- Transferring gasoline, fuel oil, and diesel fuel to day tanks, and
- Circulating lube oils through engines, turbines, reduction gears, and process machinery bearings

The capacity of a rotary pump, like other positive displacement pumps, is directly proportional to pump speed. Regardless of capacity or pump speed, the discharge pressure of a rotary pump is that required to force the fluid through the discharge system. The discharge pressure is limited only by the mechanical design of the pump casing, or by the fluid viscosity and torque capability of the driver.

Rotary pumps displace a fixed quantity of fluid for every revolution of the driver shaft. They have different pumping elements such as vanes, lobes, gears, and screws.

Fig. 5.1 shows three of the most commonly used rotary pumps in production operations

- Eccentric rotor
- Gear
- Screw

5.1.2 Slip

Many manufacturers rate rotary pumps by capacity rather than displacement. Capacity is displacement less slip.

Slip (S) is the quantity of fluid that leaks from the high-pressure discharge to the low-pressure suction. Thus the actual capacity of a rotary pump is less than the calculated theoretical capacity. The theoretical capacity is reduced by recirculation back through the clearances between the rotor(s) and casing. The recirculation fluid is termed “slip.” Slip occurs because all rotary pumps require clearances between the rotating elements and the pump housing. These clearances provide a leak path between the discharge and suction sides. A pump with large clearances, due to machining tolerances

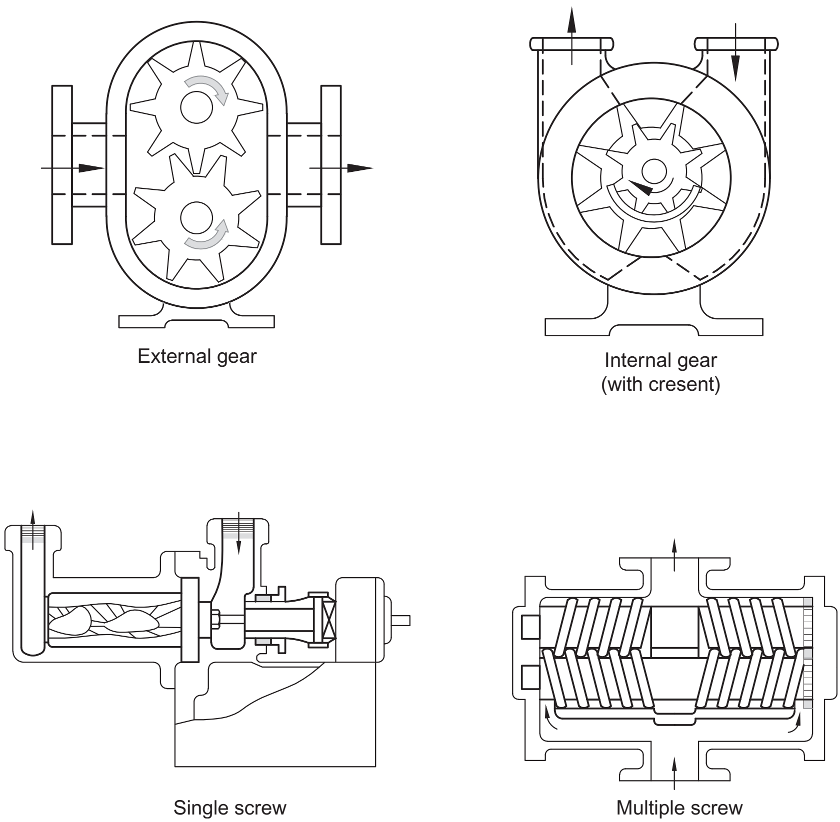


Fig. 5.1 Three of the most commonly used rotary pumps.

or wear, exhibits a proportionally larger slip. Slip is directly proportional to the pressure differential across the pump and is inversely proportional to the fluid viscosity.

The delivered capacity (Q) of a rotary pump is calculated by Eq. (5.1):

$$Q = Q_t - S \tag{5.1}$$

where:
Q = delivered capacity
Q_t = theoretical capacity
S = slip

The capacity of a rotary pump is reduced by:

- Decreasing viscosity
- Increasing differential pressure
- Increasing internal clearances between rotating and stationary parts
- Decreasing pump speed

5.1.3 Volumetric efficiency

Manufactures characterize the amount of slip as volumetric efficiency. The volumetric efficiency of a rotary pump is determined as follows:

$$E_V = \frac{Q}{Q_t} = \frac{(Q - S)}{Q_t} \tag{5.2}$$

The volumetric efficiency (E_V) of a rotary pump is a function of pump:

- Speed
- Viscosity
- Differential pressure

The E_V is one of the components that affect the pump's overall efficiency which in turn impacts pump sizing and selection. Fig. 5.2 shows the relationship between volumetric efficiency, capacity, and rated speed. Other factors that affect the pump's overall efficiency include:

- Heating losses from fluid friction
- Mechanical losses in the bearings
- Drag due to viscosity

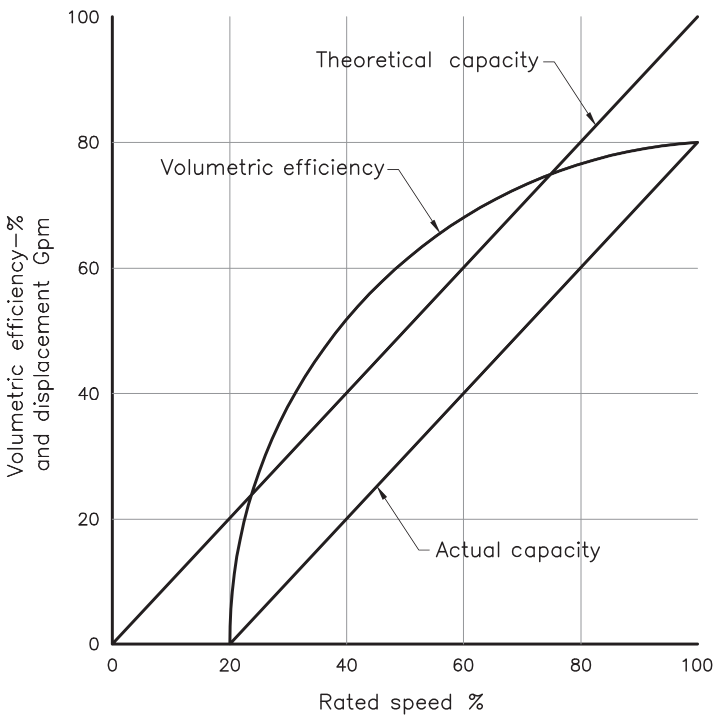


Fig. 5.2 Relationship between volumetric efficiency, capacity, and rated speed.

5.1.4 Mechanical efficiency

The mechanical efficiency (E_m) of a rotary pump is:

$$E_m = \frac{\text{HHP}}{\text{HP}} \times 100 \tag{5.3}$$

where:
HP = input horsepower
HHP = hydraulic horsepower

$$= \frac{(Q)(\text{DP})}{1715}$$

Q = flow, gpm
DP = differential pressure, psi

The mechanical efficiency of a rotary pump ranges from 60% to 70%. That said, with low slip and very favorable conditions the E_m may be as high as 80%, or as low as 50% under unfavorable conditions. Fig. 5.3 illustrates the aforementioned efficiencies and capacities, with resultant horsepower, assuming constant speed and viscosity. Similar curves could be drawn as a function of speed or viscosity. The "output horsepower" in Fig. 5.3 is the HHP described in Eq. (5.3).

5.1.5 Suction considerations

Most problems in rotary pump applications are the result of suction rather than discharge conditions. Rotary pumps are self-priming but are not designed to run dry for extended periods. Suction conditions have a great effect on performance. It is important to completely fill the moving cavities of the pump to ensure quiet and

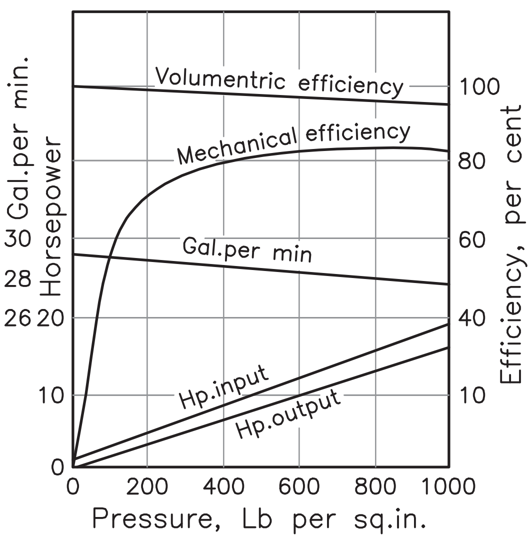


Fig. 5.3 Rotary pump curve types. Courtesy of Kristal and Annett, 2nd ed., 1953.

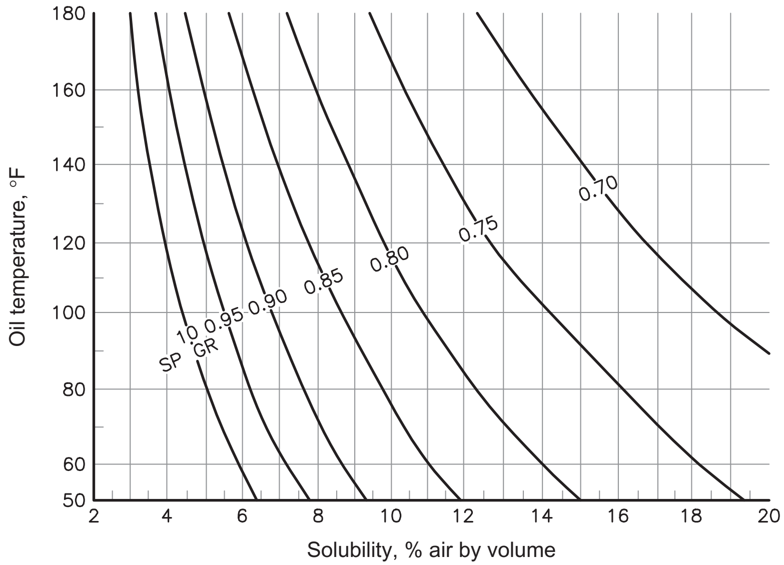


Fig. 5.4 Solubility of air in oil.
Courtesy of Kristal and Annett, Pumps, Types, Selection, Installation, Operation and Maintenance, 2nd ed., McGraw Hill, 1958.

efficient operation. For best operation, there must be enough fluid at the suction port to keep the pumping chamber completely filled.

As the viscosity of a fluid increases the greater the resistance to flow. Thus the rate of filling the moving cavities is slower for higher viscosity fluids. In addition, if the cavities are not moving too fast they will not fill completely. The speed of the pump must then be limited to satisfy the effect of high viscosity and the $NPSH_A$. Rotary pumps can handle viscosities ranging from 2 to 225,000 centistokes (35 to 1,000,000 SSU) but are normally used in the 20 to 50,000 centistokes (100 to 250,000 SSU) range. Fig. 5.4 illustrates the solubility of air in oil.

Rotary pump manufacturers sometimes report the $NPSH_R$ as *maximum suction lift available* (MSL_A).

$$NPSH_R = (\text{Inlet Vesel Pressure}) - MSL_A \tag{5.4}$$

The MSL_A reported by manufacturers should not be exceeded. This is the same as saying $NPSH_A$ must be greater than $NPSH_R$, that is, a margin of at least 3 psi between $NPSH_A$ and $NPSH_R$.

Rotary pumps cannot handle nonlubricating fluids such as water or fluids containing hard or abrasive particles. Rotary pumps can handle compressible fluids, such as a mixture of oil and gas, for short periods of time without losing prime. This must be taken into account when specifying liquid capacity. The effects of dissolved gas may be reduced by lowering the suction lift through relocating the pump, increasing the suction line size, or changing the piping arrangement.

When handling compressible fluids, the volume within each closure is reduced as it comes in contact with the discharge pressure. This produces pressure pulsations where the intensity and frequency depend on the discharge pressure, the number of closures per revolution, and the speed. Under some conditions, the pressure pulsations are of high magnitude and can cause damage to the piping system and/or pump and will normally be accompanied by undesirable noise.

One must be aware that there is a difference between entrained or dissolved gas and liquid vapor. Adequate $NPSH_A$ of the liquid must still be maintained for acceptable performance, even though the pump may be specified to handle entrained or dissolved gas or vapor.

5.2 Rotary pump types

There are three general types of rotary pumps: eccentric rotor, gear, and screw. There are several variations of each.

5.2.1 Eccentric rotor

Eccentric rotor pumps are available in several designs, most of which have any practical use in the oil and gas industry.

5.2.1.1 Sliding vane

The sliding vane is the most commonly used eccentric vane pump used in the oil and gas industry. It consists of an eccentrically rotating drum with radial slots that house sliding vanes. The vanes slide in and out of the rotor. The rotor is mounted off center in the casing. As the vanes rotate past the suction port they slide out of the rotor while maintaining constant contact with the casing. The vanes form and seal the fluid cavities that shrink and elevate the fluid pressure as the drum rotates. Springs or sealer rings (Fig. 5.5) help hold the vanes against the casing, thus the vanes make a close seal, or fit, against the casing wall. Trapped fluid is forced from the suction port to the discharge port. Sliding vanes are capable of delivering medium capacity and head, constant flow rate for a set speed, and provides a uniform discharge pressure with negligible pulsation. They work well with low-viscosity fluids and are somewhat self-compensating for wear. However, sliding vane pumps are inexpensive and unreliable due to frictional wear and vane breakage. They are not suitable for use with highly viscous fluids since thicker fluids interfere with sliding action of the vanes.

5.2.1.2 Flexible vane

The flexible vane is similar to the sliding vane except that the vanes are generally a soft, pliable material and are integral with the rotor. As the rotor turns, the vanes bend and conform to the eccentric shape of the cylinder. They are simple, inexpensive, and are capable of developing a high vacuum. However, they can only be used in

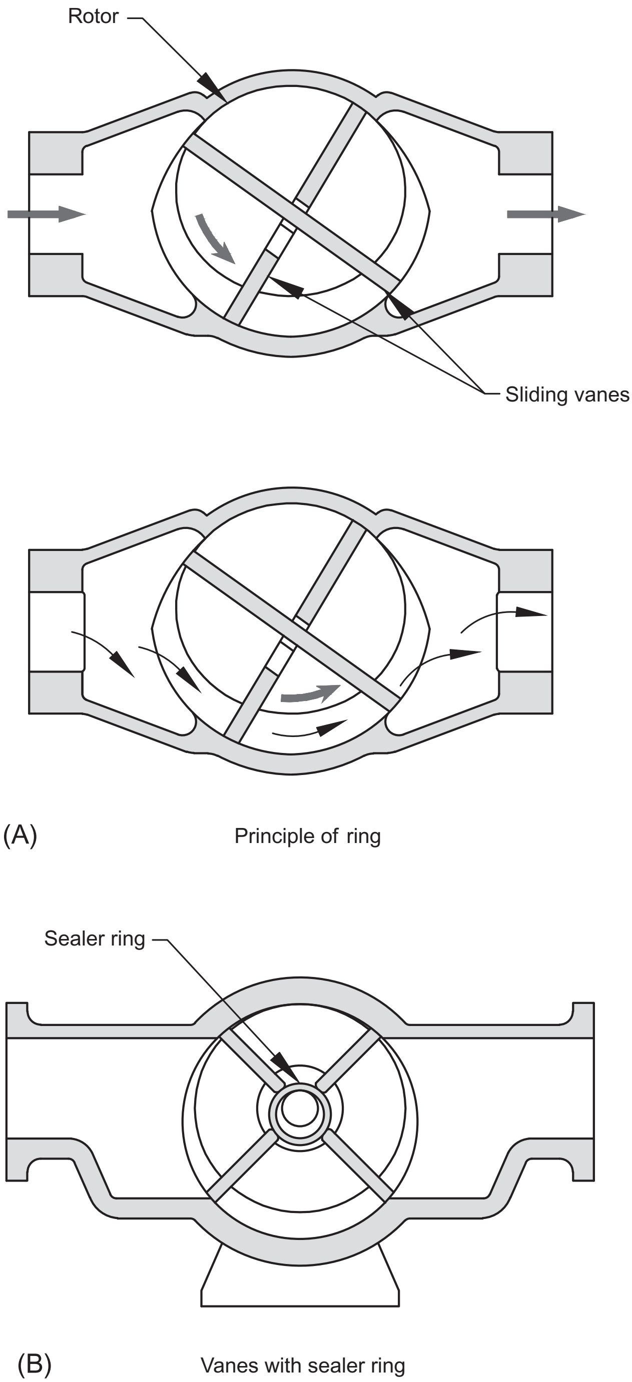


Fig. 5.5 Sliding vane rotary pump: (A) Vanes with springs; (B) Vanes with sealer rings.

low-capacity, low-pressure nondemanding service. They should not be allowed to run dry and should only be used with low-temperature fluids.

5.2.2 Gear pumps

There are three general types of gear pumps: external gear, internal gear, and lobe. Gear pumps are used in lube oil, heavy fuel, crude oil transfer services, and in high pressure bearing and seal oil services for centrifugal compressors. They generally cost less than screw pumps but have a shorter life.

5.2.2.1 External gear

The external gear pump (Fig. 5.6) consists of two equal sized meshing gears, one driving and the other an idler, which rotates inside a housing (Fig. 5.7). As the gears unmesh at the suction side of the pump, a vacuum is formed. Atmospheric pressure forces the fluid into the pump where the fluid is carried between the gear teeth and the case to the discharge. At the discharge, the meshing of the gear teeth creates a boundary that prevents the fluid from returning to the suction. The rotors of external gear pumps are supported by antifriction bearings. Gear pumps operate equally well when driven in either direction. Precautions should be taken to ensure that the shaft rotation is correct when special features, such as built-in relief valves or a bleed back of the shaft seal, are used.

Variations include models that use multiple sets of gears on one shaft so as to produce more capacity and use spur, helical, and herring bone gears. Fig. 5.8 shows the three commonly used types of gears in gear pumps.

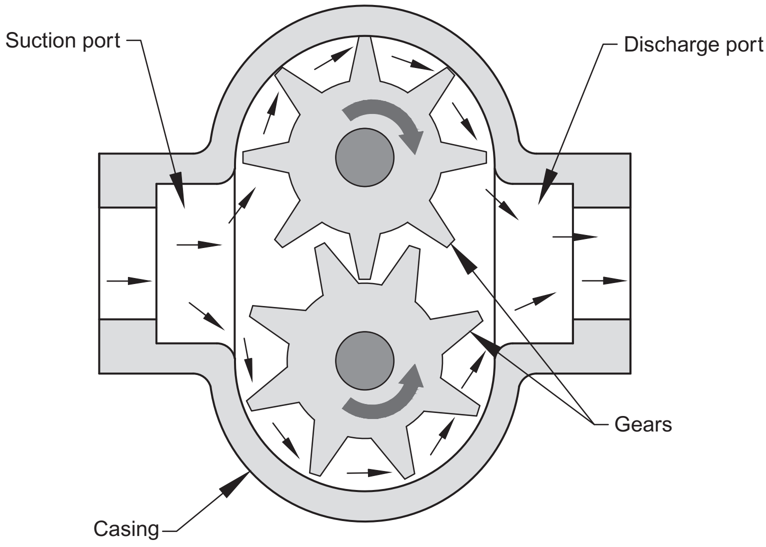
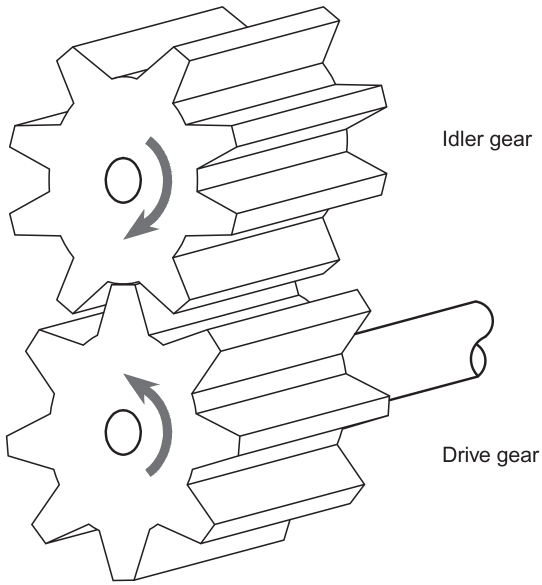


Fig. 5.6 Schematic diagram of an external gear rotary pump.

Fig. 5.7 Schematic showing the driver and idler gears on an external gear rotary pump.



- *Spur gears* are used on low-capacity, high-pressure applications since they offer a good contact between the teeth. This close tolerance reduces slip and increases the capacity. They tend to be noisy and deflect.
- *Helical gears* eliminate shaft deflection problems and are less expensive but require thrust bearings.
- *Herring bone* gears are not noisy, do not deflect, do not produce thrusts, thus there is less chance of slipping. However, they are the most expensive.

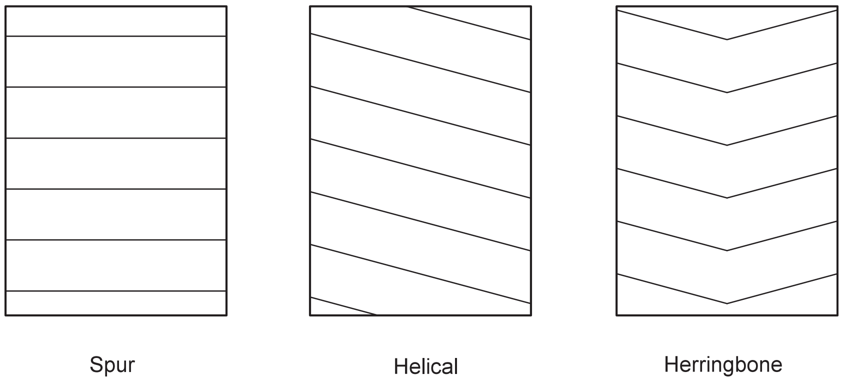


Fig. 5.8 Diagram showing three commonly used gears in gear pumps; Spur, Helical, and Herringbone.

External gear pumps:

- Are compact in size
- Can produce high pressures
- Well suited for highly viscous fluids
- Easily manufactured in a broad range of materials so as to ensure compatibility with the pumped fluids
- Well suited for use with shear-sensitive materials

Due to the close tolerances external gear pumps should be used with clean fluids.

5.2.2.2 Internal gear

The internal gear pump (Fig. 5.9) is similar in principle to the external gear except the drive shaft turns a ring gear with internal teeth. The outer gear of this pump rotates concentrically in the casing and is coupled to the driver. The inner gear is eccentrically mounted and is an idler. The crescent is stationary and seals the tooth cavities between suction and discharge. Internal gear pumps are used only in clear services, but material options are available that make them more suitable for services with some suspended solids than internal bearing, external gear, and screw pumps.

The internal gear pump is a reliable, inexpensive pump. The main disadvantage involves excessive backpressure or handling fluids that contain solids. Small gear pumps are limited to a backpressure of 100 psi (689 kPa) and require a pressure relief valve on the discharge side. Since small clearances exist, the manufacturer should always be consulted before any gear pump is used with fluids handling solids.

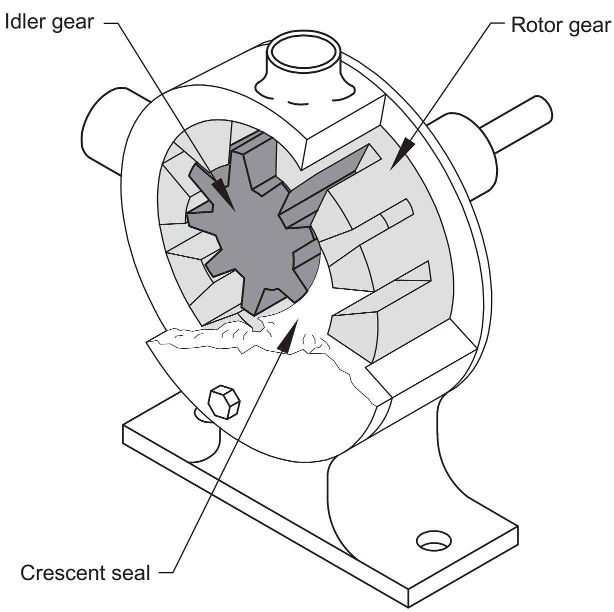


Fig. 5.9 Diagram of an internal gear rotary pump.

5.2.2.3 Lobe

Lobe rotary pumps (Fig. 5.10) operate in the same manner as the gear pumps except the rotating elements have two, three, or four lobes instead of gear teeth (Fig. 5.11). Lobes cannot drive each other so timing gears are used. Lobes never come into contact with each other, so the pump can be allowed to run dry. The large volume of voids created between the casing and lobes allows many products to be pumped without damaging the product itself.

Lobe pumps are used where product integrity must be maintained (food processing industry) or in chemical processing applications where compounds are shear sensitive.

Lobe pumps have no metal-to-metal contact between the lobes, thus the possibility of traces of iron, steel, or other pump construction materials ending up in the product due to wear is greatly reduced. However, they are more expensive than gear or vane pumps, difficult to repair and maintain, require timing gears, thus increases

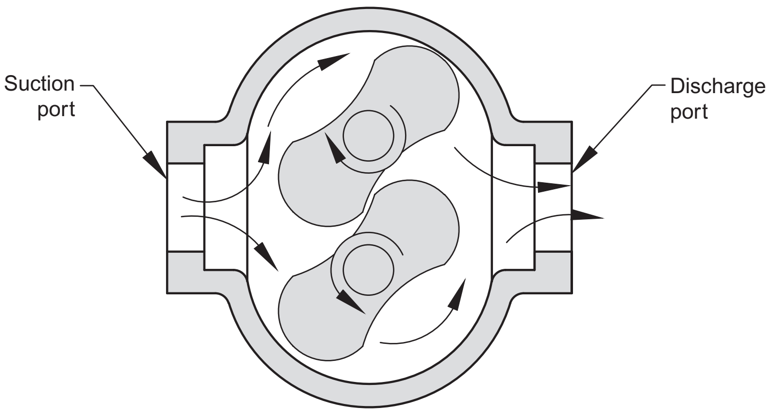


Fig. 5.10 Schematic of a lobe rotary pump.

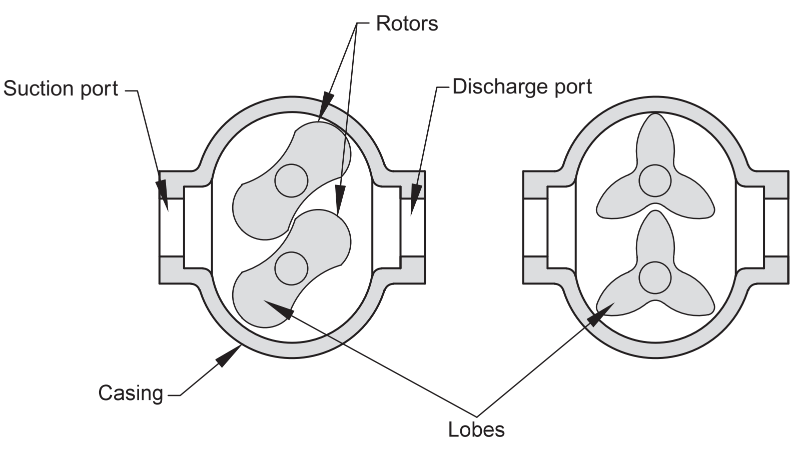


Fig. 5.11 Schematic illustrating two- and three-lobe rotary pumps.

complexity and cost. There is the possibility of product contamination if timing gears are run in an oil bath.

5.2.3 Screw pumps

Screw pumps are by far the most commonly used rotary pump used in the oil and gas industry, Fig. 5.12 shows three types of screw pumps: single-screw, two-screw, and three-screw. Screw pumps can be single (progressive cavity) or multiple-rotor (intermeshing) design. They are relatively high-speed pumps, but because of the reversal of flow required to enter the suction passage, NPSH can often be a problem.

5.2.3.1 Single-screw

The single-screw (Fig. 5.13), also referred to as a "progressive cavity" pump, consists of a single screw rotating in a rubber or elastomer stator. The single-screw is a slow-speed pump and it is physically large for the amount of fluid being pumped. Fluid is trapped between the threads of a rotating screw and the threads of the internal stationary element.

Single-screw pump can be used in viscous fluids and fluids with high solids content. It can tolerate small amounts of solids and dissolved gas in the pumped fluid and is often chosen for that reason. They can produce significant suction lift and relatively high pressures. They provide uniform discharge pressure with little pulsation and can handle fluids ranging from clean water to sludge without changing clearances or components.

Single-screw pumps are expensive, large, bulky, difficult to maintain, and replacement parts are costly.

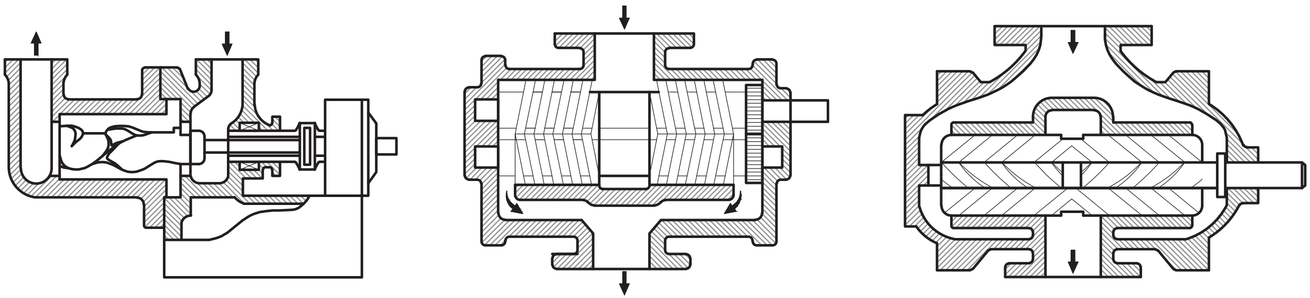
5.2.3.2 Two-screw

The two-screw pump (Fig. 5.14) has two screws that rotate within the stator or casing. The two screws are designed with greater clearance, so they rarely contact the stator. External timing gears prevent tooth contact between screws. The screws are supported by antifriction bearings (ball bearings) for precise alignment of the rotors. The bearings can be mounted internally or externally. In the external bearing version, the timing gears and bearings are mounted in external oil-lubricated housings. The shafts protrude through the casing in four locations, making four mechanical seals (or packing boxes).

The advantage of the external-bearing two-screw over the three-screw or internal bearing two-screw pump is that it is less susceptible to wear in services with suspended solids; the tradeoff is higher cost. The external bearing can be used in services with lower viscosities and lubricating capability. Neither the internal-bearing two- or three-screw pump should be used in services with suspended solids.

5.2.3.3 Three-screw

The three-screw pump (Fig. 5.12) has three screws that rotate within the stator or casing. The casing supports the rotors along their entire length and functions as a journal bearing. The rotors, stator, and mating teeth between the rotors are all lubricated by the process fluid. Thus the three-screw is referred to as the "internal-bearing" version of



Single screw pump Double screw pump Three screw pump

Fig. 5.12 Types of screw pumps: single-screw, two-screw, and three-screw.

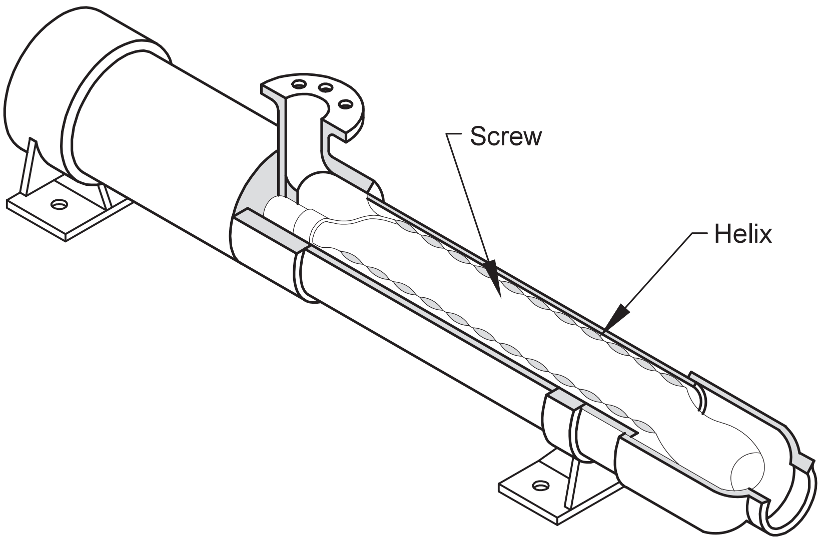


Fig. 5.13 Single-screw (progressive cavity) rotary pump.

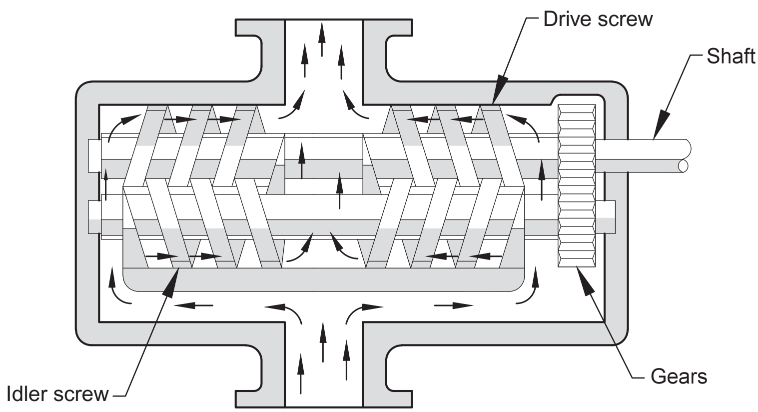


Fig. 5.14 Two-screw rotary pump.

the screw pump. The center screw, or power rotor, is coupled to the driver and drives the other two screws, called idlers. A major advantage of the three-screw pump is that it requires only one mechanical seal.

In small sizes, they are used to supply lubricating oil to engines and machinery. In large sizes, they are used to load and unload barge and tankers.

5.3 Rotary pump selection guidelines

Table 5.1 and the guidelines later can be used to help the designer select the appropriate type of rotary pump for a specific service.

Table 5.1 Pump application chart for rotary pumps

Pump description	Two-screw w/ext. three-screw bearing		Single-screw	Ext. gear w/lnt. brngs.	Ext. gear w/ext. brngs.	Internal gear
Diff. pressure range (PSI)	3000	1500	200	70	700	200
Flow range (GPM)	1000	5000	450	600	600	150
Max. temp. (F)	500	500	180	250	250	250
Horsepower range (HP)	800	2000	800	400	400	40
Speed range (RPM)	3600	1800	800	1800	1800	1800
Viscosity range (SSU)	1,000,000	1,000,000	1,000,000	250,000	250,000	250,000
Variable viscosity	N	Y	M	N	M	N
Relative cost to (#) purchase	3	5	4	2	3	1
Relative cost to install (#)	3	4	2	2	3	1
Relative cost to maintain (#)	4	3	5	2	3	1
Self-priming	Y	Y	M	Y	Y	Y
Can run dry—short time	N	M	N	N	M	N
Will emulsify	M	N	N	N	N	M
Field alignment required	Y	Y	Y	Y	Y	Y
Good for some entrained gas	Y	M	Y	Y	Y	M
Good for abrasives	N	N	Y	N	N	M
Parallel or series recommended	P	P	P	P	P	P
Bearings lub. (oil, grease, stock)	S	O	S, G	S	O	S
Coupling rigid or flexible	F	F	F	F	F	F

Y, yes; *N*, no; *M*, marginal; #, relative number (1–5, 1 is best); *N/A*, not applicable.
(1) Pumps are commercially available outside the parameters shown. Pumps with values outside the parameters should be avoided or special care should be taken to maintain their reliability.
(2) Rotary pumps are, by nature, self-priming. However, they require liquid in the pump in order to lubricate internal parts.

5.3.1 Eccentric rotor

5.3.1.1 Vane

Inexpensive, but wears excessively on all but the cleanest lubricating stocks. Seldom used. May be used in services where the pump may need to handle entrained or dissolved gas. They may also be applied in services with moderate vacuum suction conditions. Discharge pressures to 200 psig (1379 kPa).

5.3.2 Gear

5.3.2.1 External gear

The "multipurpose" rotary pump. Used in clean lubricating services. Sometimes used as an expendable pump in nonlubricating services because of lower cost. Discharge pressures to 700 psig (4.826 kPa) and capacity to 600 gpm (138 m^3/hr).

5.3.2.2 Internal gear

Usually less expensive than external gear pumps, and usually the first choice in a gear pump. Occasionally used in nonlubricating or slightly solid-bearing fluid services. Discharge pressure to 200 psig (1379 kPa) and capacity to 150 gpm (34.5 m^3/hr). For filtered liquids and power fluid services a special design exists for pressures up to 1000 psig (6895 kPa). Smaller sizes can be directly coupled to driver, but usually require a gear or belt drive.

5.3.2.3 Lobe

Not common. May be found in services that might emulsify from liquid shearing. In such service the pump would be run at a very slow speed to prevent agitation.

5.3.3 Screw

Often the only type available for higher pressures or larger capacities. Usually more expensive but will generally have a longer pump life than gear pumps.

5.3.3.1 Single-screw

Seldom the appropriate choice for any application because of low reliability. Occasionally applied to fluids with entrained solids, water treating, and oil sludge handling services. In such a service the pump would be run at low speed and would have a rubber stator to minimize erosion. May find one applied to a non-Newtonian or other shear-sensitive fluid. Discharge pressure to 200 psig (1379 kPa) and capacity to 450 gpm (104 m^3/hr).

5.3.3.2 Two-screw

Usually more expensive than a three-screw. Available in capacities exceeding the range of three-screw pumps. Often applied in fuel oil, crude oil, and asphalt transfer services, especially at marine and rail terminals. Discharge pressures up to 1500 psig (103 kPa) and capacity to 5000 gpm (1150 m^3/hr). Can handle lower viscosities than three-screw types.

5.3.3.3 Three-screw

The most common screw pump type and usually the first choice in screw pumps. Considered the standard pump for lube-oil and seal-oil circulation for major machinery trains, barge unloading, and pipeline services. Also the standard pump for fuel oil delivery to burners. Discharge pressures to 3000 psig (206 kPa) and capacities to 1000 gpm (230 m^3/hr). Can be directly coupled to the driver.

5.4 Rotary pump service considerations

The following discussion can be used to help select the appropriate rotary pump when handling specific services.

- *Clean services*: All rotary pumps will perform well with clean fluid that lubricates (i.e., viscosity greater than 2 centistokes (35 SSU)). In general, screw pumps can be directly coupled to the driver and are more efficient at high pressures, high flow rates, and high viscosities than gear pumps. They are also more expensive. Gear pumps are competitive in the lower flow, lower pressure applications and usually require gear or belt drive arrangements.
- *Dirty service*: The single-screw should be the only pump considered for handling fluid with significant solids such as abrasive slurries, sewage, or sludge. An alternative for intermittent low-pressure situations is use of small, inexpensive internal gear pump which can be replaced as needed. The use of external timing gears in two-screw pumps helps maintain clearances between screws and case, so they can tolerate fine, low-hardness contaminants. Another alternative in low flow services might be an air-driven diaphragm pump.
- *Lubrication and hydraulic systems*: Normally, three-screw pumps are used in this service. With clean, lubricating fluids their continuous operation is reliable and extended. Three-screw pumps perform best at high pressures with clean, fixed viscosity fluid, where external bearings are not required.
- *High pressures*: Screw pumps are better suited than gear pumps in services exceeding 400 to 500 psig (2758–3447 kPa). They have a higher mechanical efficiency at higher pressure because of greater volumetric efficiency. However, at low pressures their mechanical efficiency is lower than gear pumps because of the greater friction losses due to larger internal surface areas. Higher pressures are possible with three-screw pumps than two-screw pumps.
- *High viscosity*: Two-screw pumps are more versatile than gear or three-screw in handling fluids with variable viscosity. The most common two-screw pumps use external timing gears and bearings, which make them more expensive than three-screw pumps.
- *Self-priming*: All rotary pumps can operate with considerable dissolved or entrained gas in fluid pumped without losing suction. Continuous handling of appreciable quantities of vapor along with liquid, especially under cavitating conditions, can cause excessive noise and

vibration and contribute to rapid wear. Typically, screw pumps provide greater suction lift than gear pumps, and three-screw pumps are better than two-screw pumps.

- *Vacuum service*: Rotary pumps lubricated by special oils are often used in vacuum service to pump air or other gases or vapors. Low-vapor pressure oils are used to lubricate the pumps and seal the clearance spaces. Liquid-ring vacuum pumps are also available.
- *Nonpulsating flow*: Typically, screw pumps provide smoother fluid discharge than gear pumps. Three-screw pumps are better than two-screw pumps.
- *Drive arrangement*: Typically, gear pumps and single-screw pumps operate at slow speeds which require belt or gear drives. Three-screw and two-screw pumps almost always operate at motor speed, eliminating the need for a belt or gear drive.
- *Emulsifying affect*: All types are less likely than centrifugal to promote stock emulsification. The lower the speed, the better.
- *Running dry*: These pumps are unforgiving if run dry; they fail quickly. Twin screw and twin gear, with external gears/bearings, can tolerate running dry slightly better than others.

5.5 Rotary pump configurations

This section describes five commonly used rotary pumps.

5.5.1 External gear rotary pump (Fig. 5.15)

Typical service:	Low-pressure, low-capacity lubricating fluids Pipelines—wet crude or fluids Lubrication and hydraulic oil systems Air, gas, or vapor vacuum service
Pressure range:	0–700 psig (0–213 m)
Capacity range:	0–600 gpm (0–138 m^3/hr)
Speed range:	300–1800 rpm
Max allowable temperature:	250°F (122°C)
Standard materials:	Can be external or internal bearing Carbon steel case for hydrocarbon service Steel, bronze, cast iron gears Can be packed or mechanical seals Close internal clearances (0.005–0.015 inch)
Specification:	API 676
Typical control method:	Variable speed or flow bypass
Advantages:	Self-priming Constant delivery at high efficiency over a wide range of pressures Minimizes fluid emulsification
Limitations:	Cannot handle external forces from piping Cannot run dry Pumped fluid must provide lubrication Low tolerance for abrasives Can have high maintenance cost Usually has four stuffing boxes

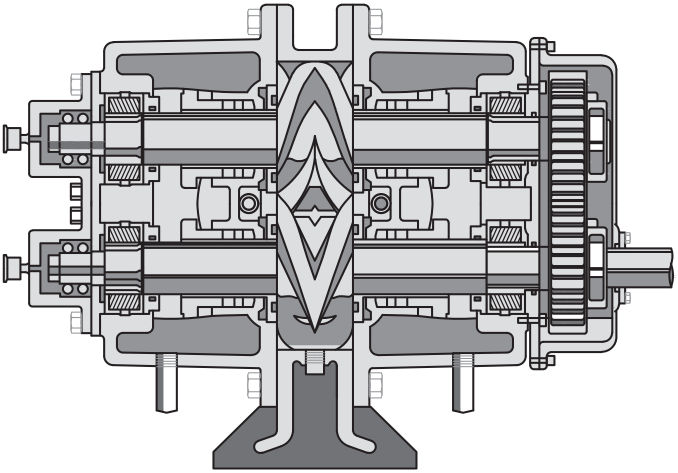


Fig. 5.15 External gear rotary pump.
Courtesy of Kinney Vacuum Company, Inc.

5.5.2 Internal gear rotary pump (Fig. 5.16)

Typical service:	High-pressure, low-capacity lubricating fluids Pipelines—wet crude or fluids. Lubrication and hydraulic oil systems Air, gas, or vapor vacuum service
Pressure range:	0–200 psig (0–1379 kPa)
Capacity range:	0–150 gpm (0–35 m^3/hr)
Speed range:	300–1800 rpm
Max allowable temperature:	250°F (122°C)
Standard materials:	Can be external or internal bearing Carbon steel case for hydrocarbon service Steel, bronze, cast iron gears Can be packed or mechanical seals Close internal clearances (0.005–0.015 inch)
Specification:	API 676
Typical control method:	Variable speed or flow bypass
Advantages:	Self-priming Constant delivery at high efficiency over a wide range of pressures Minimizes fluid emulsification Handles viscous stocks and two-phase flow Material options available for suitability in service with some suspended solids
Limitations:	Have no timing gears More complex machining required than with external gear pumps

Fig. 5.16 Internal gear rotary pump.
Courtesy of Viking Pump.

5.5.3 Single-screw (progressive cavity) rotary pump (Fig. 5.17)

Typical service:	Viscous crude or fluid with suspended solids or abrasives Sludge Services with wide variations in viscosity
Pressure range:	0–200 psig (0–1379 kPa)
Capacity range:	0–450 gpm (0–104 m^3/hr)
Speed range:	0–800 rpm
Max allowable temperature:	180°F (82°C)
Standard materials:	Steel rotor in cast iron Soft Teflon, Buna N, or cast iron stator Steel in hydrocarbon service
Specification:	API 676
Typical control method:	Variable speed or flow bypass
Advantages:	Self-priming Will pass solids and vapor Usually less expensive than two-screw pumps Can remove rotor with pump in the line Uniform, nonpulsating discharge Will not emulsify fluid Less expensive than reciprocating pumps Can handle non-Newtonian or shear-sensitive fluids Can be mounted horizontally or vertically
Limitations:	Less reliable than other positive displacement pumps Cannot run dry Uses large amount of floor space for amount of fluid pumped Expensive to maintain and unreliable Stator materials must be inert to the fluid in order to prevent seizure.

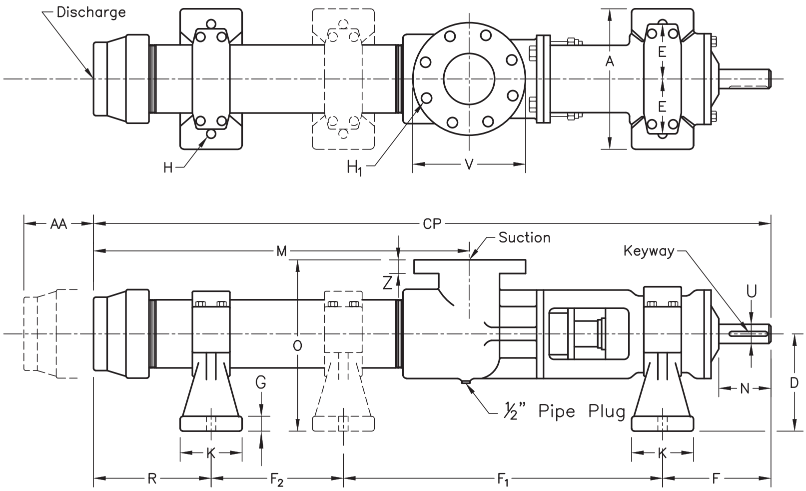


Fig. 5.17 Single-screw rotary pump.
Courtesy of Moyno Industrial Pumps, a Division of Robbins and Myers.

5.5.4 Two-screw rotary pump (Fig. 5.18)

Typical service:	Heavy fuel oil Pipeline Crude oil Unloading Lubricating fluids to 225,000 centistokes (1,000,000 SSU)
Pressure range:	0–1500 psig (0–10,342 kPa)
Capacity range:	0–5000 gpm (0–1150 m^3/hr)
Speed range:	200–1800 rpm
Max allowable temperature:	500°F (260°C)
Standard materials:	Driven rotor meshes with idler rotor and both are driven by timing gears Generally double end to balance thrust in larger pumps Four stuffing boxes or mechanical seals Steel case in hydrocarbon service Cast iron or steel rotors Externally lubricated timing gears and bearings Close internal clearances (0.005–0.015 inch)
Specification:	API 676
Typical control method:	Variable speed or flow bypass
Advantages:	Quieter, more efficient, smoother flow than gear pumps Self-priming Nonpulsating flow Will not emulsify Will pass vapors Constant capacity over wide range of pressures with high efficiency

Limitations:	More expensive than single-screw, three-screw, or gear pumps Poor with nonlubricating or corrosive fluids or abrasives Cannot run dry Lower tolerance for abrasives than reciprocating pumps Lower mechanical efficiency than gear pumps at low pressures

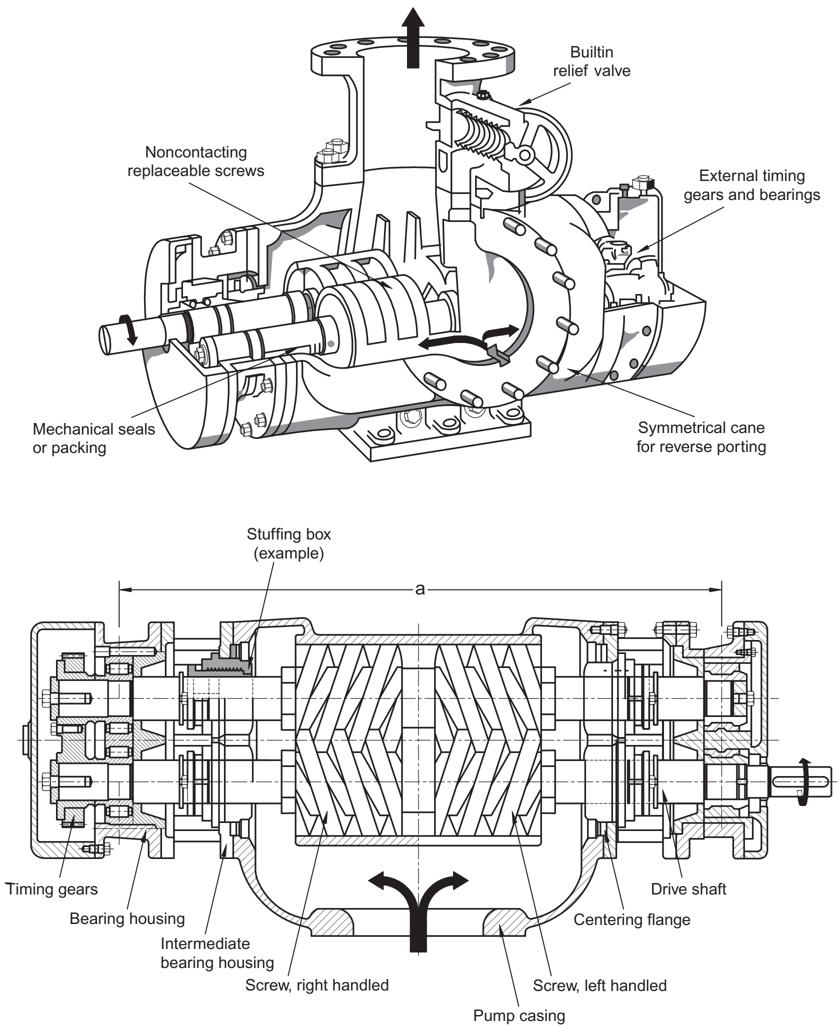


Fig. 5.18 Two-screw rotary pump.
Courtesy of IMO Industries: Pump Division.

5.5.5 Three-screw rotary pump (Fig. 5.19)

Typical service:	Heavy fuel oil Pipeline Crude oil Unloading Lubricating fluids to 225,000 centistokes (1,000,000 SSU)
Pressure range:	50–3000 psig (345–20,684 kPa)
Capacity range:	0–1000 gpm (0–230 m^3/hr)
Speed range:	200–3600 rpm
Max allowable temperature:	500°F (260°C)
Standard materials:	Driven rotor drives 2 idler drives Generally double end to balance thrust in larger pumps Steel case in hydrocarbon service Cast iron or steel rotors One bearing needed to center rotors One stuffing box or head seal Close internal clearances (0.005–0.015 inch)
Specification:	API 676
Typical control method:	Variable speed or flow bypass
Advantages:	Quieter, more efficient, smoother flow than gear pumps at higher pressures Self-priming Nonpulsating flow Will not emulsify Will pass vapors Fewer seals and bearings than two-screw pumps Constant capacity over wide range of pressures with high efficiency Can be mounted vertically or horizontally
Limitations:	More expensive than most gear pumps Poor with nonlubricating or corrosive fluids or abrasives Cannot run dry Lower tolerance for abrasives than reciprocating pumps

5.6 Installation considerations

5.6.1 Drivers

Rotary pumps are normally driven by electric motors, geared motors, or belt driven with motors. Sometimes a rotary pump is driven by a steam turbine and in very rare cases by an internal combustion engine.

5.6.2 Instrumentation and control considerations

Capacity control is obtained either by varying the speed of the pump or by recirculating a portion of the discharge. Recirculation is a common method of control, as rotary pumps are often operated at constant speed with electric motor drivers.

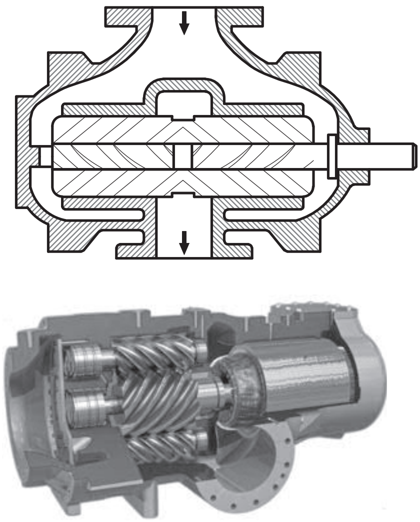


Fig. 5.19 Three-screw rotary pump.
Courtesy of IMO Industries: Pump Division.

5.6.3 Strainers

Permanent suction strainers should be installed to protect rotary pumps from damage by foreign materials. This requirement applies to most rotary pump installations, except for inexpensive pumps in relatively clean systems where the initial cost does not justify a strainer to protect the pump. A dirty or plugged strainer may also cause pump damage by increasing the friction loss in the suction piping and reducing the $NPSH_A$. A pressure safety low (PSL) downstream of the strainer may alleviate this problem.

Even though strainers are provided, the suction lines to rotary pumps should be flushed out before final connection of piping to the pumps.

5.6.4 PSVs

Pressure relief valves (PSVs) are required in the discharge or rotary pumps to prevent overpressure. Blocking in a rotary pump will cause failure of the weakest component in the drive train, usually a cracked pump casing. Many rotary pumps can be obtained with internal "built-in" PSVs. API 676 prohibits use of internal PSVs, because they are usually unable to protect the pump from overheating in a relief situation. Separate PSVs are preferred, piped back to the vessel in the inlet system.

5.6.5 Lubrication prior to start-up

Because of the possible or unnecessary internal contact, rotary pumps must be lubricated before starting. When they are installed to handle appreciable quantities of vapor even for a short time, as in priming, special precautions may be required to ensure proper lubrication before start-up. In self-priming installations for low-viscosity fluids, rotary pumps should ordinarily be installed at the bottom of a "U" in the piping that can be filled with liquid to provide lubrication during start-up of if the pump loses suction. In intermittent services, foot valves are also recommended for low-viscosity fluids.

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Special purpose pumps

6

6.1 Artificial lift pumps

Throughout the life of a producing well, the reservoir pressure may become insufficient to flow naturally through the surface production facility. In these cases artificial lift pumps are commonly used to maintain production. The four major methods used for artificially lifting the well fluid include:

- Electric submersible pumping (ESP)
- Sucker rod pumping
- Hydraulic turbine pumping
- Gas lift

Each method has advantages and limitations when compared to each other. Fig. 6.1 shows the capability of each lift method under different operating conditions. Selection of any method must be based on the well and reservoir characteristics. The most important selection criterion for an artificial lift system is to find the system that will deplete the well economically.

The following characteristics should be considered when choosing an artificial lift system.

- Name and location of well and its proximity to other wells
- Reservoir data
- Bottom-hole pressure and productivity index
- Oil and water production
- Gas/oil ratio
- Depth of lift required
- Casing size, condition, and single or multiple completion
- Type and condition of existing lift equipment, if applicable
- Operating problems, such as scale, paraffin, corrosion, and so on
- Availability and cost of gas and/or electricity
- Degree of automation desired
- Ability of the system to meet changing production conditions, if required
- Safety
- Investment and operating costs

6.1.1 Electric submersible pumps (ESPs)

There are two types of ESPs: oilfield ESPs and shallow water well pump.

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00006-5

Operating condition	Sucker rod pumps	Hydraulic pumps	Electric pumps	Gas lift
Scale	Good	Fair	Poor	Fair
Sand	Fair	Fair	Poor	Excellent
High volume	Poor	Good	Excellent	Excellent
Flexibility	Good	Excellent	Poor	Good
Wax	Fair	Good	Good	Poor
Corrosion	Good	Good	Fair	Good
High entrained gas	Fair	Fair	Fair	Excellent
Deviated well	Poor	Good	Fair	Excellent
Depth	Fair	Good	Good	Good
Simple design	Yes	No	No	No

Fig. 6.1 Capability of each lift method under specific operating conditions.

6.1.1.1 Down-hole oilfield ESP

The down-hole ESP (Fig. 6.2) is very common in oil field production operations. Typical manufacturers are TRW/REDA, Cenrilift-Hughes, and Oil Dynamics, Inc. ESPs consist of the following four major sections:

- Submersible motor and cable
- Seal section
- Pump (sometimes supplied with a gas separator)
- Topside transformer and electrical controls and instrumentation

The ESPs run at 3600 rpm; each motor/transformer set is matched. The setting depth, cable selection, and power rating of the motor determine the required topside voltage and transformer rating. Thus, every installation is unique. The setting depth can be as much as 13,000 ft (3962 m) and produce flows between 200 and 20,000 bpd. Motors are usually less than 200 hp (149 kW).

The pump, seal section, motor, and tubing are designed to fit inside standard casing sizes. Motors and pumps may be 5-1/2-in. or 7-in. diameter. Due to the restriction in diameter, motors may be 30 or 400-ft (9 to 122 m) long and pumps have up to 400 stages. This also restricts the allowable diameter of the shaft, typically about 1 inch (2.54 cm). The reliability of these pumps decreases as well temperature, deviation, and horsepower increase. The average run life is between one and two years.

The *advantages* of an ESP system are as follows:

- Economically produce high volumes of fluid
- Work well in locations with minimal surface area, such as offshore platforms
- Low initial cost
- Use a single tubing string and vent gas through the annulus

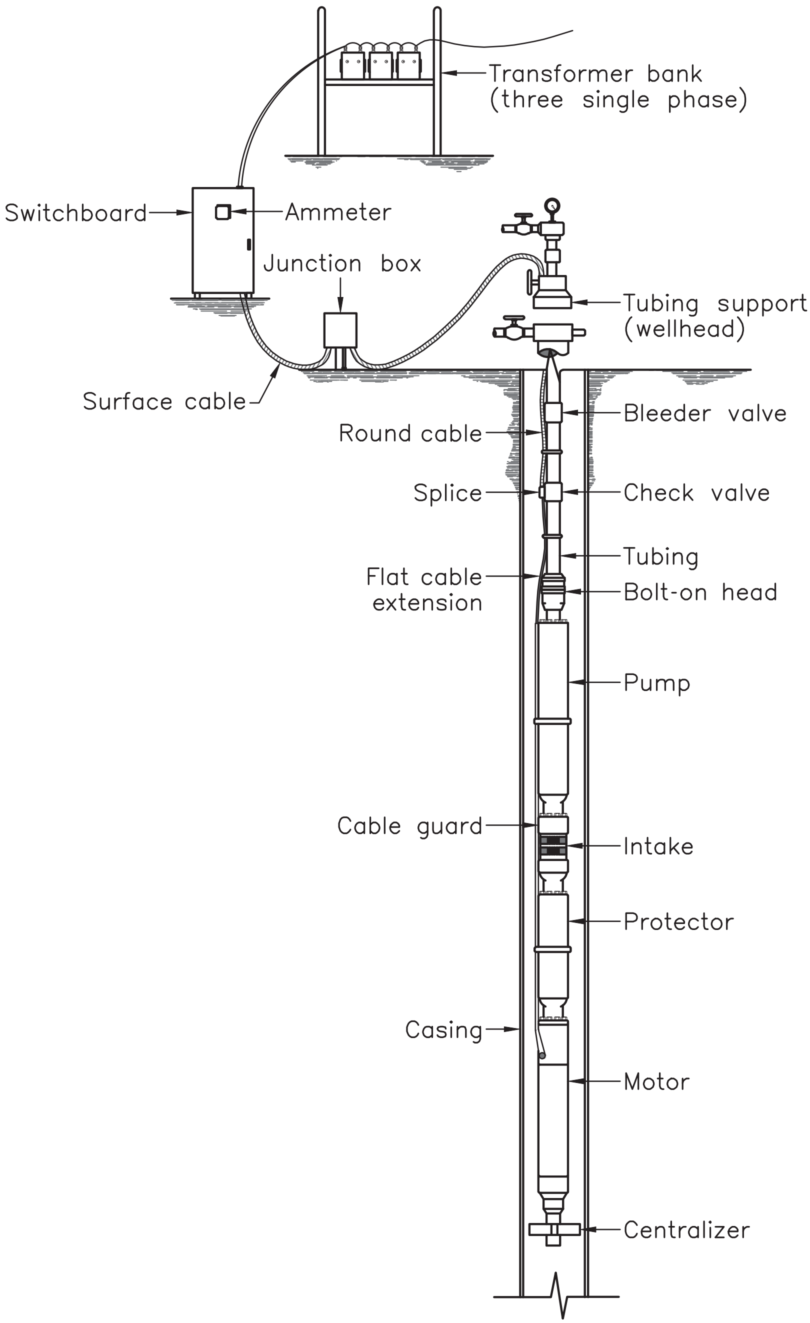


Fig. 6.2 Electrical submersible pump system.
Courtesy of Oil and Gas Journal.

The *disadvantages* of an ESP system are as follows:

- Poor flexibility in lifting unexpected, rapidly changing, or low-volume production
- The electric power supply cable is affected by depth, corrosion, temperature, or handling, which can cause cable failures
- For wells with gas in addition to liquid, the ESP must have good gas separation for efficient pumping
- Abrasives, wax, or scale decrease the ESP run life. (This is true for all down-hole pumps)
- Relatively poor reliability
- The inherent low efficiency of a centrifugal pump makes the overall efficiency of ESPs poor
- Pulling and repair costs are high

For further information on ESPs, refer to one of the following:

- API RP-11R Recommended Practice for Electric Submersible Pump Installations
- API RP 115 Recommended Practice for Operation, Maintenance and Trouble-Shooting Electrical Submersible Pump Installations
- Handbook for Oilfield Subsurface Electrically Driven Pumps. (Courtesy of Centrilift-Hughes, Inc.)

6.1.1.2 Shallow water well pumps

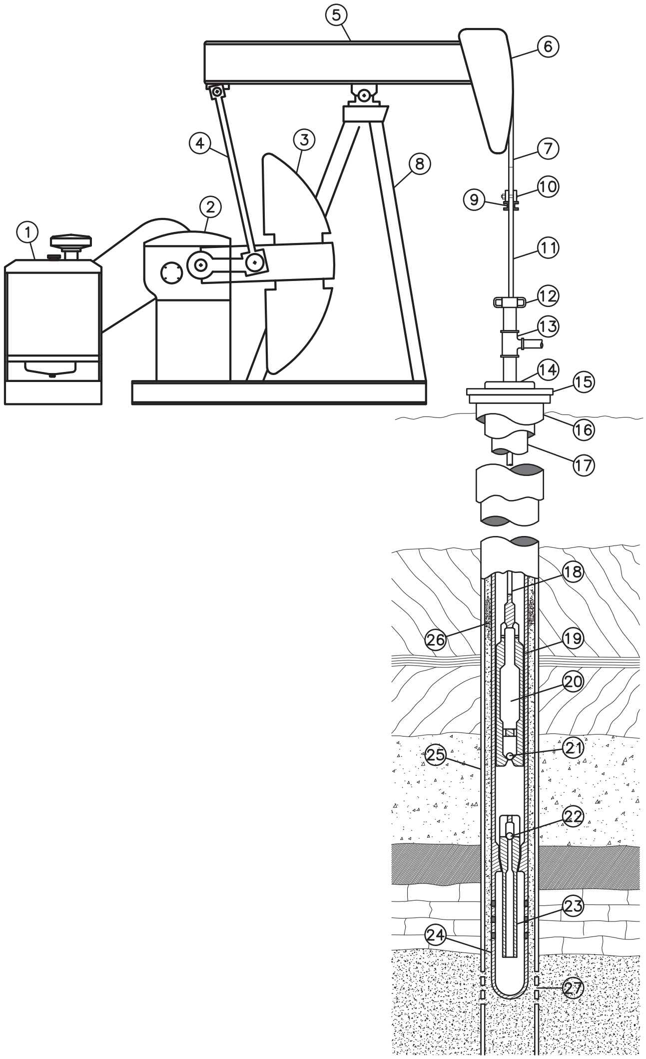
Shallow water well pumps are not specifically designed for deep well installations and never used in oil production. However, they are used in seawater lift services and other shallow, cool water services. Shallow water well pumps consist of a submersible motor and pump. The seal is built into the motor; the pump does not require one. The electrical support systems are standard and do not require special voltages or dedicated transformers. The pump and motor are packaged by the pump manufacturer. Typical manufacturers of this type of submersible motor are TRW/Plueger, Hayward-Tyler, and Byron-Jackson. The motors are larger in diameter, up to 14 inches and do not have the pressure balancing capabilities of the down-hole ESP. The motors are much more shorter in length, usually about 10 ft and are capable of up to 700 hp. These motors normally operate at 1800 rpm.

These motors can be installed onto almost anyone's standard vertical turbine pump. Typical manufacturers include Goulds, Peerless, Ingersoll-Rand, Bingham, and Dresser-Worthington.

The shallow well WSP is usually more expensive than the deep well ESP, with little overlapping coverage between them.

6.1.2 Sucker rod pump

Sucker rod (walking beam) pumps are normally limited to down-hole applications in certain producing areas. These systems (Fig. 6.3) are designed for lower producing rates (up to 1000 bpd) from shallow to moderate well depths (up to 11,500 ft) (3505 m). In a sucker rod pump system, a surface pumping unit converts the rotary motion of a prime mover (gas engine, electric motor, etc.) into a reciprocating action. The reciprocating action is transferred to a positive displacement pump located down-hole.



Sucker rod pumpin system: (1) Prime mover, (2) Gear reducer, (3) Crank and counterweight, (4) Pitman, (5) Walking beam, (6) Horsehead, (7) Bridle, (8) Samson post, (9) Carrier bar, (10) Polished rod clamp, (11) Polished rod, (12) Stuffing box, (13) Pumping tee, (14)Tubing ring, (15) Casing head, (16) Casing surface string, (17) Tubing string, (18) Sucker rod, (19) Pump barrel, (20) Pump plunger, (21) Traveling valve, (22) Standing valve, (23) Mosquito bill, (24) Gas anchor, (25) Casing oil string, (26) Fluid level, (27) Casing perforations. (National supply)

Fig. 6.3 Typical sucker rod "walking beam" pump.
Courtesy of McGraw Hill, Pump Handbook, Karassik, Krutz, Fraser and Messina.

The advantages of sucker rod lift systems are as follows:

- They are the oldest and most widely used means of artificial lift, accounting for 85% of artificial lift in the United States. Consequently, sucker rod pumps are the best understood artificial lift method.

- Low initial costs in shallow to moderate depth wells
- Flexibility to handle changing production volumes
- They use a simple tubing string and vent natural gas through the annulus

The *disadvantages* of sucker rod pumps are as follows:

- Increasing the depth and/or the produced volumes increases the system's cost and reduces the system's production capabilities
- Require a pulling unit for running and retrieving the down-hole pump or replacing sucker rods
- Volumetric efficiency is reduced in wells with high GOR, solids, wax, H_2S, or corrosion

The following references provide detailed information on sucker rod pumps and their corresponding pumping units:

- API RP-11L Recommended Practice for Design, Calculations for sucker rod pumping systems (Field Units)
- API BUL 11L4 Bulletin containing curves for selecting beam pumping units
- API 11AR Recommended Practice for care and handling of subsurface pumps
- API RP 11BR Recommended Practice for care and handling sucker rods
- API RP SPEC 1B Specification for Oil Field V-Belting (includes a design procedure for power application of V-Belts)
- API RP 11G Recommended Practice for installation and Lubrication of Pumping Units
- API SPEC 11AX Specification for Subsurface Sucker Rod Pumps and Fittings
- API SPEC 11B Specification for Sucker Rods (pony rods, polished rods, couplings, and subcouplings)
- API STD 11E Specification for Pumping Units
- Artificial Lift Sucker Rod Pumping. Royalty Enterprises, Inc. Garland, Texas, USA

6.1.3 Hydraulic turbine driven pumps

Hydraulic turbine driven pumps (Fig. 6.4) can handle up to 12,000 bpd production from shallow to deep well depths (up to 17,000 ft) (5182 m). In hydraulic turbine driven pump systems, a down-hole hydraulic turbine receives its energy by means of a fluid, usually oil or treated water. The power fluid is pumped down-hole by means of a surface pump.

The *advantages* of this lift system are as follows:

- Economics for hydraulic pumps improve with depth and produced volume requirements
- Economical and flexible range in handling changing produced volume requirements
- The down-hole pump can be run in and out of the well by circulation
- Work in deviated, directional, or inaccessible well locations
- Power fluid (either produced oil or water) can be a treated fluid to provide heat, chemical, or diluent
- Hydraulic pumps work well in pumping crudes because the crudes are diluted by the lighter power fluid
- When using hydraulic powered pumps, a centrally located pump station can serve a number of wells

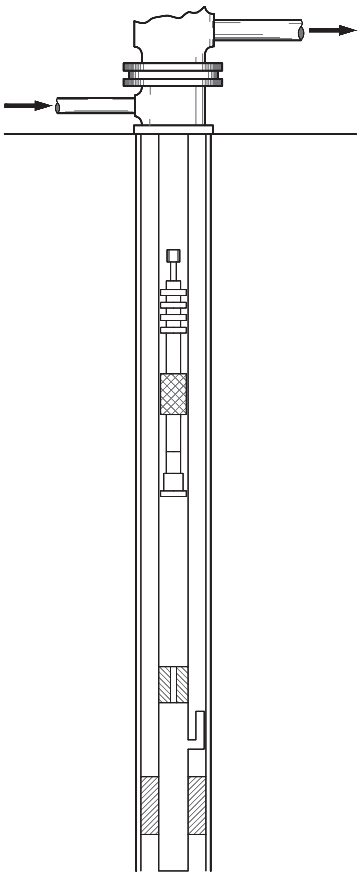


Fig. 6.4 Hydraulic pump.
Courtesy of PennWell Publishing, Dictionary of Petroleum Exploration, Drilling and Production by Norman J Hyne.

The *disadvantages* of hydraulic pumps are as follows:

- Cannot vent gas with a single tubing string
- When used with central station for multiwell system:
  - Power fluid requires treatment
  - Long high pressure lines for power fluid are required
  - Production testing is more difficult
  - Initial capital costs are high
- Initial capital costs are high
- When oil is used, the power fluid volume required may become very expensive
- High maintenance cost
- Sonic bottom-hole pressure testing cannot be performed on subsurface completion types. A pressure bomb cannot be run with a reciprocating-type hydraulic pump. The jet-type hydraulic pump allows pressure bombs to be run in the chamber at the bottom of the pump

- Power requirements are high due to the combined inefficiencies of power fluid pumps, down-hole pumps, and hydraulic turbines

6.1.4 Gas lift systems

Gas lifting (Fig. 6.5) can lift high volumes of fluid (up to 20,000 bpd on continuous injection) from moderate to deep well depths (up to 14,000 ft) (4267 m). In gas lifting, the reservoir fluids are lifted from the wellbore by injection (intermittent or continuous) of a high-pressure gas to supplement the reservoir's energy.

The *advantages* of gas lifting are as follows:

- Gas lifting is very economical where gas is readily available at high pressures
- Gas lifting handles changing volumes with economic flexibility
- Gas lifting works in low productivity wells with high GORs
- When using gas lift, abrasive material (sand, etc.) offers fewer problems

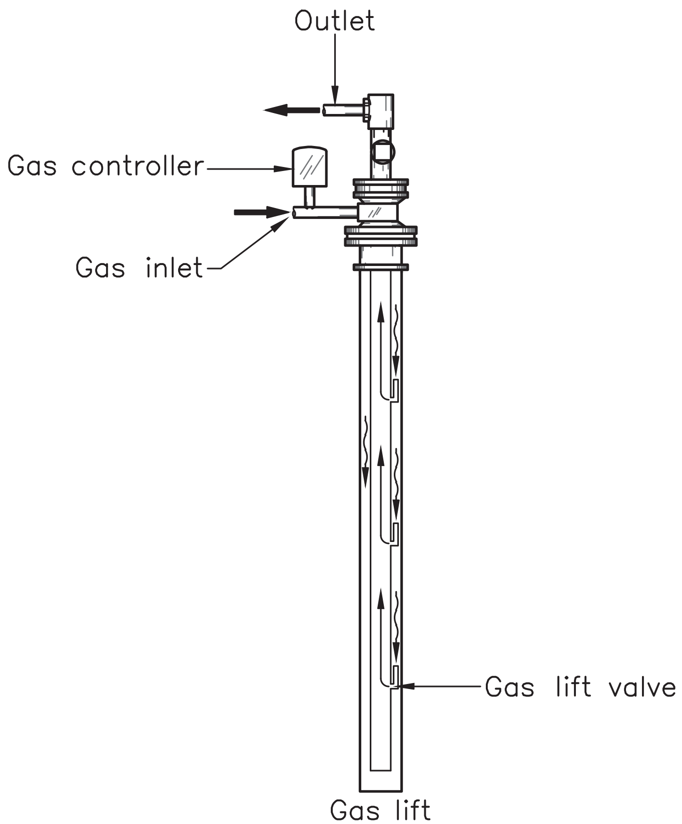


Fig. 6.5 Gas lift system.
Courtesy of PennWell Publishing, Dictionary of Petroleum exploration, drilling and Production by Norman J Hyne.

- Gas lifting works in deviated, directional, or small casing wells, or where surface area is minimal, such as offshore
- Gas lift valves are retrievable with wireline, eliminating the need for a workover rig in some instances

The *disadvantages* of gas lifting are as follows:

- Gas lifting requires a continuous high-pressure gas supply, operating costs will increase as gas prices do
- Gas lifting efficiency generally decreases with lower produced volumes and shallower depths
- Gas lifting cannot pump down a well because hydrostatic pressure from the produced fluid column is still on the formation
- Gas lifting can increase hydrate and paraffin accumulation due to the cooling effect of gas expansion
- Depletion of low bottom-hole pressure (BHP) well is difficult with gas lifting
- For gas lifting, the casing must be completely intact (without leaks) and capable of withstanding the lift pressure

6.2 Air-diaphragm pumps

Air-diaphragm pumps (Fig. 6.6) are commonly used in utility, sump pumps, and chemical services. They do not meet either ANSI or API standards and are most often used because of their portability and ability to run on compressed air.

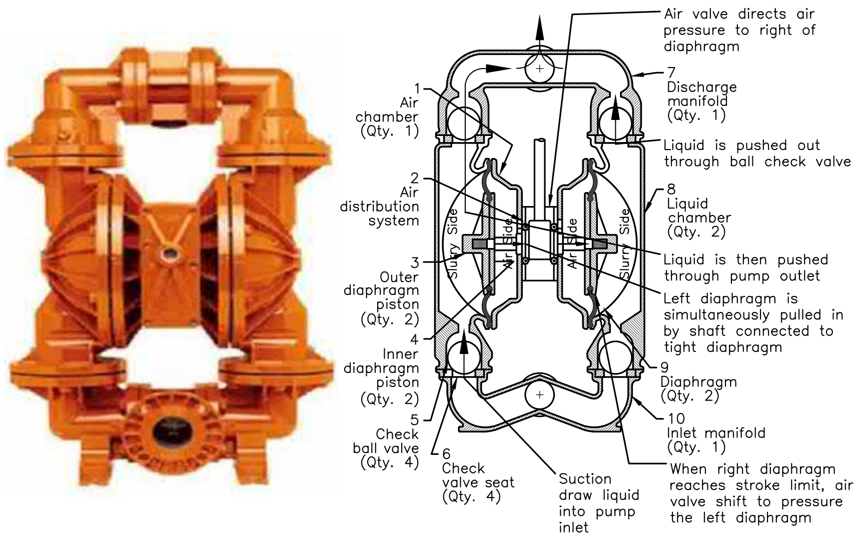


Fig. 6.6 Cross-section of an air-diaphragm pump.
Courtesy of Wilden Pumps and Engineering Company.

The *advantages* of air-diaphragm pumps are as follows:

- Portable and reusable
- Operate on compressed air
- Can handle entrained air and solids
- Simple and relatively dependable
- No shaft seals or packing complications are involved
- Nonsparking
- Available in corrosion-resistant materials
- Self-priming
- Low initial cost

The *disadvantages* of air-diaphragm pumps are as follows:

- They do not meet normal standards for continuous duty process service
- Air supply valves occasionally become plugged. (These valves require oil lubrication from a self-contained reservoir. This is normally not a problem, but the valves do need oil to operate.)
- Ball check valves wear and occasionally stick
- Limited to relatively low temperature (approximately 200°F) (93°C) and pressures (approximately 120 psi) (827 kPa)
- Noisy operation

6.3 Regenerative turbine

Regenerative turbine or "Disk-Friction" pumps are sometimes also called "Westco" pumps, which was the name used by one of the earliest manufacturers of this type of pump. The designation "turbine type" is frequently confusing and not at all descriptive of the principle of operation involved.

Externally, a regenerative turbine pump appears similar to a centrifugal pump, but the principle of operation is quite different. Fig. 6.7 shows several internal views of a regenerative turbine. The pump consists basically of an impeller wheel or disk, with vanes on the periphery, rotating in a concentric case. The case provides an open passage around the impeller except at the cutoff point between the discharge and suction nozzles. Liquid enters the pump at the periphery of the impeller and is carried by the friction of the rotating impeller to the discharge port. The cutoff point between the discharge and the suction is close fitting to prevent excessive internal leakage. Also, the plates on each side of the impeller must be close fitting to prevent excessive leakage along the sides of the impeller. A pump usually contains several impellers.

The characteristics of a regenerative turbine pump make them principally suited for relatively low capacities at medium-high heads. Fig. 6.8 shows a typical performance curve for a regenerative turbine pump at 1750 rpm. Such pumps are made in capacities up to 150 gpm or more, but ordinarily these are not economical compared to centrifugal pumps at capacities above 15 to 20 gpm.

As shown in Fig. 6.8, the total head falls off rapidly as capacity is increased. Maximum brake horsepower is required at shutoff, with the horsepower dropping as

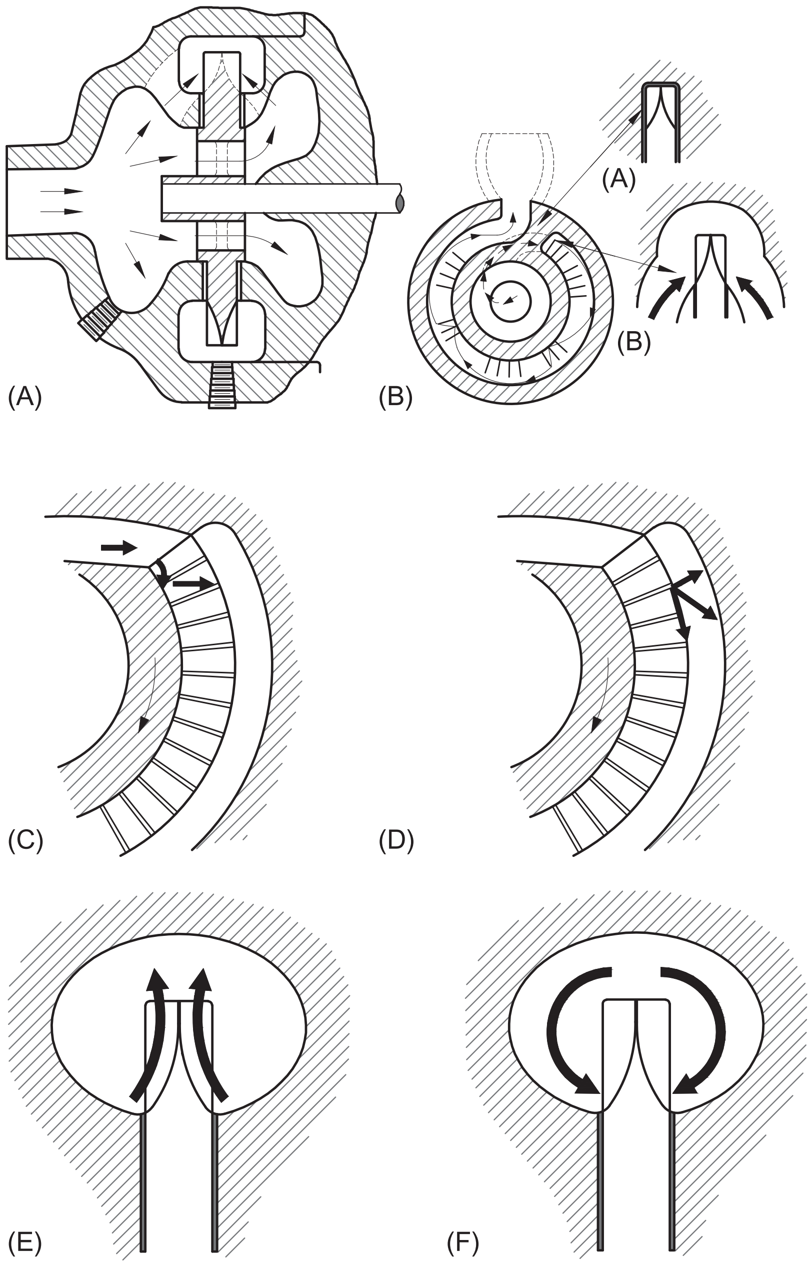


Fig. 6.7 Several cross-sectional internal views of a regenerative turbine (disk friction) pump. (A) Liquids enters the pump and into the impeller wheel or disk (B). The liquid is carried by the friction of the rotating impeller (C) and (D). Close fitting plates (E) and (F) on each side of the the impeller prevent excessive internal leakage.
Courtesy of Marcel Dekker, Pumps for chemical processing by J.T. McGuire, 1990.

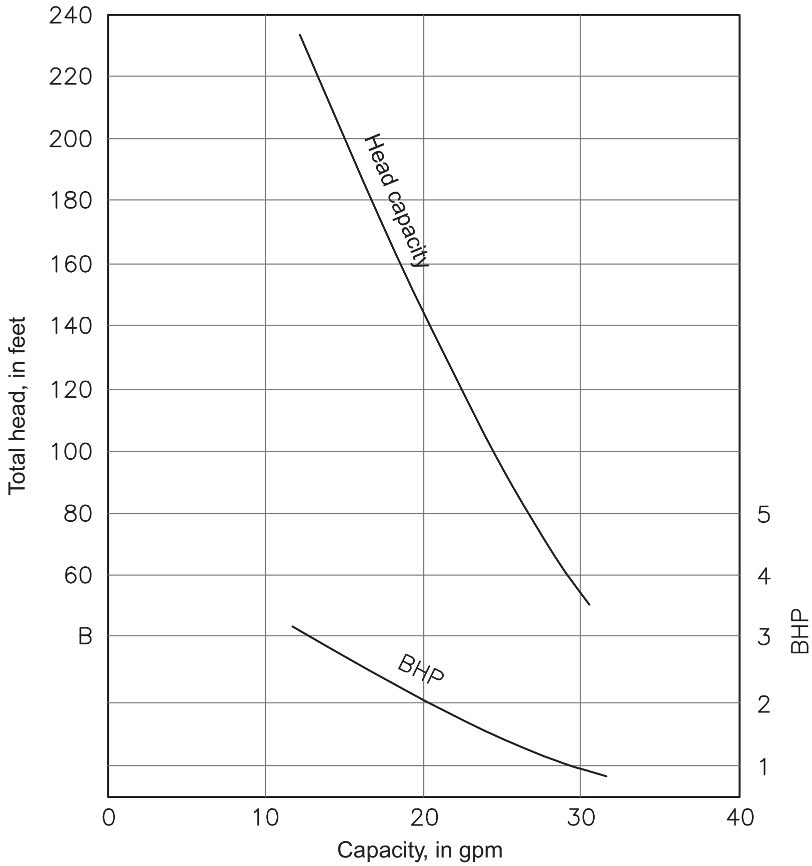


Fig. 6.8 Typical performance curve for a disk friction pump.
Courtesy of Marcel Dekker, Pumps for chemical processing by J.T. McGuire, 1990.

capacity is increased. Frequently, drivers are not sized for shutoff, but a PSV is installed to prevent overloading at low flow rates.

As one would expect, the efficiencies of regenerative turbine pumps are never very high, but for small capacity applications they are often more effective than centrifugal pumps. For example, the efficiency shown in Fig. 6.8 at 6 gpm for 170 ft (52 m) of head is about 35%. The efficiency of a centrifugal pump for this rating would be about 15%.

An advantage of the regenerative turbine pump over the centrifugal pump is that the regenerative turbine pump can handle up to about 20% by volume of vapor along with the liquid pumped. Regenerative turbine pumps are also self-priming, provided the case is filled with liquid to act as a seal.

Regenerative turbine pumps are ordinarily used in clean, nonviscous services at flow rates less than 20 gpm. For low flow rates, regenerative turbine pumps cost less than centrifugal pumps of comparable capacity.

The efficiency of the pump falls rapidly as the close clearances between the rotating disk and the case are increased by wear. Sand, mill scale, or similar foreign particles in the liquid pumped may expand the clearances to a point where satisfactory operation can no longer be obtained. This is the primary weakness of the regenerative turbine pump and has significantly reduced its application. Regenerative turbine pumps are best used for condensate return or small boiler feed services or for LNG loading where the liquids are usually clean.

6.4 Jet pumps

A jet pump (Fig. 6.9) produces a high-velocity jet of almost any fluid to pump another fluid by entrainment with pressure recovery in a diffuser. The theory of jet pump operation can be found in most engineering handbooks and therefore is not presented in this book. Since the performance of jet pumps is developed empirically from manufacturers' tests, application of jet pumps to specific service is usually a matter of clearly stating the requirements to be met.

Jet pumps are usually less efficient than other pumping devices and require a source of high-pressure fluid operation. This considerably limits their range of application.

Jet pumps are primarily used to produce and maintain a vacuum by removing vapors from a closed system. In this service, they are usually called "ejectors." Ordinarily, steam provided the motive power. Jets for this purpose are frequently combined with condensing equipment, and the entire system is purchased as a unit. Steam jets are occasionally used when boiler water capacities are involved. Because jet pumps

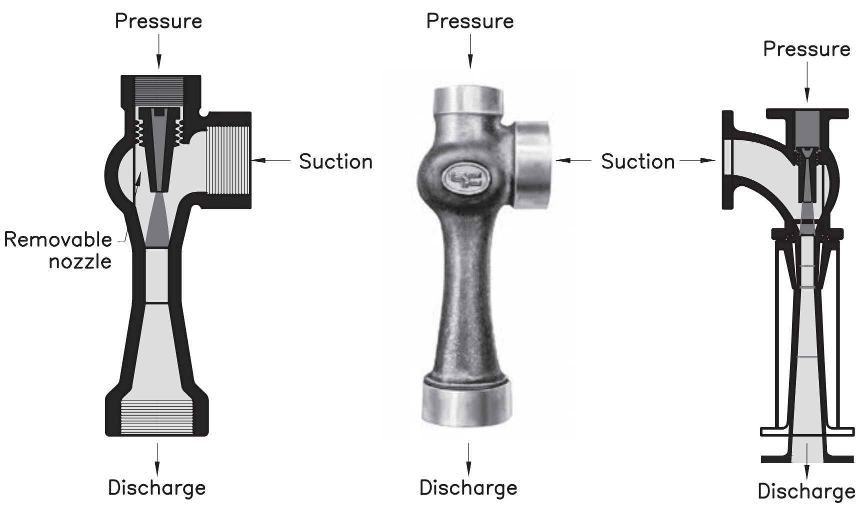


Fig. 6.9 Typical jet pumps (ejectors).
Courtesy of Schutte and Koerting Division, Ketema, Inc.

have no moving parts, they are sometimes used to dewater sumps, especially where gritty or dirty liquids are handled.

Jet pumps are sometimes used for mixing liquids in a vessel. Some of the contents are pumped back into the vessel through a submerged jet designed to entrain and circulate the rest of the tank.

6.5 Slurry pumps

Positive displacement progressive cavity-type pumps are the most commonly used type of pump used for slurry services in both upstream and downstream operations. Progressive cavity pumps lend themselves well to broad ranges of moderate discharge pressures, flow rates less than 100 gpm, and slurry percentages. Initially, progressive cavity pumps were designed for sewage sludge services. They are not designed for the oil and gas industries. Progressive cavities, like all slurry pumps, have finite lives. Manufacturer's emphasis is usually on quick disassembly and parts sales rather than precision design.

Other than progressive cavities, very few other types of positive displacement pumps have worked satisfactorily in slurries. Centrifugal pumps are often the best selection for slurry services with flows above 50–100 gpm and heads below 150 ft (46 m), particularly with nonhazardous fluids. Impeller speed/erosion relationships usually limit higher head designs. The mining industry employs a broad range or rubber lined centrifugal pumps which should be considered for water slurries. Several pump companies supply horizontal and vertical centrifugals to the cement and paint industries which resist wear with hardened iron materials.

6.6 Peristaltic pumps

Peristaltic pumps are positive displacement pumps. Peristaltic pumps (Fig. 6.10) are used for pumping fluids such as waste sludges, lime and cement mortar, adhesives, and shear-sensitive fluids such as latex paints.

The pumping action of peristaltic pumps results from alternative compression and relaxation of the hose, which is manufactured of Buna or natural rubber. The hose is compressed within a circular housing by a rotor with lubricated shoes that come in direct contact with the outer walls of the hose. The pumped fluid is in contact only with the inner walls of the hose.

Peristaltic pumps are available in capacities up to 300 gpm and pressures up to 220 psig (1517 kPa).

The *advantages* of peristaltic pumps are as follows:

- Low fluid shear
- No seals or packing
- No valves
- Dry running

Fig. 6.10 Peristaltic pump.
Courtesy of Watson-Marlow/Bredel Pumps.

The *disadvantages* of peristaltic pumps are as follows:

- Limited hydraulic range
- Pulsating flow
- Limited life due to the component life of the hose

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Compressor fundamentals

7

7.1 Engineering principles

7.1.1 Background

Compressors are used whenever it is necessary to flow gas from a low-pressure system to a high-pressure system. Flash gas from low-pressure vessels used for multistage stabilization of liquids, oil treating, water treating, and so on, often exists at too low a pressure to flow into the gas sales pipeline. Sometimes the gas is used as fuel and the remainder is flared or vented. In many cases, it is economically attractive to compress this gas to a high enough pressure so it can be sold.

Compression may also be required due to environmental reasons. Flash gas, which might otherwise be flared, may be compressed for sales, or gas produced with oil (associated gas) may be compressed for reinjection to avoid flaring or to help maintain reservoir pressure. Flash gas compressors are characterized by low throughput rates, high-pressure drops, and overall compression ratios between 5 and 20.

In some marginal gas fields and in many large gas fields that experience a decline in flowing pressure with time, it may be economical to allow the wells to flow at surface pressures below that required for gas sales. In these cases, a booster compressor (one in which the ratio of discharge to suction pressure is low) may be installed. Booster compressors are also used on long pipelines to restore pressure lost to friction. Booster compressors are characterized by high throughput rates and overall compression ratios between 2 and 5.

The use of compressors is probably more prevalent in oil field facilities than in gas field facilities. Oil wells often require low flowing surface pressures, and the gas that flashes off the oil in the separator must be compressed. Often, natural gas is injected into the tubing of the well to lighten the column of liquid and reduce hydrostatic back-pressure. This "gas lift" gas is produced with the well fluids at low pressure. Compressors are used so the gas lift gas can be recirculated and injected back into the well at high pressure. Gas lift compressors are characterized by both high overall ratios of discharge to suction pressure (pressure ratios) and relatively high throughputs.

The designer must select the type of compressor to use for each application; determine power requirements; design the piping system associated with the compressor; and specify materials and details of construction for bearings, cylinders, and so on. On standard applications, the engineer may allow the vendor to specify materials and construction details for the specified service conditions. However, even in these instances the engineer should be familiar with different alternatives so that he can better evaluate vendor's proposal and alternative proposals.

Most work involving compressors falls into one of the three categories:

- Purchasing and installing new compressors
- Troubleshooting problems during start-up or when in-service

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00007-7

- Modifying compressors to resolve problems or to accommodate service changes, for example, different flow rates, gas pressure, and so on

The goal in the earlier cases is the same, that is, to maximize company profits while at the same time provide safe, reliable equipment that satisfies the operating requirements and local environmental constraints.

Profitability is a long-term goal that involves the following factors:

- Meeting environmental (including noise restrictions) and safety needs
- Initial cost
- Installation and commissioning expense
- Lifecycle energy consumption which is a major expense of compressors
- Reliability
- Maintaining production which is often the overwhelming economic factor
- Operability (troublesome equipment wastes resources that can be used for profitable work)
- Start-up on time, the first time, in critical services
- Maintenance expense
- Operating flexibility

Even though each of the earlier factors must be considered when making decisions, there is some degree of conflict between them. For example, buying an inexpensive unit may keep initial costs down but it may also be less reliable and require frequent maintenance. On the other hand, buying the most reliable unit may be prohibitively expensive in up-front costs and unnecessary to ensure acceptable performance.

It is imperative one use engineering judgment when deciding which of the aforementioned factors are the most important. It is especially important to communicate with the field personnel who are responsible for operating and maintaining the unit. Their input on the relative priority of the earlier factors is invaluable. At the end of the day, engineering judgment will always be necessary.

7.1.2 Compressor classification

As shown in Fig. 7.1, compressors are usually classified into two main categories—*Dynamic and Positive Displacement*. Dynamic compressors compress gas continuously. Gas enters the compressor, is acted upon, and is discharged without interruption of flow at any point in the process. Compressor performance is described by a head-capacity curve, sometimes called a characteristic curve. Compression is fairly efficient, so long as the system head requirement matches the head, the compressor can produce at a given capacity. Positive displacement compressors compress gas intermittently. Compression is cyclic, in that a fixed quantity of gas is drawn into the compressor, is acted upon, is discharged, and then the cycle is repeated. The discharge pressure requirement has only a small secondary effect on the volume of gas compressed. To change capacity, one must change the number of cycles. The types within this category include:

- Reciprocating
- Rotary
  - Screw
  - Sliding Vane

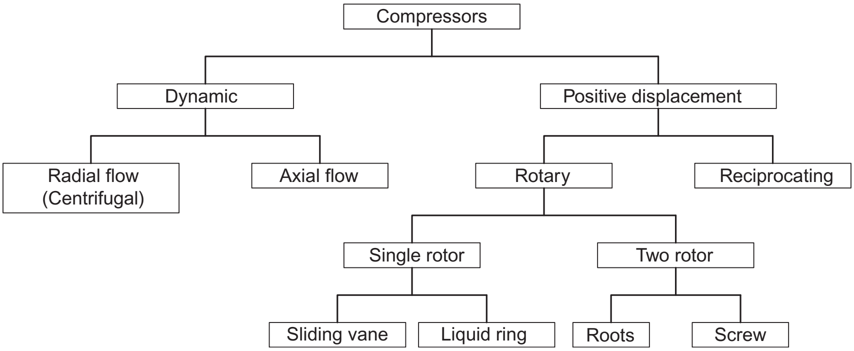


Fig. 7.1 Compressor classification.

- Liquid Ring
- Roots (Straight Lobe) Blower

Reciprocating compressors are positive displacement machines in which the compressing and displacing element is a piston moving linearly within a cylinder. Applications usually require moderate inlet volume with a high compression ratio beyond the capability of dynamic compressors. Compression is usually accomplished in stages to permit intercooling which reduces power requirements and prevents high discharge temperatures that might damage the compressor components if left unchecked. Compressor cylinders have a progressively higher pressure rating in successive stages.

Rotary compressors are positive displacement machines in which compression and displacement of gas is accomplished by the positive action of rotating elements. Typically used in applications requiring low-cost compact and yet very reliable compression system, for example, refrigeration compressor.

Dynamic compressors develop a rise in pressure by increasing the kinetic energy of the gas flow on a continuous basis. The types within this category include:

- Centrifugal (radial)
- Axial

Centrifugal (radial) and axial compressors are rotary continuous flow machines where rotating impellers impart a velocity head to the fluid which is converted to pressure head by the case.

Another means of compressing gases on a continuous flow basis is the ejector. This device has no moving parts, but requires a motive fluid that mixes with the gas being compressed. Because of its rather low efficiency and limited scope of application, the ejector is not covered in the text. Typical applications include vacuum service on refinery vacuum distillation columns and air ejection from steam condensers.

The range of application of compressors varies widely, with inlet pressures ranging from vacuum to several thousand psi and discharge pressures ranging from less than atmosphere to above 15,000 psi (1034 bar). Gases handled vary from hydrogen, with a molecular weight of 2, to refrigerants and gases having molecular weights in the low hundreds. The size, types, and construction of compressors vary greatly to accommodate this diversity of service.

Compressor application ranges by type:

- *Centrifugal* compressors have the widest range of application.
- *Reciprocating* compressors can compress lower volumes than centrifugal compressors and are suitable for high-pressure applications.
- *Axial* compressors are suitable for high capacity applications.
- *Rotary* compressors: Centrifugal and reciprocating compressors cover the entire range of application for rotary compressors. As a result, rotary compressors generally are selected for reasons other than their pressure and capacity range.

7.1.3 Compressor types

7.1.3.1 High-speed vs. slow-speed "separable" units

Reciprocating compressors are classified as either "high-speed" or "slow-speed." Typically, high-speed compressors run at a speed of 900 to 1200 rpm while slow-speed units run at 200-600 rpm.

High-speed reciprocating compressors are available in sizes ranging from 50 to 2500 HP (37.3 to 1864 kW). Most are equipped with 2, 4, or 6 compressor cylinders. Fig. 7.2 shows a cross-section of a high-speed separable reciprocating compressor.

The advantages of high-speed compressors over slow-speed compressors are that they can be skid mounted and self-contained for easy installation, they are less expensive than comparably sized low-speed units, they are available in sizes suitable for field gathering offshore and onshore, and they are easily moved to new locations.

Slow-speed reciprocating compressors are available in two size ranges. Some small, skid-mounted, one-and-two-power cylinder field gas slow-speed compressors are rated from 140 HP to 360 HP (104.4 kW to 268.5 kW). Most slow-speed units are in the 1500 HP to 6000 HP (1118.6 kW to 4474.2 kW) range, but larger sizes, up to 13,000 HP (9694.1 kW), are available.

The advantages of slow-speed units over high-speed units include lower fuel consumption, longer operating life, and lower operation and maintenance costs. However, slow-speed units require heavy foundations and a larger degree of vibration and pulsation suppression. The larger sizes usually must be field erected, and thus have higher installation cost.

Fig. 7.3 shows a high-speed compressor frame and cylinders. The upper compressor is called a two-throw machine because it has two cylinders attached to the frame and running off the crankshaft. The lower compressor is called a four-throw machine because it has four cylinders attached to the crankshaft. The number of throws refers to the crankshaft, as more than one cylinder may be attached to a crank throw. That said, the number of crank throws and the number of cylinders are usually the same.

A compressor may have any number of stages. Each stage normally contains a suction scrubber to separate any liquids that carry over or condense in the gas line prior to

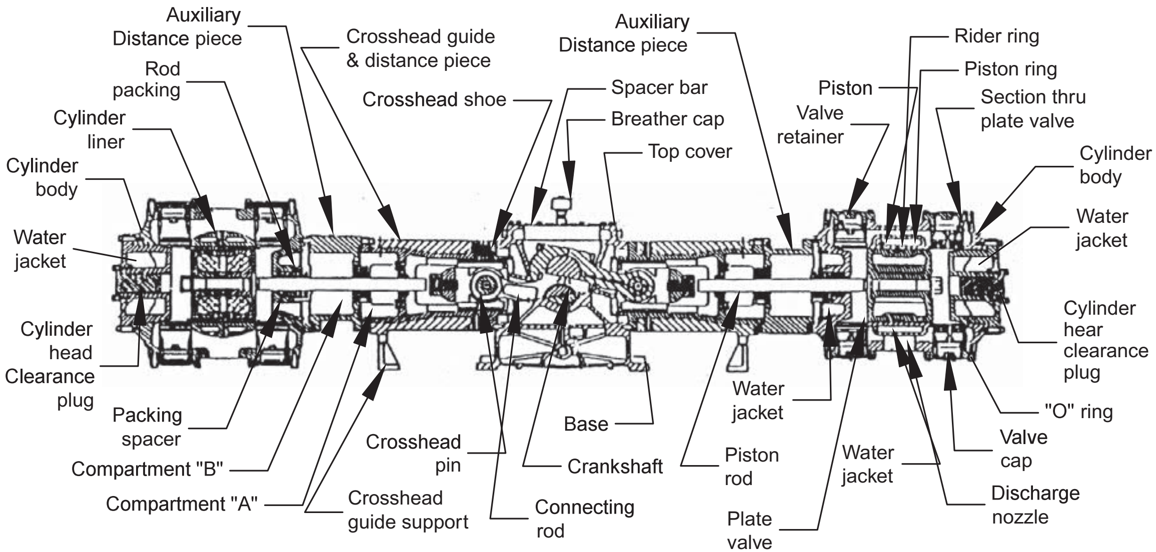


Fig. 7.2 Cross-section of a high-speed separable reciprocating compressor.

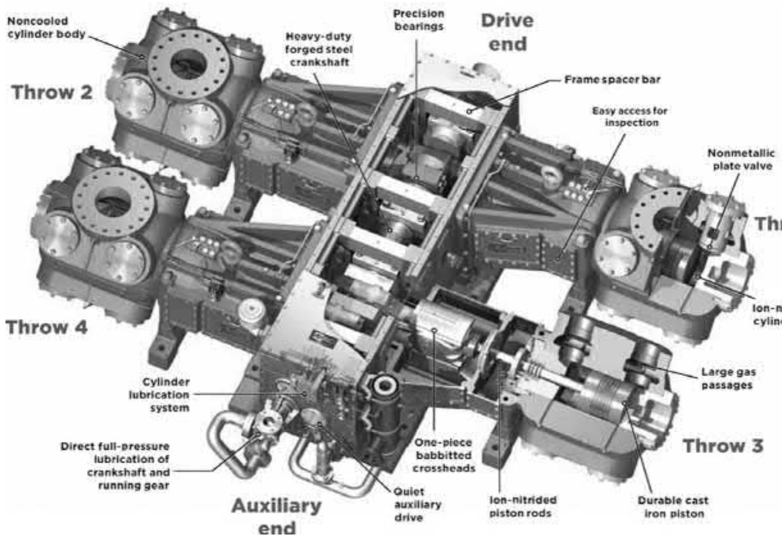


Fig. 7.3 Example of high-speed reciprocating compressor: two-throw and four-throw. Courtesy of Dresser Rand Co.

the compressor cylinder (or case, for centrifugal compressors). When gas is compressed, its temperature increases. Therefore, after passing through the cylinder, the gas is usually cooled before being routed to another suction scrubber for another stage of compression. A stage of compression consists of a scrubber, cylinder, and after-cooler. The discharge from the final cylinder may not be routed to an after-cooler.

The number of throws is not the same as the number of stages of compression. It is possible to have a two-stage, four-throw compressor. In this case, there would be two sets of two cylinders working in parallel. Each set would have a common suction and discharge. Units greater than 1000 HP normally have a cooler on a separate skid.

7.1.3.2 High-speed separable

High-speed units are normally "separable." That is, the compressor frame and the driver are separated by a coupling or gear box. This is opposed to an "integral" unit, in which power cylinders are mounted on the same frame as the compressor cylinders, and the power pistons are attached to the same crankshaft as the compressor cylinders.

High-speed units are typically engine or electric motor driven, although turbine drivers have also been used. Engines or turbines can be either natural gas or diesel fueled. By far the most common driver for a high-speed compressor is the natural gas-driven engine.

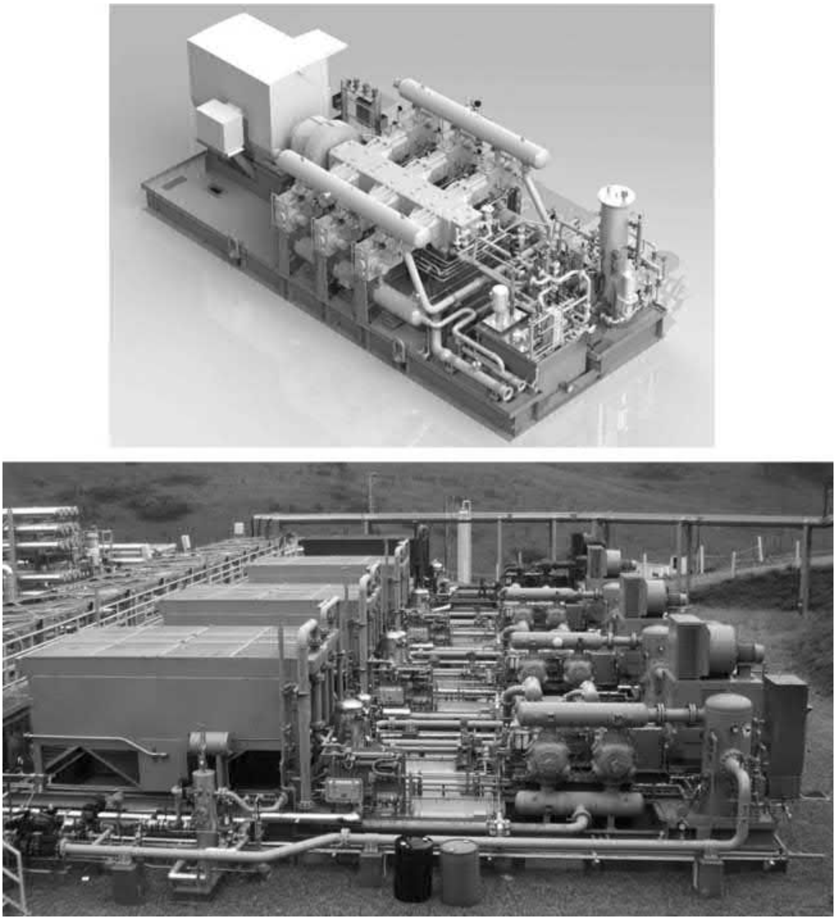


Fig. 7.4 Examples of high-speed reciprocating compressor packages.
Courtesy of Dresser Rand Co.

Fig. 7.4 shows a high-speed engine-driven compressor package. The unit typically comes complete on one skid with driver, compressor, suction scrubbers, and discharge coolers for each stage of compression, and all necessary piping and controls. On large units, greater than 1000 HP (746 kW), the cooler maybe shipped on a separate skid.

7.1.3.3 Low-speed integral engine compressors

Low-speed units are typically integral in design, as shown in Fig. 7.5. "Integral" means that the power cylinders that turn the crankshaft are in the same case, or housing, as the cylinders that compress the gas. There is one crankshaft with engine pistons and compressor pistons attached.

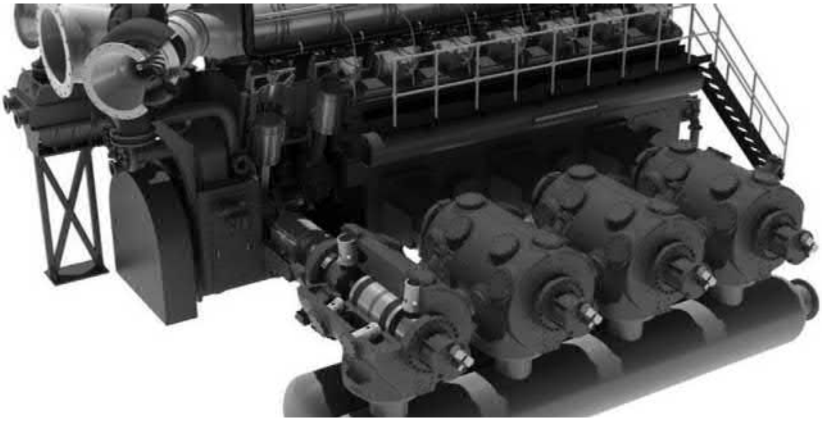


Fig. 7.5 Cross-sectional view of a low-speed integral engine compressor. Courtesy of Cooper Industries Energy Services Group.

Fig. 7.6 shows a very large integral compressor that is typical on compressors in the 2000 HP to 13,000 HP (149.4 kW to 9694.1 kW) range. This particular unit has sixteen power cylinders (eight on each side) and four compressor cylinders. Obviously one of these large integrals requires a very large and expensive foundation and must be field erected. Often, even the compressor cylinders must be shipped separate from the frame due to weight and size limitations.

Large integrals are much more expensive both to purchase and to install than either high-speed separable or centrifugal. For this reason, even though they are the most fuel-efficient choice for large power needs, large integrals are not often installed in oil and gas fields. They are more common in plants and pipeline booster service, where their fuel efficiency, long life, and steady performance outweigh their higher initial investment.

7.1.3.4 Rotary vane compressors

A rotary vane compressor is shown in Fig. 7.7. They are available in both single- and two-stage tandem units. The maximum discharge pressure is 400 psig. They are available in sizes ranging from 1 HP to 500 HP (0.75 kW to 373 kW). They are commonly used under 125 HP (93 kW).

Rotary vane pumps are good in vacuum service, do not produce pulsating flow, requires less space, and inexpensive for low HP (kW) vapor recovery or vacuum service. On the other hand, vane rotary compressors require clean air or gas, require 5%–20% more horsepower than a reciprocating compressor, use ten times the oil of reciprocating compressors, and in hydrocarbon service, require seal oil with an after-cooler and separator to recycle oil.

Fig. 7.6 Low-speed integral engine compressor.
Courtesy of Cooper Industries Energy Services Group.

Fig. 7.7 Typical vane rotary compressor.

7.1.3.5 Rotary screw (helical lobe) compressors

Fig. 7.8A and B shows sectional views of screw (helical Lobe) compressors. Screw compressors are available in single- and two-stage tandem units. The maximum discharge pressure is 300 psig (21 bar). They are available in sizes ranging from 1 HP to 6000 HP (0.746 KW to 4476 KW). They are commonly used under 400 HP (298 KW) in hydrocarbon service and under 800 HP (596 KW) in air service.

Screw compressors are available as nonlubricated especially for air service. They can handle dirty gas, moderate amounts of liquids, but no slugs. They do not produce pulsating flow. At low discharge pressures (less than 50 psig (3.44 bar)) they can be more efficient than reciprocating compressors. They are also compact in size and do not require large foundations. Their efficiency is comparable to reciprocating compressors. Their initial installation expense is one half that of a reciprocating compressor.

In hydrocarbon service screw compressors require seal oil with an after-cooler and separator to recycle oil. At discharge pressures over 50 psig, they use 10%–20% more horsepower than reciprocating compressors. They have a low tolerance to change in operating conditions of temperature, pressure, and compression ratio.

7.1.3.6 Centrifugal compressors

Centrifugal compressors are available in axial and radial split units (Figs. 7.9 and 7.10). Centrifugal compressors run at speeds between 20,000 and 30,000 RPM. They are typically engine or turbine driven. Turbines are commonly used due to high rotating speeds. Centrifugal compressors are used in pipeline booster service where high volume and low compression ratios exist. They are also used in very high flow rate gas lift service. They are not very well suited for high ratio, low-volume applications.

Centrifugal compressors are available in sizes ranging from 500 HP to 20,000 HP (373 KW to 14,920 KW) and are typically available in 1000 HP (746 KW) increments.

Centrifugal compressors deliver high horsepower per unit of space and weight, are easily adapted to combined cycle and cogeneration for high fuel efficiency, easily automated for remote operations, easily skid mounted and self-contained. They also offer lower initial costs, lower maintenance cost than reciprocating compressors, and have a high availability factor. On the other hand, centrifugal compressors have the lowest compressor efficiency, have limited flexibility for capacity changes, and have a higher fuel rate than reciprocating units.

Table 7.1 presents the comparison between centrifugal and reciprocating compressors.

7.2 Thermodynamic principles of compressors

7.2.1 Introduction

The principles of compression are derived from the laws of thermodynamics. Thermodynamic principles are applicable to both positive displacement and continuous

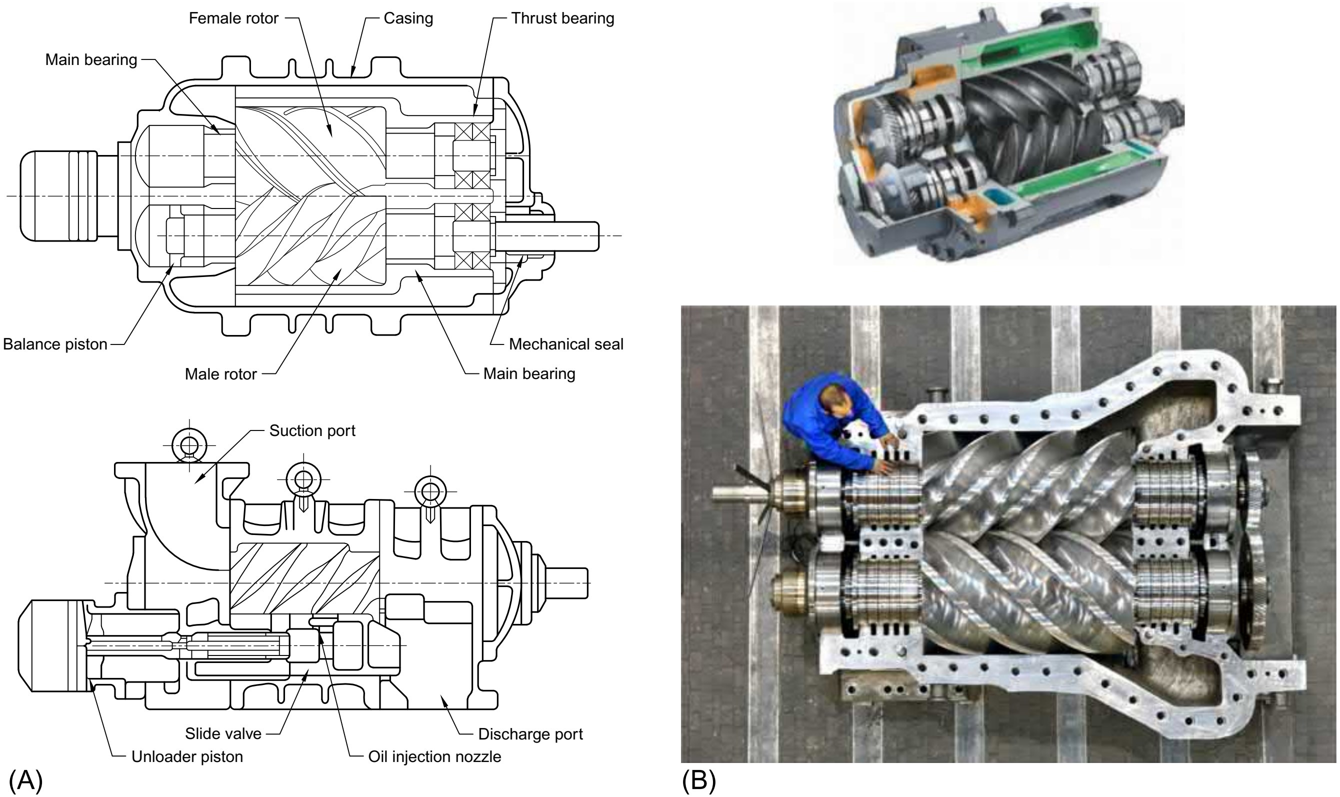


Fig. 7.8 (A) Schematic of a rotary screw (helical lobe) compressor. (B) Sectional views of a rotary screw (helical lobe) compressors.

Fig. 7.9 *Top:* Rotor assembly being installed in casing; *Bottom*: Rotor assembly installed in an axially split centrifugal compressor.

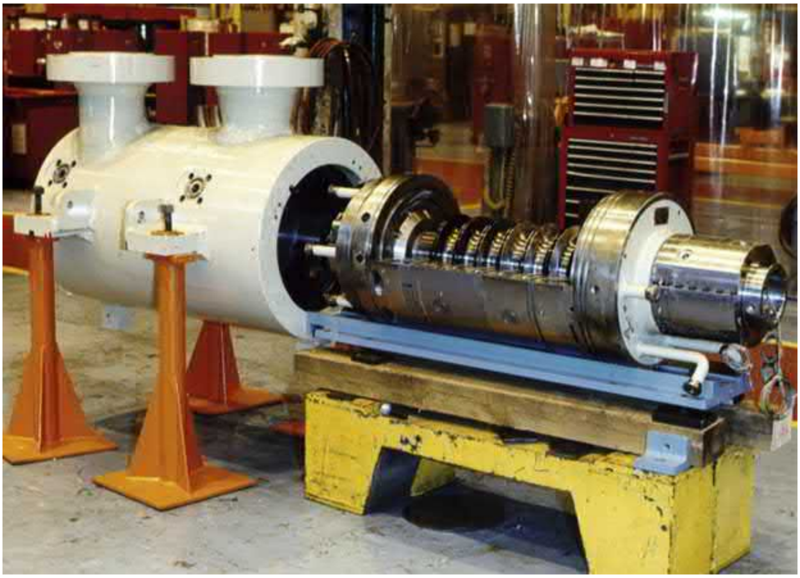


Fig. 7.10 Radially split centrifugal compressor.

Table 7.1 Centrifugal versus reciprocating comparison

Factor	Centrifugal	Reciprocating
Initial Investment	Low	High
Operating Cost (excluding fuel)	Less	More
Fuel Consumption	High	Low
Flexibility	Less	More
Efficiency	Better at Low Ratios	Better at High Ratios
Project Life	Short Life Favorable	Long Life Favorable
Space Requirements	Less	More
Relocation	Less Difficult	More Difficult

flow compressors. Compressing gases involves complications that pumping liquids does not. The compressible nature of gases requires one to account for their more complex behavior through the application of thermodynamic principles.

The information contained in this section provides a detailed tutorial on compression fundamentals with the intent to help one understand how compressors work. Understanding thermodynamics of compression is helpful for the following reasons:

1. It helps in selecting the best category and mechanical design for the required flow rate and differential pressure. These selections can significantly impact the total lifecycle cost, including installation and energy costs.

2. Properly defining the mass- and volumetric-flow rates is essential in design and specification, including rerates, and may also be helpful in troubleshooting.

For example, there are several commonly used conventions for defining flow rate:

- Pounds/hour (lb/hr)
- Million standard cubic feet per day (MMSCFD)
- Standard cubic feet per minute (SCFM)
- Actual cubic feet per minute (ACFM)
- Inlet cubic feet per minute (ICFM)
- Moles per hour (mol/hr)

1. The thermodynamic properties of the gas, or mixture of gases, affect the energy required to do the compression. The energy requirements affect both the size of the driver and the mechanical design of the compressor. Both are critical factors in new applications and rerates.
2. The thermodynamic properties of a mixture of gases can be estimated based on the properties of the individual components. Most compression applications involve mixtures.
3. In gases with water vapor, the water content also needs to be accounted for.

The concepts that follow apply to all categories of compressors.

7.2.2 General "perfect" gas law and compressibility factor

Eq. (7.1), known as the General Gas Law, defines the behavior of a "Perfect" gas in terms of variables shown: pressure, temperature, volume, and so on. This is a useful starting point, even though few gases actually are "perfect":

$$pV = WRT \tag{7.1}$$

where:
p = absolute pressure, lb/ft^2
V = volume, ft^3
W = weight, lbs
$R = R_o/M$ = constant for specific gas
R_o = universal gas constant
= 1545.3 ft-lb/lb-mol °R
T = absolute temperature, °R
M = molecular weight

For a continuous flow process, Eq. (7.1) is modified as follows:

$$PQ = \frac{(10.73)wT}{M} \tag{7.2}$$

where:
Q = actual volumetric flow rate, ACFM
w = weight flow, lbs/min
P = is now in psia

To correct for deviations from a "perfect" gas, a compressibility factor, Z, is added to Eq. (7.2) (refer to Eq. 7.3). "Z" is an empirical factor to correct the equation for actual, real gases that deviate from "perfect."

$$PQ = \frac{(10.73)wTZ}{M} \tag{7.3}$$

For example, at standard conditions (14.7 psia, 60°F) (1 bar, 15.5°C) the "Z" factor of most gases is generally assumed to be 1.0. However, some gases deviate appreciably even at standard conditions. For example, normal butane has a Z_{std} value of 0.975 where Z_{std} denotes the factor is at standard conditions.

Values for Z are available in charts for the gas being compressed. If chart is not available, or if the gas is a mixture, generalized compressibility charts may be used. To use these charts, it is necessary to compute the "reduced pressure" (P_r) and "reduced temperature" (T_r) as shown in Eqs. (7.4), (7.5).

$$\mathrm{P_r} = \frac{P}{P_c} \tag{7.4}$$

and

$$T_{\mathrm{r}} = \frac{T}{T_c} \tag{7.5}$$

where:
P_r = reduced pressure
P = actual absolute pressure, psia
P_c = critical pressure of the gas, psia
T_r = reduced temperature
T = actual absolute pressure, °R
T_c = critical temperature of the gas, °R

The critical pressure and temperature of a gas mixture are in detail in *Surface Production Operations, Volume 2-Gas Handling Facilities*. Volume 2 contains a collection of compressibility curves for specific gases and generalized charts.

The compressibility of some pure gases, specifically steam and ammonia, cannot be accurately predicted using the generalized charts. However, steam tables and an individual chart for pure ammonia are available. When water vapor or ammonia content of a mixture is small, less than 5%, the generalized charts may be used for the mixture with relatively good accuracy.

For gas mixtures containing hydrogen or helium, effective values of critical pressure and temperature for helium and hydrogen must be used to derive acceptable accuracy from the generalized charts. These effective values are included in standard handbooks such as *Gas Processors Suppliers Association* (GPSA) Engineering Data book.

7.2.3 Gas properties

7.2.3.1 Gas mixtures

Knowing the mole fractions in a mixture leads to calculation of several important properties of the mixture:

- Molecular weight, M_m
- Molal specific heat, $MC_{p(m)}$
- Critical pressure, $P_{c(m)}$
- Critical temperature, $T_{c(m)}$

Fig. 7.11 shows a sample calculation of gas mixture properties.

The mole fraction "X" can be determined from Eqs. (7.6A), (7.6B), and (7.6C).

$$X_1 = N_1/N_m \tag{7.6A}$$

$$X_2 = N_2/N_m \tag{7.6B}$$

Properties of a gas mixture - natural gas

Compound	Mol frac Xi	Mol. wt. M	M x Xi	Pc (psia)	Pc x Xi (psia)	Tc (R)	Tc x Xi (R)	MCp @ 150F (BTU/lb-mol-R)	MCp x Xi @ 150F (BTU/lb-mol-R)
Air	0.02	28.97	0.449	547.00	8.5	239.00	3.7	6.98	0.108
O_2	0.00	32.00	0.000	732.00	0.0	278.00	0.0	7.07	0.000
N_2	0.00	28.02	0.000	492.00	0.0	227.00	0.0	6.98	0.000
H_2O	0.00	18.02	0.000	3187.00	0.0	1165.00	0.0	8.94	0.000
CO	0.00	28.01	0.000	507.00	0.0	242.00	0.0	6.97	0.000
CO_2	0.01	44.01	0.405	1073.00	9.9	548.00	5.0	9.37	0.086
H_2S	0.00	34.08	0.000	1306.00	0.0	673.00	0.0	8.28	0.000
H_2	0.00	2.02	0.000	327.00	0.0	83.00	0.0	6.94	0.000
Methane	0.90	16.04	14.490	673.00	607.9	344.00	310.7	8.95	8.084
Ethylene	0.00	28.05	0.000	742.00	0.0	510.00	0.0	11.39	0.000
Ethane	0.06	30.07	1.699	708.00	40.0	550.00	31.1	13.77	0.778
Propylene	0.00	42.08	0.000	667.00	0.0	657.00	0.0	16.79	0.000
Propane	0.01	44.10	0.525	617.00	7.3	666.00	7.9	19.53	0.232
i-Butane	0.00	58.12	0.052	529.00	0.5	735.00	0.7	25.75	0.023
n-Butane	0.00	58.12	0.081	551.00	0.8	766.00	1.1	25.81	0.036
i-Pentane	0.00	72.15	0.029	483.00	0.2	830.00	0.3	31.67	0.013
n-Pentane	0.00	72.15	0.014	489.00	0.1	846.00	0.2	31.82	0.006
Benzene	0.00	78.11	0.000	714.00	0.0	1012.00	0.0	23.51	0.000
Hexane	0.00	86.17	0.069	440.00	0.4	915.00	0.7	38.17	0.031
Heptane	0.00	100.20	0.000	397.00	0.0	973.00	0.0	47.49	0.000
Octane	0.00	114.22	0.000	362.00	0.0	1025.00	0.0	57.00	0.000
	-----		-----		-----		-----		-----
	1.00		17.81		675		361		9.40
HC mixture properties			Mm		Pc(m) (psia)		Tc(m) (R)		MCp(m) @ 150F (BTU/lb-mol-R)
			17.81		675		361		9.40

k @ 150 F
MCp(m) / [MCp(m) - 1.986] = 1.27

(k-1)/k = 0.211
@ 150F

Fig. 7.11 Sample calculation of gas mixture properties.

$$X_3 = N_3/N_m \tag{7.6C}$$

where:
N_m = total moles in a mixture
N_1, and so on = number of moles of each individual component

A "mole" is actually a number of molecules, approximately 6×10^{23}. A "mole fraction" is the ratio of molecules of one component in a mixture. For example, if the mole fraction of methane in natural gas is 0.90, then this means that 90% of the molecules are methane. Since the volume fractions are equivalent to mole fractions, the mixture is also 90%, by, methane.

The mixture fractions could also be calculated on a mass or weight basis. The mole (volume) basis is used in compressor calculations because it is a simpler, less confusing method.

The molal specific heat is used to determine the "k" value, known as the ratio of specific heats, as follows. The "k" value is often called the adiabatic exponent and is a value used in the calculation of horsepower, adiabatic head, and adiabatic discharge temperature. (Refer to Isentropic (Adiabatic) compression). The "k" value is determined by Eq. (7.7).

$$k = \frac{C_p}{C_v} = \frac{MC_{p(m)}}{MC_{p(m)} - \dfrac{Ro}{778}} = \frac{MC_{p(m)}}{MC_{p(m)} - 1.986} \tag{7.7}$$

where:
$MC_{p(m)}$ = molal specific heat (heat capacity) of mixture at constant pressure
778 = conversion factor, ft-lb/BTU
C_p = specific heat at constant pressure
C_v = specific heat at constant volume
R_o = See Eq. (7.1) for R_o definition

$MC_{p(m)}$ should be taken at the desired temperature (usually the average suction and discharge temperature). This aspect is covered in Isentropic (Adiabatic) Compression. Note that the "k" value of the mixture must be determined by first determining the molal heat capacity of the mixture (see Eq. 7.6). A common mistake is to multiply the:k: value of the individual gas components by their respective mole fraction to determine the "k" value of the mixture.

7.2.3.2 Specific gravity

The specific gravity of the gas mixture is determined by dividing the molecular weight of the mixture by that of dry air.

$$SG = \frac{M_m}{28.96} \tag{7.8}$$

7.2.3.3 Humidity

For air compressors one must account for the water vapor content. It is important to know the moisture content accurately when a process requires a definite quantity of dry air. Furthermore, the moisture in the inlet air affects the power requirement and water dropout in intercoolers and after-coolers.

In some components in a process stream it is important to know the water vapor content. In those cases, the content is usually available from process engineering, gas and chemical engineering, and so on.

The following discusses how to account for water content in air.

Relative humidity, in percent, may be determined from Eq. (7.9).

$$\%RH = \frac{P_v(100)}{P_{sat}} \tag{7.9}$$

where:
P_v = partial pressure of actual water vapor content
P_{sat} = partial pressure of water vapor when air is fully saturated at the temperature of interest (can be found in steam tables)

Specific Humidity is the ratio of the weight of the water vapor content to the weight of dry air at the existing conditions of pressure and temperature and is determined as follows:

$$S.H. = \frac{W_v}{W_{da}} = \frac{18}{28.96}\left\{\frac{P_v}{P - P_v}\right\}$$

$$= 0.622\left\{\frac{P_v}{P - P_v}\right\} \tag{7.10}$$

where:
W_v = weight of water vapor
W_{da} = weight of dry air
P = total pressure of the gas mixture, usually atmospheric, in absolute

Relative and specific humidity may be obtained from a psychrometric chart when the wet bulb and dry bulb temperature are known. However, most psychrometric charts are based on the International Standard sea-level pressure of 14.7 psia (1 bara), and are, therefore, accurate only for that barometric pressure.

For example, if a standard (14.7 psia (1 bara)) psychrometric chart were used for conditions of 5000-ft elevation (12.23 psia), 80°F wet bulb, the indicated specific humidity would be low by about 25%, and the relative humidity low by 10%. If the altitude is more than 200 or 300 ft above sea level, the following equation should be used instead of a psychrometric:

$$P_v = P_{v(wb)} - \frac{P - P_{v(wb)}}{2830 - 1.44t_{wb}}(t - t_{wb}) \tag{7.11}$$

where:
$P_{v(wb)}$ = vapor pressure in psia corresponding to wet bulb temperature (from steam tables)
t = dry bulb temperature, °F
t_{wb} = wet bulb temperature, °F

Knowing P_v, the relative and specific humidities can be calculated with Eqs. (7.9), (7.10). The volumetric or mole percent of the water vapor can be calculated from Eq. (7.6) as follows:

$$\text{mol }\%H_2O = \frac{P_v}{P}(100) \tag{7.12}$$

The mole percent of dry air is then 100 minus the mole percent of the water vapor. The other properties of the mixture of air and water vapor (molecular weight, MC_p, etc.) may then be calculated.

7.2.4 Flow measurements

Flow through a compressor may be stated in a number of different ways:

- MMSCFD
- Moles/hour
- SCFM
- Mass (weight) flow
- ACFM

7.2.4.1 MMSCFD

MMSCFD, denoting millions of standard cubic feet per day, is the most common term used by the industry to describe volumetric flow since it is independent of actual gas pressures or temperatures. It is the volume per unit time using pressure and temperatures based on some "standard" inlet conditions. The standard inlet conditions (pressure, temperature, molecular weight, and compressibility) must be known and remain constant. The standard conditions used in this text are:

Pressure = 14.7 psia
Temperature = 60°F
Compressibility = 1.0
Molecular weight = MW of subject gas

Standard volume flow should be specified on a dry gas basis. Standard volume flow, like mass (weight) flow, should be based on dry gas. Where applicable, the water content must be known for proper compressor sizing. Thus, it may be better to convert standard volume flow directly to mass (weight) flow.

This notation is often used in gas plant, gas transmission, and refinery applications.

7.2.4.2 Moles/hour (MPH)

Process engineers often use moles/hour (MPH) in material balance calculations. (A "mole" is a fixed quantity of molecules. This concept greatly simplifies process calculations.) A mole of any gas occupies approximately 379.4 cubic feet at standard conditions (14.7 psia, 60°F), and it has a weight in pounds equal to the molecular weight of the gas. For example, a mole of methane (CH_4) would have a volume of 379.4 cubic feet at standard conditions and that volume would weigh 16.04 pounds. Knowing the moles per hour, the MMSCFD may be determined from Eq. (7.13):

$$\text{MMSCFD} = \frac{\text{MPH}(379.4)(24)}{10^6} \tag{7.13}$$

7.2.4.3 SCFM

SCFD, denoting standard cubic feet per minute, is commonly used in compression applications.

7.2.4.3.1 Mass (weight) flow

Mass (weight) flow is expressed as a unit of mass per unit of time, normally pounds-mass per minute ($\text{lb}/_{\text{m}}/\text{min}$). It is a specific value that is independent of gas properties and compressor inlet conditions. It may be given on a wet (water vapor included) or dry basis. The basis used for a particular application may not be correctly communicated to the compressor manufacturer and can result in an improperly sized compressor.

The Mass (Weight) flow, "w," may be calculated from any conditions of interest using Eq. (7.14) (derived from Eq. 7.3):

$$w = \frac{P_1\, Q_1\, M}{10.73\, T_1\, Z_1} \tag{7.14}$$

Mass (weight) flow can also be determined from SCFM:

$$w = \frac{14.7(\text{SCFM})\text{M}}{10.73\ (520)Z_0} = \frac{(\text{SCFM})\text{M}}{379.4\ Z_0} \tag{7.15}$$

Z_0 is often taken as 1.0 regardless of its actual value. It is important to use the same value for Z_0 in all calculations. Although the discrepancy would generally be no more than one or two percent in a single calculation, it could be compounded after conversions are made back and forth by parties involved with the compressor project. It is important to be consistent.

When specifying compressors, it is best to use mass (weight) flow and MMSCFD or SCFD, and to clarify the standard conditions to everyone involved.

Mass (weight) flow can also be determined from Eq. (7.16)

$$w = (53.09)\text{SO}_g \tag{7.16}$$

where:
w = mass (weight) flow, lb_m/min
S = gas specific gravity
Q_g = standard volume flow, MMSCFD
53.09 = constant based on unit conversion and specific volume of the gas

7.2.4.4 ACFM

Actual cubic feet per minute (ACFM) at the inlet of the compressor, often called "Q," is related to the physical size of the compressor. Several design parameters are based on "Q." ACFM at inlet is also abbreviated ICFM (inlet cubic feet per minute). ACFM at the compressor discharge is sometimes of interest, and in this text it will be abbreviated DCFM (discharge cubic feet per minute). However, ICFM is the more appropriate term to use when referring to inlet conditions. In many cases, ACFM is often used interchangeably with ICFM. If there is any doubt, be sure to get a clarification.

SCFM may be converted to ACFM, or "Q," by

$$Q_1 = \text{ACFM} = (\text{SCFM})\left(\frac{14.7}{P_1}\right)\left(\frac{T_1}{520}\right)\left(\frac{Z_1}{Z_0}\right) \tag{7.17}$$

where:
P_1 = absolute pressure, psia
T_1 = absolute temperature, ^{o}R
Z_1 = compressibility at the condition of interest

The actual volume flow is a function of mass (weight) flow and the specific volume of the gas at suction conditions, expressed mathematically as

$$\text{ACFM} = wv_1 \tag{7.18}$$

where:
w = mass flow, lb_m/min
v_1 = specific volume at suction conditions, ft^3/lb_m

The specific volume at suction conditions is calculated using Eq. (7.19):

$$V_s = \frac{13.08}{S} \tag{7.19}$$

where:
V_s = standard specific volume, ft^3/lb_m
S = gas specific gravity
13.08 = specific volume of air, ft3/lbm

and

$$V_1 = V_s Z_s \left(\frac{14.7}{P_s}\right)\left(\frac{T_s}{520}\right) \tag{7.20}$$

where:
V_1 = specific volume at suction conditions, ft^3/lb_m
V_s = standard specific volume, ft^3/lb_m
Z_s = compressibility factor at suction conditions
P_s = absolute suction pressure, psia
T_s = absolute suction temperature, °R
14.7 = standard pressure, psia
520 = standard temperature, °R

Substituting the values of W and V_1 into Eq. (7.18) and *simplifying* we obtain

$$ACFM = 19.6\frac{Z_s T_s Q_g}{P_s} \tag{7.21}$$

where:
ACFM = Actual ft^3/min

Relative humidity. The rated flow through the compressor on the front end of a gas turbine is universally based on ISO conditions.

The ACFM volume flow is important when estimating compressor performance using performance curves

7.2.4.5 Other conventions for standard conditions

Standard conditions of 14.7 psia and 60°F have been referred in this section. This standard is prevalent in the petroleum and natural gas industries in the United States. American Petroleum Institute (API) Standards use these standard conditions. However, in working with air compression systems, "Standard Air" is adopted by American Society of Mechanical engineers (ASME) and is defined as air at a pressure of 14.7 psia, a temperature of 68°F, and a relative humidity of 36%. These conditions correspond to an air density of 0.0750 pounds per cubic foot.

In the metric system, the normal cubic meter per hour is a widely used flow term. Normal refers to conditions of 760 mmHg absolute (14.7 psia) and 0°C (32°F). Weight flow is generally stated in kilograms per hour. The SI system uses kilopascals for pressure (q kPa = 0.145 psi). Other metric units such as kilograms per square centimeter or Newton's per square meter are used.

The matter of standard conditions is further confused by the ISO conditions for base rating a combustion gas turbine. These conditions are 760 mmHg absolute, 15°C, and 60% relative humidity. The rated flow through the compressor on the front end of a gas turbine is universally based on ISO conditions.

7.2.5 Application of compression-thermodynamic theory

7.2.5.1 Introduction

Compression theory is derived from the laws of thermodynamics. The theory is applicable to both positive displacement and continuous flow compressors.

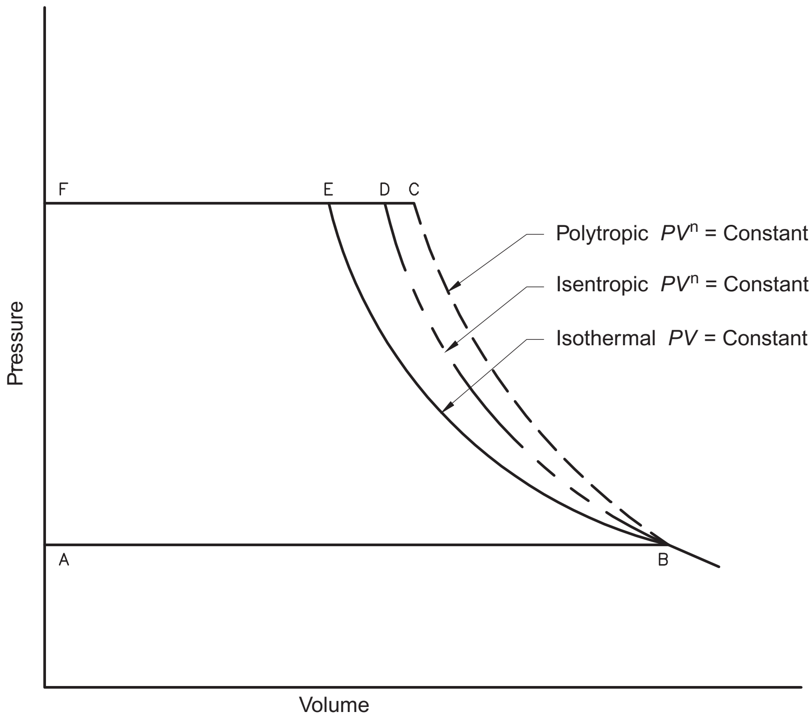


Fig. 7.12 Theoretical compression paths on *P–V* diagram for three different processes.

The two processes used to calculate thermodynamic relationships are isentropic (adiabatic) and polytropic. The polytropic process is a modification of the adiabatic process, involving an efficiency to more closely represent actual conditions. These calculations are the basis for determining capacity, driver size, and mechanical design. The following discussion explains the differences and when they are used.

Fig. 7.12 shows the compression paths of three theoretical processes: isothermal, isentropic, and polytropic. The theoretical work needed for isothermal compression is described by the area ABEF. It can be seen that the isothermal work is appreciably less than that of the isentropic area ABDF. Similarly, the isentropic area is smaller than the polytropic area ABCF. Since the isothermal area is considerably less than the isentropic and polytropic it would be the process for greater compression economy.

None of the three processes is commercially attainable; however, they are useful as a basis for calculations and comparisons.

The differences can be attributed to differences in heat transfer (cooling). The isothermal process would require continuous cooling during compression to negate the entire temperature rise. It is never commercially possible to remove the heat of compression as rapidly as it is generated. Thus, this process is not as logical a working base as the adiabatic although it was used for many years. Compressors are designed for as much heat removal as possible. In an actual compressor the theoretical isentropic discharge temperature can sometimes be achieved by a moderate amount of cooling

Table 7.2 Summary of the characteristics of thermodynamic cycles

Thermodynamic cycle	Property remaining constant	Compressor type
Isothermal	Constant temperature pv = Constant	Ideal cycle—No Commercial Types
Isentropic (adiabatic)	Constant heat content pv^k = Constant	Positive displacement compressors
Polytropic	pv^n = Constant	Dynamic compressors

during compression. Even so, the resultant process will not be purely isentropic due to other losses in an actual machine. The polytropic path BC best represents an actual process where there is no cooling during compression.

In practice, the isentropic and polytropic methods of analysis are both usable for designing and predicting the performance of compressors.

The adiabatic process occurs when there is no heat added or removed from the gas during compression. In practice the isentropic (adiabatic) process is commonly applied to positive displacement compressors, both reciprocating and rotary, since these machines are often equipped with a cooling system that cools the casing or cylinder during compression, making the actual temperature rise approach that of the theoretical adiabatic process.

The polytropic process is a modification of the adiabatic process. The polytropic process is typically applied to dynamic compressors in which there is no cooling during the compression that takes place in any individual stage. (There may be cooling between each stage or series of stages, but not within a given stage.)

Table 7.2 presents a summary of the characteristics of thermodynamic cycles commonly used to describe the different types of compressors.

7.2.5.2 Isothermal compression

In an isothermal process, the temperature is kept constant (unchanged) as the pressure increases during compression. Although it is impossible to build a machine that will compress isothermally, isothermal performance is approached as the number of intercoolers or other cooling devices is increased.

Furthermore, although isothermal compression cannot actually be attained in practice, it is often used as the basis for compression with other compression processes. The effect of the number of coolers on compression power will be covered under Polytropic Compression.

Eq. (7.22) applies to an isothermal compression process:

$$P_1V_1 = P_2V_2 = \text{Constant} \tag{7.22}$$

"Head" is a term used for the work input to the compression process. The units of head are foot-pounds (force) divided by pounds (mass). In general practice, the unit of head is usually taken as "feet." The theoretical head for an isothermal process is:

$$H_{isot} = RT_1 \ln R \tag{7.23}$$

where:

$$R = \frac{P_2}{P_1} \tag{7.24}$$

Eq. (7.24) may be used to evaluate other compression processes with various amounts of cooling.

7.2.5.3 Isentropic (adiabatic) compression

Isentropic means constant entropy (a definition of entropy is beyond the scope of this text). Adiabatic describes a process wherein no heat is added or removed from the gas during compression. For the sake of this discussion, it can be assumed that isentropic and adiabatic are the same (although different thermodynamically). The adiabatic compression process is never exactly obtained since with some units there may be heat losses during part of the cycle and gain in heat during another part. With other units there may be a definite heat gain. However, it is closely approached with most positive displacement units and is generally the base to which they are referred. The adiabatic compression process is commonly assumed for reciprocating but not centrifugal compressors.

In isentropic processes, the following relationship applies:

$$P_1 {V_1}^k = P_2 {V_2}^k = C \tag{7.25}$$

where:
C = constant
k = ratio of specific heats
$= \frac{C_p}{C_v}$

Fig. 7.13 shows a variety of ways in which a gas may expand according to the law of $PV^n = C$ from an initial pressure P_1 in the clearance volume through the stroke to the final volume V_2. Curves lying above the isothermal curve have a value of $n < 1$ and those lying between the isothermal and adiabatic curves have a value of $n > 1$ but less than k. Those curves lying below the adiabatic curve have a value of $n > k$.

7.2.5.3.1 Adiabatic head

In order to change a volume of gas from one pressure to a higher pressure, work or energy must be added to the gas. The compression ratio is the proportional comparison of the absolute inlet (suction) and outlet (discharge) pressures. It is a dimensionless number that indicates the amount of energy needed and is used in many of the compressor equations. For positive displacement compressors, energy is added by a reciprocating or piston moving back and forth in a cylinder. For centrifugal (radial) and

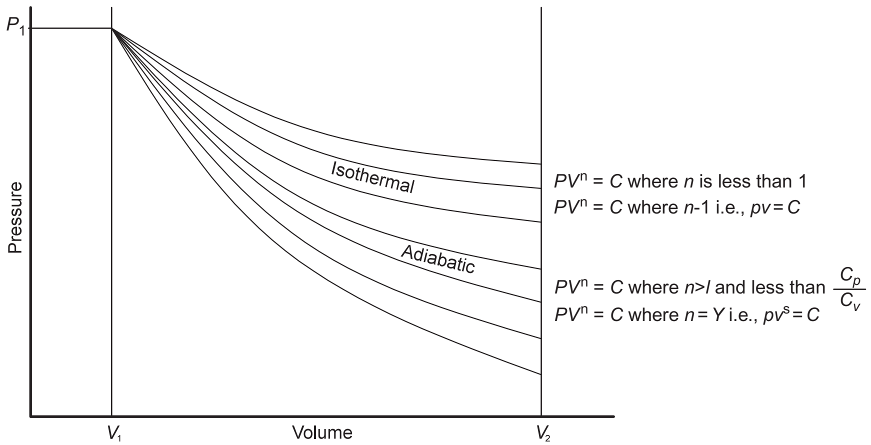


Fig. 7.13 Isothermal and Adiabatic expansion and compression of a gas.

axial compressors, energy is added via acceleration of the gas by rotating impellers. To accomplish the required pressure ratio, the compression produces a certain head. In an ideal process, the required pressure increase is achieved without losses and all work done by the impeller is converted to pressure energy.

The isentropic (adiabatic) process is shown in Fig. 7.14. The adiabatic process is isentropic with constant entropy which is a special case of the more general polytropic process. The adiabatic head is a function of

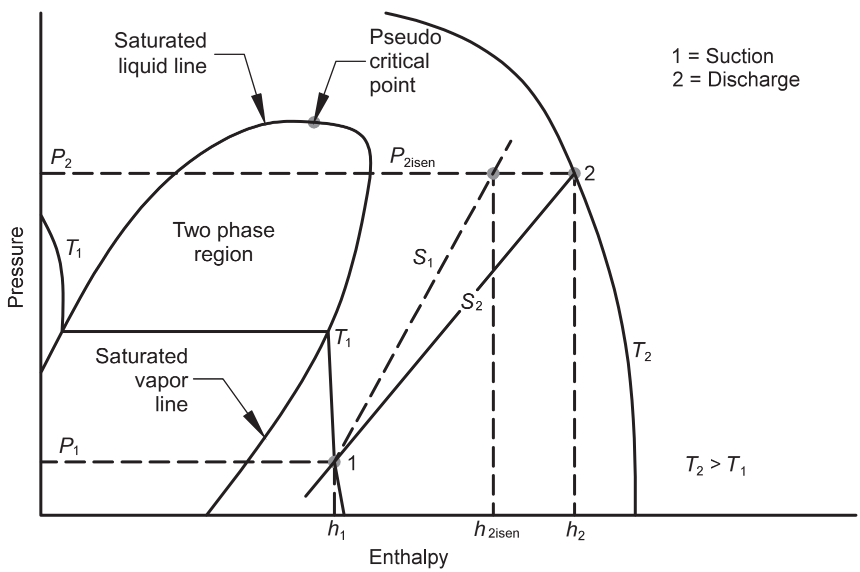


Fig. 7.14 Adiabatic (isentropic) compression process.

- Available suction pressure (P_1)
- Required discharge pressure (P_2)
- Suction gas temperature (T_1)
- Ratio of specific heats (k)
- Gas compressibility (Z_{AVG})
- Gas specific gravity (S)

Adiabatic head is determined from the following equations:

$$H_{ad} = RT_1 \frac{\left(R_t^{\frac{k-1}{k}} - 1\right)\left(\frac{Z_1 + Z_2}{2}\right)}{\frac{k-1}{k}} \tag{7.26}$$

where:
H_{ad} = adiabatic head, ft.

Adiabatic head can also be written as follows:

$$H_{ad} = 53.3 Z_{avg} \left[\frac{T_1}{S}\right]\left[\frac{k}{k-1}\right]\left[\frac{P_2}{P_1}\right]^{\left[\frac{k-1}{k}\right]} - 1 \tag{7.27}$$

where:
H_{ad} = adiabatic head, ft-lb/lb
Z_{avg} = average compressibility
$= \frac{(Z. + Z_2)}{2}$
T_1 = suction temperature, °R
k = ratio of specific heats, $\frac{C_p}{C_v}$
P_2 = discharge pressure, psia
P_1 = suction pressure, psia

7.2.5.3.2 Adiabatic discharge temperature

For convenience, the factor "x" is defined as follows:

$$x = r^{\frac{k-1}{k}} - 1 \tag{7.28}$$

The theoretical adiabatic discharge temperature, assuming 100% adiabatic efficiency, is determined from the following equation:

$$T_{2(\text{theo})} = T_1(x + 1) \tag{7.29}$$

where:
$T_{2(\text{theo})}$ = theoretical adiabatic discharge temperature, °R

The actual adiabatic discharge temperature is determined from the following equation:

$$T_2 = T_1\left(1 + \frac{x}{\eta_{ad}}\right) \tag{7.30}$$

where:
T_2 = actual discharge temperature, °R
η_{ad} = adiabatic efficiency

7.2.5.3.3 Adiabatic gas horsepower

The adiabatic gas horsepower is determined from the following equations:

$$\mathrm{GHP} = \frac{wH_{\mathrm{ad}}}{33{,}000\eta_{\mathrm{ad}}} \tag{7.31}$$

where:
GHP = gas horsepower

The break horsepower (BHP), which accounts for mechanical efficiency, can be determined from the following equation:

$$\mathrm{BHP} = 0.0857(Z_{\mathrm{AVG}})^{\frac{1}{k}}[Z_1]^{\frac{k-1}{k}}\left[\frac{(Q_g)(T_1)}{\eta_m\eta_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{kn_p}} - 1\right] \tag{7.32}$$

where:
BHP = brake horsepower per stage
Q_g = volume of gas, MMSCFD
T_s = suction temperature, °R
Z_1 = suction compressibility factor
Z_{D} = discharge compressibility factor
η_m = mechanical efficiency
high-speed reciprocating units
- use 0.93 to 0.95

low-speed reciprocating units
- use 0.95 to 0.98

centrifugal units
- use 0.95

η_{ad} = adiabatic efficiency (isentropic efficiency) reciprocating units
- use 0.85 to 0.90

centrifugal units
- use 0.72 to 0.85

k = ratio of specific heats, c_p/c_v
n_p = polytrophic efficiency
= 1.0 for reciprocating compressors
= 0.8 for centrifugal compressors

P_1 = suction pressure of stage, psia
P_2 = discharge pressure of stage, psia
$Z_{ay} = \frac{(Z_1 + Z_2)}{2}$

Notice that Eqs. (7.26), (7.32) have been corrected by an average compressibility, $\frac{(Z_1+Z_2)}{2}$. Averaging is a fairly accurate approximation of the correction required.

Because of the nonideal (nonperfect) behavior of most gases, the "k" exponent does not remain constant during compression. For air, diatomic gases, and inert gases, the change in the "k" is small when the pressures are moderate. However, for most hydrocarbon gases, the variance of "k" during compression is substantial. The usual correction is to calculate "k" using MC_p (see Eq. 7.7) at the average of the compressor (or stage) suction and discharge temperature.

Using the MC_p at atmospheric pressure and average temperature for compressor head and power calculations is sufficiently accurate for most applications. However, for very high pressures or other unusual conditions, further corrections are necessary. Such corrections are covered under Polytropic Compression (see Section 7.2.5.4)

7.2.5.3.4 Adiabatic efficiency

Regardless of the thermodynamic process considered, the compressor must produce enough head to supply the required compression ratio in addition to compensating for mechanical losses. The efficiency is the ratio of the work output to the work input. The work output per mass is the head required to achieve the desired pressure ratio. Work input is the power required from the driver. The power required from the driver will be the same whether the compression process is considered adiabatic or polytropic, thus the work input will be the same regardless of the process considered.

Since the change in entropy is not zero in an actual adiabatic compression process, an adiabatic efficiency (η_{ad}) is used in Eqs. (7.30), (7.31), and (7.32). In order to calculate MC_p at average compression temperature, it is necessary to estimate the adiabatic efficiency to arrive at a discharge temperature as per Eq. (7.30). If the estimate is inaccurate, a second iteration may be required.

7.2.5.3.5 Thermodynamic diagrams

Thermodynamic property diagrams account directly for deviations of a real gas from ideal relationships. These diagrams are a plot of gas properties, commonly including enthalpy, entropy, pressure, and temperature. Occasionally, a special diagram is developed for a widely used gas mixture such as a refrigerant. However, a note that few charts are available for mixtures and this method is therefore not commonly used for hydrocarbon mixtures.

When a diagram is used to predict changes of state during compression, compressibility and variance of "k" are not needed because these variables are already factored into the diagrams. In general, then, this method is more accurate than Eq. (7.26), and when charts are available, it is certainly more convenient.

Diagrams are often used in compressor calculations for heavier hydrocarbon gases such as propane and propylene that tend to deviate considerably. Diagrams for many pure gases are well established.

The following equations pertain to the use of diagrams for compressor calculations. Note that for an isentropic process, there is no change in entropy, "*S*."

$$S_{2(\text{theo})} = S_1 = 0 \tag{7.33}$$

where:
$S_{2(\text{theo})}$ = S_1 = entropy at suction conditions

The theoretical change in enthalpy is defined as follows:

$$\Delta h_{(\text{theo})} = h_{2(\text{theo})} - h_1 \tag{7.34}$$

where:
h_1 = enthalpy at suction conditions, BTU/lb
$h_{2(\text{theo})}$ = theoretical enthalpy at discharge pressure S_1, Btu/lb

The actual enthalpy at discharge pressure and temperature is defined as follows:

$$h_2 = \frac{\Delta h_{(\text{theo})}}{\eta_{\text{ad}}} + h_1 \tag{7.35}$$

where:
h_2 = actual enthalpy at discharge pressure and temperature, Btu/lb
η_{ad} = adiabatic efficiency

Note that the actual discharge temperature "T_2" may now be found on the thermodynamic diagram at the point corresponding to h_2 and P_2. The adiabatic head (H_{ad}) can be determined from the following equation:

$$H_{\text{ad}} = (778)\Delta h_{(\text{theo})} \tag{7.36}$$

7.2.5.4 Polytropic compression

Polytropic compression is commonly assumed for dynamic (centrifugal (radial) and axial) compressors.

As discussed previously, the adiabatic process showed that its relationships need mathematical corrections to make creditable predictions. The corrections are compromises between theory and actual gas deviations, and they do not always yield sufficiently accurate predictions for some types of applications. Unfortunately, even this process requires adjustments to account for the nonideal behavior of most gases.

The polytropic compression process is defined mathematically as follows:

$$P_1V_1{}^n = P_2V_2{}^n = C \tag{7.37}$$

where:
n = polytropic exponent

The polytropic exponent "n" is experimentally determined for a given type of machine and may be lower or higher than the adiabatic exponent "k." In positive displacement and internally cooled dynamic compressors, $n < k$. Although "n" actually changes during compression, an average or effective value, calculated from experimental information, is used.

7.2.5.4.1 Polytropic efficiency

The polytropic efficiency is determined from the following equation:

$$\eta_p = \frac{\dfrac{k-1}{k}}{\dfrac{n-1}{n}} \tag{7.38}$$

where:
η_p = polytropic efficiency

In Eq. (7.38), "k" is ordinarily taken at the average compression temperature by most compression manufacturers. Thus, when estimating overall flange-to-flange performance, one should use "k" at the average flange-to-flange temperature to yield results very close to those of stage-to-stage calculations. In the case of single-stage units, the difference between "k" at inlet temperature and average temperature is generally very small. Therefore, in this text, "k" at average compression temperature will be used.

7.2.5.4.2 Polytropic head

The polytropic head is determined from the following equations:

$$H_{\text{poly}} = \text{RT}_1 \frac{\left(r^{\frac{n-1}{n}} - 1\right)}{\dfrac{n-1}{n}} \left(\frac{Z_1 + Z_2}{2}\right) \tag{7.39}$$

where:
H_{poly} = polytropic head, ft.

The polytropic head can also be determined from the following equation:

$$H_p = 53.3 Z_{AVG} \left[\frac{T_1}{S}\right] \left[\frac{kn_p}{k-1}\right] \left[\frac{P_2}{P_1}\right]^{\left[\frac{k-1}{kn_p}\right]-1} \tag{7.40}$$

where:
n_p = Polytrophic efficiency
= 1 for reciprocating compressors
= 0.72 to 0.8 for centrifugal compressors

A thermodynamic

7.2.5.4.3 Polytropic discharge temperature

The polytropic discharge temperature is determined from the following equation:

$$T_2 = T_1 r^{\frac{n-1}{n}} \tag{7.41}$$

7.2.5.4.4 Polytropic gas horsepower

The polytropic gas horsepower is determined from the following equations:

$$\text{GHP} = \frac{wH_{\text{poly}}}{33{,}000\eta_p} \tag{7.42}$$

The polytropic break horsepower (BHP), which accounts for mechanical losses, may be determined from the following equation:

$$\text{BHP} = 0.0857(Z_{AVG})^{\frac{1}{k}}[Z_1]^{\frac{k-1}{k}}\left[\frac{(Q_g)(T_1)}{\eta_m\eta_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{kn_p}} - 1\right] \tag{7.43}$$

where:
BHP = Brake horsepower per stage
Q_g = Volume of gas, MMSCFD
T_1 = Suction temperature, °R
Z_1 = Suction compressibility factor
Z_2 = Discharge compressibility factor
η_m = Mechanical efficiency
High-speed reciprocating units
– use 0.93 to 0.95
Low-speed reciprocating units
– use 0.95 to 0.98
Centrifugal units
– use 0.95
η_a = Adiabatic efficiency (isentropic efficiency)
Reciprocating units
– use 0.85
– use 0.72 to 0.85
k = Ratio of specific heats, c_p/c_v
n_p = Polytrophic efficiency
= 1.0 for reciprocating compressors
= 0.8 for centrifugal compressors
P_1 = Suction pressure of stage, psia

P_2 = Discharge pressure of stage, psia
$Z_{av} = (Z_s + Z_D)/2$

7.2.5.4.5 Thermodynamic diagrams

A thermodynamic diagram can be used for a polytropic calculation by first determining the adiabatic head (H_{ad}) using Eqs. (7.34), (7.36). Polytropic head (H_{poly}) can then be determined by:

$$H_{poly} = H_{ad}\left(\frac{\eta_p}{\eta_{ad}}\right) \tag{7.44}$$

The relationship between polytropic and adiabatic efficiencies is:

$$\eta_{ad} = \frac{r^{\frac{k-1}{k}} - 1}{r^{\left(\frac{k-1}{k\eta_p}\right)} - 1} \tag{7.45}$$

This relationship is graphically represented in Fig. 7.15.

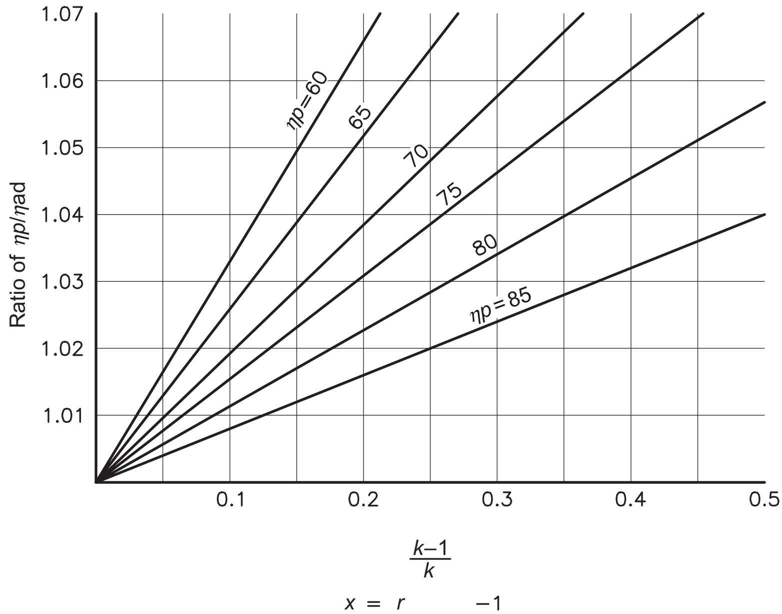
From the previous discussion, it should be obvious that "k" is not equal to "η." In some early compressor tests, the "k" and "η" exponents were erroneously treated as the same. This error probably was one of nomenclature. In any case, it is important to recognize that "k" is associated with the adiabatic process while "η" is associated with the polytropic process.

7.2.5.5 Miscellaneous considerations and clarifications

A number of gases have large deviations from ideal behavior near their critical conditions or at high temperatures. For example, carbon dioxide (CO_2) at 1500 psia and 100°F has a compressibility factor, "Z," of about 0.27. Furthermore, if the temperature is increased by only 20°F, there is a 40% increase in the compressibility factor. If a small amount of methane (CH_4) is mixed with carbon dioxide, the compressibilities change significantly, and predictions of these compressibilities by generalized charts and graphs are not reliable.

There are a few compressor applications that deal with widely deviating gases. The values for "Z" and "k" can vary so much that conventional methods of calculations for the compressor gas properties do not have sufficient accuracy. For these rare occasions, various equations of state are used. There are a number of these empirical relationships in existence, and each set of relationships tends to have some advantages over the other sets for certain gas compositions.

Several equations of state that have been developed to predict thermodynamic properties include the Benedict-Webb-Rubin, Peng-Robinson, Redlich-Kwong, and Martin-Hou equations. Typical gas compositions to which these correlations are applied include ultrahigh pressure hydrocarbons, high-pressure carbon dioxide, some refrigerants, and some chemical plant gases. These relationships are generally processed with a computer. A description of these complex correlations is beyond



Example :
Given :

P_1 = 100 psia
P_2 = 305 psia
k = 1.25
Poly. eff. = 0.70

$R = \frac{300}{500} = 3.05$, $\frac{k-1}{1} = \frac{0.25}{1.25} = 0.20$

$X = (3.05)^{0.2} - 1 = 0.25$

Enter curve @0.25 and poly eff. = 0.70 ; read ratio of 1.05

Adiabatic eff. $= \frac{0.70}{1.05} = 0.67$

Fig. 7.15 Adiabatic and polytropic efficiency relationships.

the scope of this text. To meet our needs, when dealing with compressor applications where the gas deviations are very large, advice should be sought from a process engineer with experience in the use of these equations of state. (Refer to Volume 2 of *Surface Production Operations Gas Handling Facilities* for a more detailed discussion of the various equations of state.)

7.3 How to select a compressor

7.3.1 Preliminary selection considerations

The items that must be determined before a preliminary compressor selection can be made include:

- Number of stages
- Brake horsepower
- Type

Information needed for selection

Q_g = volume of gas to be compressed, MMSCFD
P_1 = suction pressure, psia
P_2 = discharge pressure, psia
T_1 = suction temperature, °R (460 + °F)
S = gas specific gravity (air = 1)

7.3.2 Approximating the number of stages

A first approximation can be made assuming a maximum compressor ratio per stage of 3.5. The compression ratio per stage is given by Eq. (7.46):

$$R = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \tag{7.46}$$

where
R = compression ratio per stage
P_2 = required discharge pressure, psia
P_1 = required suction pressure, psia
N = number of stages

Compression ratios are normally between 1.2 and 5.0

The compressor ratio is limited to 3.5 so as to limit the discharge temperature due to the following temperature limitations:

Packing life = 250°F to 275°F
Lube oil degradation = 300°F
Ignition if oxygen present = 300°F
Maximum = 350°F to 400°F

Fig. 7.16 shows the pressure-volume curve for both single- and two-stage compression (neglecting interstage losses). Cooling the gas from A to D, before beginning the compression cycle in the second stage, results in the gas at Point C to be cooler than at Point B. The area under the curve reduced by an amount equal to ABCD represents the power saved by adding the second stage.

7.3.3 Approximating the brake horsepower (BHP)

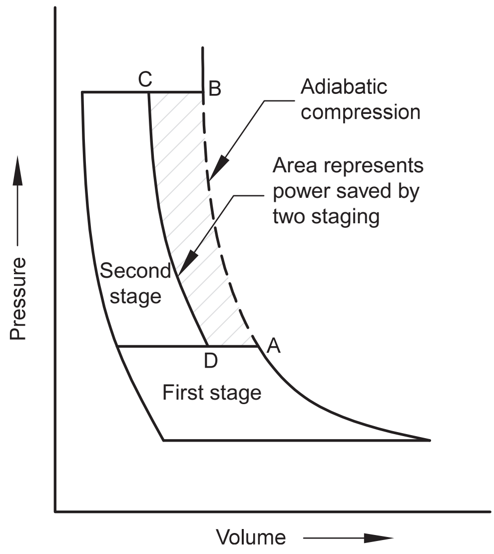
The brake horsepower can be approximated from either the

- GPSA "Quick Look" approximation curve
- "Approximate" BHP Formula

7.3.3.1 GPSA "quick look" approximation curve

The GPSA"Quick Look" approximation curve is based on the overall compression ratio (R_t) and ratio of specific heats (k). Fig. 7.17 is a "Quick Look" BHP

Fig. 7.16 Horsepower reduction by multistaging, neglecting interstage friction losses.



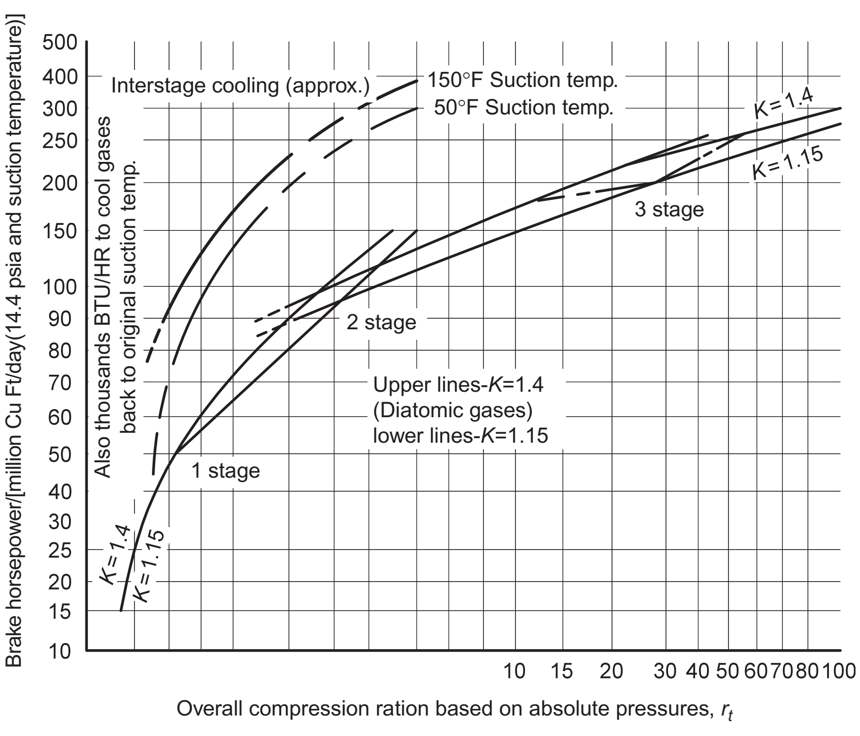


Fig. 7.17 “Quick look” brake horsepower (BHP) approximation.
Courtesy of GPSA Engineering Data Book.

Approximation curve based on a combination of theory and test data. The procedure to use the curve is as follows:

- Determine *k-value*
- Calculate the *total* compression ratio (R_t)
- Determine the *BHP/MMSCF* from Fig. 7.17
- *Correct* for suction temperature and pressure base
- *Calculate* BHP required

7.3.3.1.1 Determination of ratio of specific heats "*k*-value"

"*k*" is the ratio of the specific heat of a gas of constant pressure to the heat capacity at a constant volume. For routine calculations, an average $k = 1.25$ can be used for natural gas. Fig. 7.18 is a curve that can be used to approximate the k-value for hydrocarbons. The molecular weight of a gas can be determined from the following:

$$(MW)_g = (28.96)(S)_g \tag{7.47}$$

7.3.3.1.2 Calculate total compression ratio (R_t)

The "total" compression ratio should not result in a discharge temperature greater than 300°F (149°C) unless high rates of parts wear are acceptable. The total compression ratio is determined from the following equation:

$$R_t = \frac{P_2}{P_1} \tag{7.48}$$

Common practice is to divide the total pressure rise into stages of compression, with intercooling, thus reducing the temperature on the unit.

7.3.3.1.3 Determine BHP/MMSCF from Fig. 7.17

Enter Fig. 7.17 with the overall compression ratio, R_t, and intersect with the ratio of specific heats, k, and read the approximate BHP/MMSCFD required. The BHP/MMSCFD value in the curve is based on a 14.40 psia pressure base and a 60°F suction temperature.

7.3.3.1.4 Correct for suction temperature and pressure base

The values from the curve must be modified for actual pressure base and correct suction temperature and temperature base. The correction can be made using the following equation:

$$(BHP/MMCF)_c = \left(\frac{BHP}{MMCF}\right)\left(\frac{P_b}{14.40}\right)\left(\frac{T_s}{520}\right) \tag{7.49}$$

where:
$(\text{BHP/MMSCF})_c$ = Corrected BHP/MMSCF

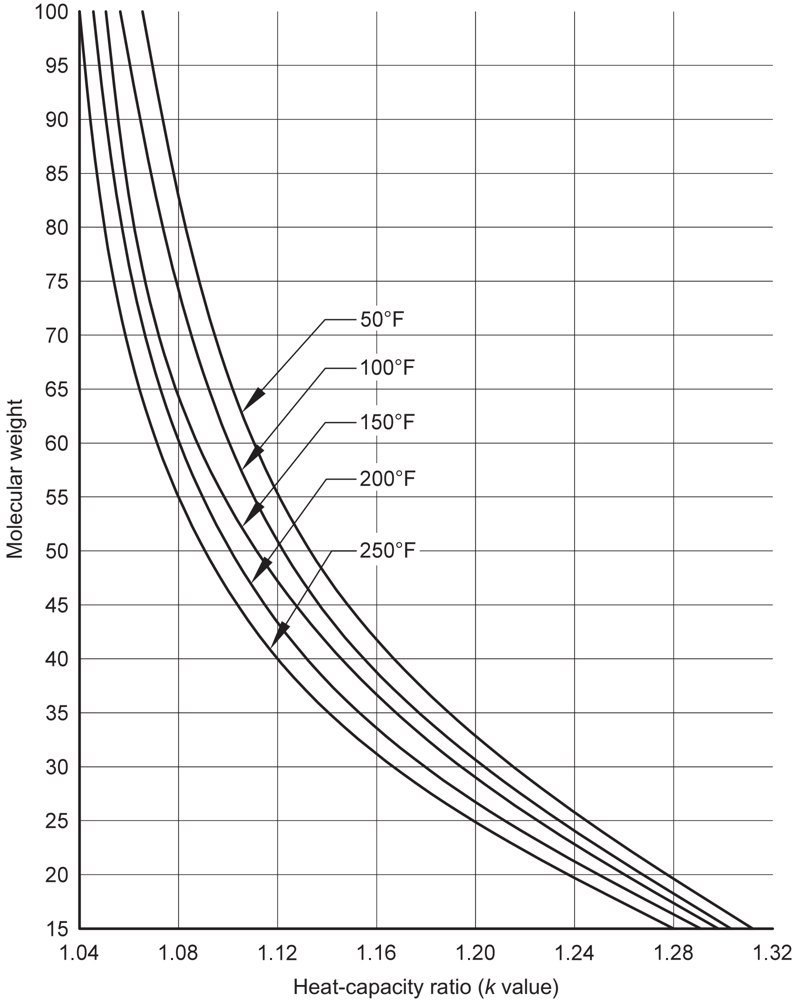


Fig. 7.18 Approximation of ratio of specific heats (*k*-value).
Courtesy of GPSA Engineering Data Book.

P_b = Actual pressure base, psia
T_s = Suction temperature, °R

7.3.3.1.5 Altitude correction

Once the brake horsepower has been determined, one must determine if a correction must be made due to altitude. Above 1500-feet altitude, gas engines must be derated from name plate horsepower ratings at sea level because of lower atmospheric pressure. At a given rpm, less air is available to burn the fuel. Turbocharged engines are

Table 7.3 Efficiency multiplier for specific gravity

R	Specific gravity				
	1.5	**1.3**	**1.0**	**0.8**	**0.6**
2.0	0.99	1.0	1.0	1.0	1.01
1.75	0.97	0.99	1.0	1.01	1.02
1.5	0.94	0.97	1.0	1.02	1.04

Table 7.4 Efficiency multiplier for low inlet pressure

R	Inlet suction pressure							
	10	**14.7**	**20**	**40**	**60**	**80**	**100**	**150**
3.0	0.990	1.00	1.00	1.00	1.00	1.00	1.00	1.00
2.5	0.980	0.995	0.990	0.995	1.00	1.00	1.00	1.00
2.0	0.960	0.965	0.970	0.980	0.990	1.00	1.00	1.00
1.5	0.890	0.900	0.920	0.940	0.960	0.98	0.980	1.00

less susceptible to this derating as they compress the air to a higher pressure for burning. Curves used for derating engines for altitude can be found in GPSA Engineering Data Book.

7.3.3.1.6 Specific gravity and low inlet pressure corrections

In addition to a possible altitude correction, one must determine if a specific gravity and low inlet pressure correction is required. The following corrections (Tables 7.3 and 7.4) should be made to calculate BHP. These corrections become more significant at lower pressure ratios.

7.3.3.2 Approximate BHP formula

Eq. (7.50) is a simplified formula to "approximate" the BHP

$$BHP = 22\ R\ n\ F\ Q_g \tag{7.50}$$

where:

$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \tag{7.51}$$

where:
BHP = approximate brake horsepower
R = ratio per stage
R_t = total compressor ratio

n = number of stages
F = allowance for interstage pressure drop
= 1.00 for single-stage compression
= 1.08 for two-stage compression
= 1.10 for three-stage compression
P_d = discharge pressure, psia
P_s = suction pressure, psia
Q_g = flow rate, MMSCFD

Choose the number of stages to give a value for R less than 3.5.

7.3.3.3 Selecting the type of compressor

Once the BHP and the number of stages are estimated, the compressor type can be determined. Items to consider include:

- Horsepower
- Flow rate
- Compression ratio
- Foundation conditions
- Fuel cost
- Waste heat requirements
- Types of drivers available
- Spare parts availability

Table 7.5 lists some typical compressor type selections. Fig. 7.19 can also be used to select a compressor type.

Table 7.5 Example of compressor type selections

Service	Flow rate MMSCFD	R	n	Approx. BHP	Most likely	Selection alternate
Booster	100	2.0	1	4400	Centrifugal	Integral (onshore only)
	10	2.0	1	440	High speed	
Gas lift	5	2.7	3	980	High speed	
	20	2.7	3	3920	Centrifugal	Integral (onshore only)
	100	2.7	3	19,602	Centrifugal	
Flash gas	2	2.0	1	88	Screw	High speed
	2	2.0	2	190	High speed	Screw
	4	2.0	2	380	High speed	
Vapor recovery	0.1	4.0	1	9	Vane	Screw
	1.0	3.0	2	143	Screw	High speed
	2.0	3.0	2	286	High speed	Screw

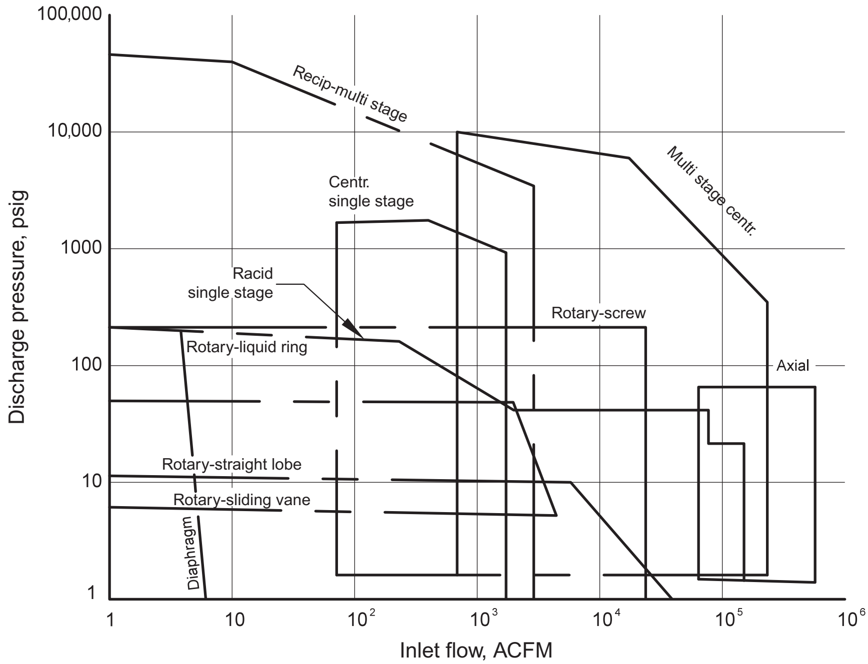


Fig. 7.19 Compressor selection chart.

7.3.4 Procedure for more accurate determination of number of stages and horsepower

The procedure to determine a more accurate determination of the number of stages and brake horsepower is shown as follows:

1. Calculate overall compression ratio ($R_t = P_d/P_s$)
2. Use one stage if:
   - Overall compression ratio is less than 5 and
   - Suction gas temperature is under 110°F (43°C)
3. Select initial number of stages such as $R_t < 5$

Assume compression ratio for each stage is equal (an adjustment may have to be made for sidestreams)

4. Calculate discharge gas temperature for the first stage
   - If discharge temperature is too high (over 300°F), select greater number of stages and recalculate compression ratio per stage
   - If discharge temperature is acceptable (220–300°F range) calculate horsepower
5. Calculate suction pressure, discharge temperature, and horsepower for each succeeding stage

If R is greater than 3, repeat the calculations assuming one additional stage.

7.3.4.1 Determination of gas discharge temperature

The gas discharge temperature can be determined from the following equation:

$$T_d = T_s \left(\frac{P_d}{P_s}\right) \frac{k-1}{kn_p} \tag{7.52}$$

where:
T_d = Gas discharge temperature, °R
T_s = Gas suction temperature, °R
P_d = Gas discharge pressure, psia
P_s = Gas suction pressure, psia
k = Ratio gas specific heats, c_p/c_v
n_p = Allowance for irreversibility and friction effects (polytropic efficiency)
= 1.0 for reciprocating compressors
= 0.8 for centrifugal compressors

Fig. 7.20 provides an approximation for the gas discharge temperature based on various values of ratio of specific heats "k" and total compression ratio "R."

Discharge temperature can be changed by

- Adding or deleting stages
- Cooling suction or inter-stage gas

7.3.4.2 Accurate determination of brake horsepower

The total brake horsepower required is equal to the sum of the required horsepower for each stage. The brake horsepower can be determined from the following equation:

$$\text{BHP} = 0.0857(Z_{\text{AVG}})^{\frac{1}{k}}[Z_1]^{\frac{k-1}{k}}\left[\frac{(Q_g)(T_1)}{\eta_m \eta_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{kn_p}} - 1\right] \tag{7.53}$$

where:
BHP = brake horsepower per stage
Q_g = volume of gas, MMSCFD
T_1 = suction temperature, °R
Z_1 = suction compressibility factor
Z_2 = discharge compressibility factor
η_m = mechanical efficiency
high-speed reciprocating units
 – use 0.93 to 0.95
low-speed reciprocating units
 – use 0.95 to 0.98
centrifugal units
 – use 0.95
η_a = adiabatic efficiency (isentropic efficiency)
reciprocating units
 – use 0.85 to 0.90

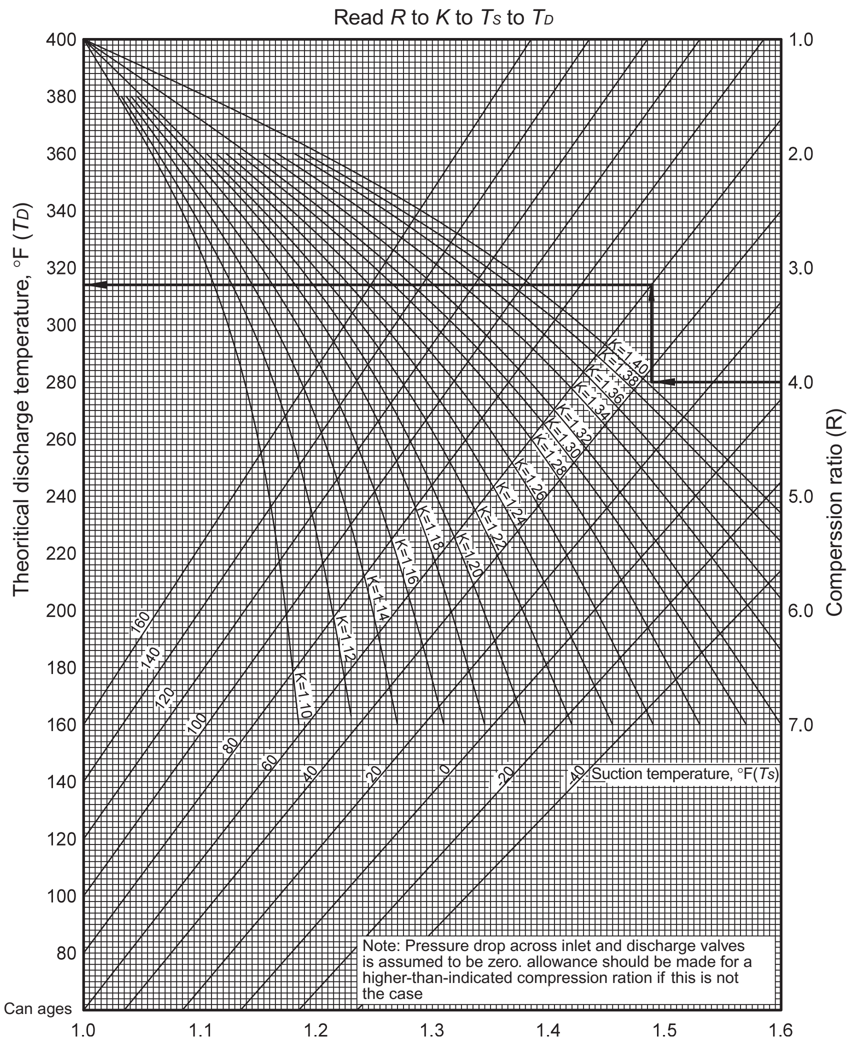


Fig. 7.20 Graph to approximate the gas discharge temperature.

centrifugal units

– use 0.72 to 0.85

k = ratio of specific heats, c_p/c_v

n_p = polytrophic efficiency

= 1.0 for reciprocating compressors

= 0.8 for centrifugal compressors

P_1 = suction pressure of stage, psia

P_2 = discharge pressure of stage, psia

$Z_{ayg} = \frac{(Z_1 + Z_2)}{2}$

7.3.4.3 Determination of interstage pressure loss

Standard industry practice is to assume 3% of absolute pressure loss from lower stage discharge to upper stage suction.

Example 7.1. Calculation of interstage pressure drop
Given:
First-stage discharge pressure = 100 psig
Determine:
Calculate second-stage suction pressure
Solution:
Calculate first-stage discharge pressure drop

$$\begin{aligned}
&\text{Pressure loss} = \text{Discharge pressure} \times 3\%\ \text{Pressure loss}\\
&= (100)(0.03)\\
&= 3\,\text{psig}
\end{aligned}$$

Thus, second-stage suction pressure

$$\begin{aligned}
&= \text{First discharge pressure} - \text{Pressure loss}\\
&= 100 - 3\\
&= 97\ \text{psig}
\end{aligned}$$

7.3.4.4 Compressor process, safety, and piping considerations

Typical process and safety considerations are illustrated in Fig. 7.21.

- An inlet shutdown valve and outlet check valve are required to isolate the compressor and minimize back flow.
- Outlet shutdown valves are also required for offshore, plant, and major onshore installations.
- Automatic blowdown valves are required to relieve trapped pressure.
- A flare valve is required on the suction and each sidestream to keep pressures from rising too high.

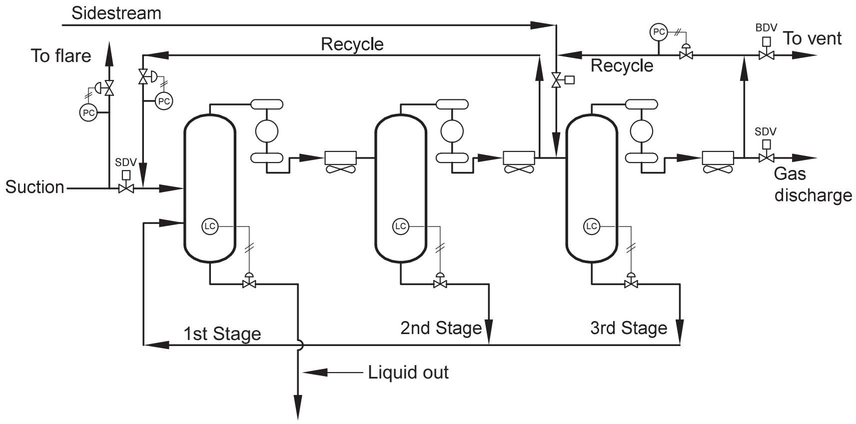


Fig. 7.21 Typical three-stage reciprocating compressor flow diagram.

- A recycle valve is required from the discharge to suction and each sidestream to keep pressure in any one stage from falling too low.
- Relief valves are needed on each discharge in case of ice blockage in a cooler.

Piping considerations include:

- Piping smaller than 2" should not be used for main gas piping
- Do not use threaded pipe where vibrations from engines or reciprocating compressors are expected
- Piping should be arranged so that compressor is accessible from all sides for maintenance
- Suction connection at each compressor stage should be provided with a cone-shaped screen. Screen should be pointed upstream and should be removed after start-up.
- Provide manual drains for suction and discharge bottles if gas condensation can be expected when unit shuts down and cools off
- Provide insulation for engine or turbine exhaust piping. Do not locate exhaust piping where hydrocarbon liquids can be spilled on it.

7.4 Design considerations

7.4.1 Overview

Compressor sizing can involve several levels of detail as defined as follows:

Level 1: Specifying the flow rate, pressure rise, and gas composition, leaving the machine design and selection to the manufacturer. This is the quickest, simplest approach, but it may not involve the company in important design details. This level is not commonly used.

Level 2: Close estimations (typically +/−10%) based on empirical correlations of energy consumption, temperature rise, and installation cost. Several sources exist and are commonly used by producing locations to size reciprocating compressors.

Level 3: A detailed analysis based on the equations developed in this text. This may be helpful in selecting the most effective category of compressor (reciprocating, centrifugal, etc.), in cases where the category has not been determined. These equations can be used to estimate energy requirements and temperatures.

Level 4: Rigorous, detailed analysis and calculations to determine compression energy, power required, gas temperatures, pressure rise, and mechanical stresses. These analyses are done by equipment vendors and are beyond the scope of this text.

7.4.2 Compressor duty

The first step in specifying compressors is to define the required compressor duty. This includes defining the following:

- Flow rate
- Gas composition
- Suction pressure
- Suction temperature
- Discharge pressure

These conditions should be confirmed and the variability of the conditions determined. In most applications, future, or alternate operating conditions can significantly affect the sizing and characteristics of the compressor system. The following is a list of typical changes affecting compressor duty:

1. Buildup in discharge pressure of gas injection compressors as the resistance of the oil field formation increases with time or as more compressors are added to the system.
2. Increase in flow rate and change in molecular weight of gas gathering system as an oil field's gas-to-oil ratio increases with time.
3. Large change in molecular weight, flow, and pressures during periodic catalyst regeneration in a process plant.
4. Increase in system pressure drop due to fouling of equipment during a run.
5. Change in feedstock to a process unit.
6. Seasonal changes of ambient temperature and cooling water temperature.
7. Start-of-run (SOR) versus end-of-run (EOR) conditions in a refining unit.
8. Routine turndown of compressor capacity.

Initially, one must accurately determine the pressure/flow envelope where the compressor operates. If available, review the process flow diagram and a pressure profile in detail with operating, or project representative to assure that all factors have been considered. If a process flow diagram is unavailable, make a sketch of the flow circuit. If one questions the pressure drop allowances, review the calculations with the appropriate operations personnel.

It is important that one accurately determine the suction pressure. For example, if a four-stage reciprocating compressor is sized for compression from 20 to 4400 psia, and the initial suction pressure turns out to be 25 psia, the actual weight flow will be 25% higher, and the horsepower required 19% higher than predicted.

7.4.3 System resistance and characteristic (performance) curves

A system resistance curve can be drawn after the pressure profile is accurately determined. This curve plots the system discharge pressure or head versus inlet volumetric flow, and reflects the complete piping system friction losses. Each point on the curve shows the head or pressure required to deliver an amount of flow through the piping system.

The system resistance curve can be superimposed on the compressor's actual characteristic (performance) curve to show the point at which the compressor will operate. Chapters 9 and 10 provide more detailed information. The following provides an introduction to system resistance and compressor characteristic curves.

7.4.3.1 System resistance curve

The system resistance met by a compressor can vary from a constant discharge pressure to a variable relationship as shown in the generalized system resistance curves in Fig. 7.22.

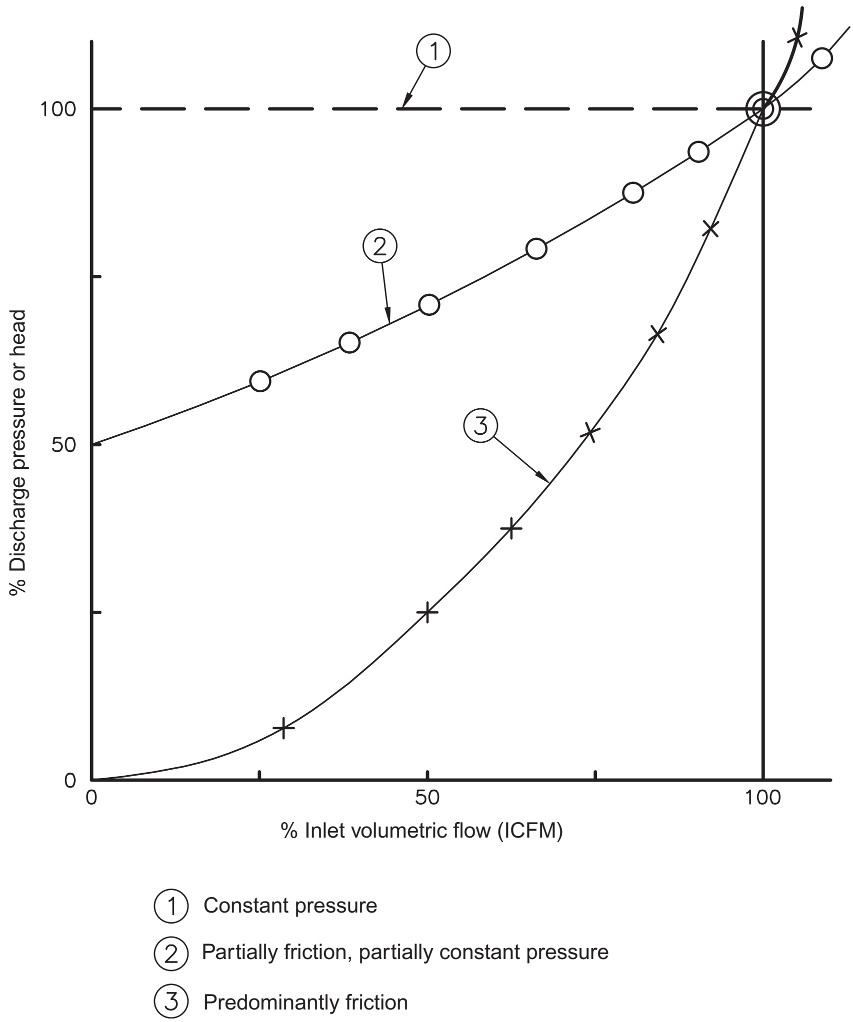


Fig. 7.22 System resistances.

Examples of the three types of curves are as follows:

Type 1: Constant Pressure

- Gas lift
- Gas injection
- Refrigeration
- Plant/Air instrument Air

Type 2: Combined Constant Pressure and Friction

Process with piping and equipment plus constant pressure drop through catalyst bed.

Type 3: Predominately Friction

Systems with only piping, fittings, and equipment such as heat exchangers.

7.4.3.2 Compressor characteristic (performance) curve

General characteristic curves for the four types of compressors at constant speed are shown in Fig. 7.23.

Centrifugal compressor capacities can be controlled by causing the compressor to move back on its characteristic curve. This is generally accomplished by throttling the discharge valve. Speed control and external recycle are also common control methods for centrifugal compressors. It may seem that speed control would be the logical control method for reciprocating compressors. However, large low-speed machines are generally driven by large synchronous motors at fixed speeds. The primary control methods are external recycle and unloaders, which either hold open the suction valves or add clearance volume to the cylinder. Table 7.6 presents the generalized control methods that fit each compressor type.

7.4.3.3 Gas analysis

A gas analysis should be provided for all present and future operating conditions and should be checked for the presence of sulfides, chlorides, or other corrosive agents. These contaminants can have a major effect on the selection of the materials

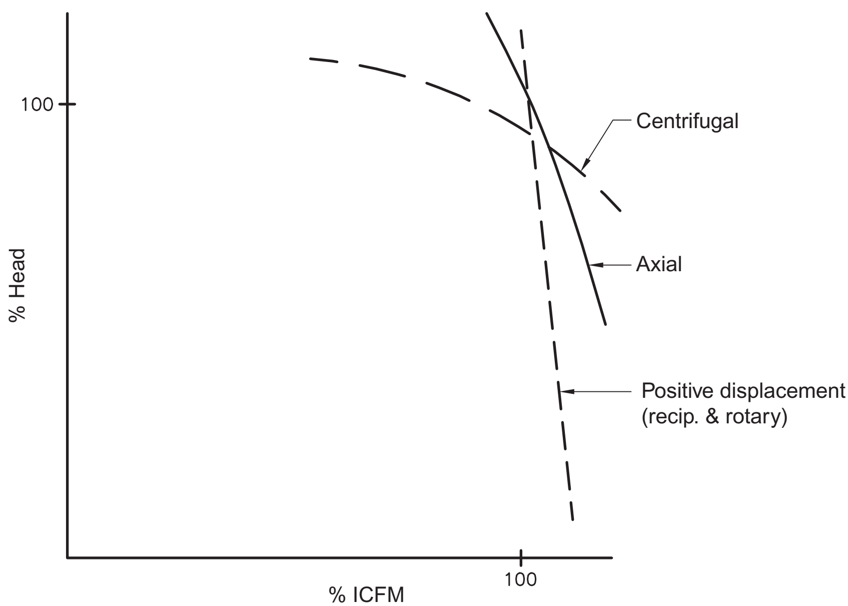


Fig. 7.23 Constant speed characteristics of various types of compressors.

Table 7.6 Generalized compressor control methods

	Centrifugal	Axial	Reciprocating	Rotary
Recycle (external)	X	X	X	X
Recycle (internal)				O
Speed	X	X	O	O
Suction pressure	O	O	N/A	N/A
Discharge pressure	X	N/A	N/A	N/A
Variable inlet valves	O	X		
Unloaders/Clr Vol			X	

of construction. Hydrogen embrittlement is a potential problem in compressing gases having a significant hydrogen partial pressure. If the gas is "wet" with water or hydrocarbon liquids, it may be necessary to provide special separation and heating equipment for the compressor suction system.

7.4.3.4 Site conditions

Climate conditions play an important role in compressor applications. Compressors are normally present in a heated enclosure in frigid climates and are sometimes enclosed in temperate climates. If the ambient temperature is −20°F (−29°C) or less, special material requirements are needed for an air compressor with an atmospheric suction, regardless of a heated enclosure. The climate also dictates the requirements for winterization and tropicalization.

Environmental restrictions on noise and emissions, area classification for electrical devices, and any special safety hazards should be reviewed.

7.4.3.5 Service requirements

The criticality of the service should be thoroughly reviewed. A service may be termed "critical" if one or more of the following conditions exist:

- Compressor does not have a spare and therefore, failure would cause an interruption in production resulting in substantial economic losses
- Service is such that a compressor failure could cause damage in the plant
- Service is such that a compressor failure could create a safety hazard.

7.5 Application and selection criteria

7.5.1 Selection basis

The selection of compressors involves satisfying the following requirements:

- *Suitability for Service*: The machinery should be sized for rated conditions and be sufficiently flexible to accommodate off-design conditions.
- *Dependability*: The design should have proven reliability and be easy to maintain.

- *Economic performance*: The efficiency should be high without undue sacrifice in flexibility and reliability.
- *Safety and environmental compatibility*: The installation should permit safe operation while complying with noise and environmental regulations.
- *Low cost*: The installed cost (factored with financial, operating, and maintenance costs) should result in the lowest evaluated cost for the payback period.

7.5.2 Approximate application ranges

Approximate application ranges in terms of ICFM and discharge pressure for four categories of compressors can be quickly approximated using Fig. 7.24.

In general, the following general compressor application ranges by type are:

- *Centrifugal* have the widest range of application.
- *Reciprocating* can compress lower volumes than centrifugal compressors and are suitable for high-pressure applications.
- *Axial* are suitable for high capacity applications.

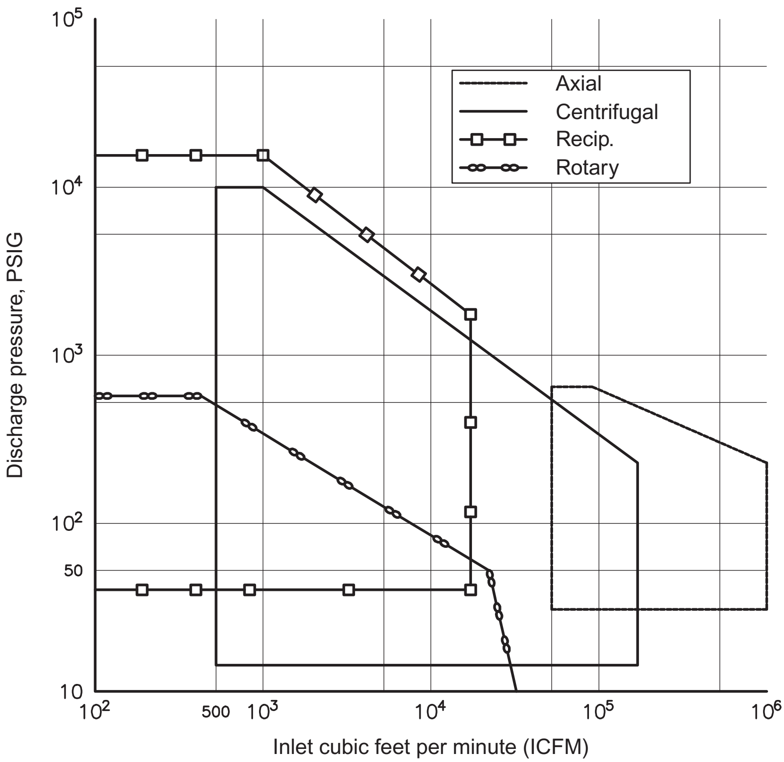


Fig. 7.24 Approximate application ranges. *Note*: Subsection 153 contains background information on using this figure.

- *Rotary*: Centrifugal and reciprocating compressors cover the entire range of application for rotary compressors. As a result, they are generally selected for reasons other than their pressure and capacity range coverage.

Figs. 7.25–7.29 provide further information to clarify the decision. These figures will help resolve conflicts selection when the application falls into an area of overlap in Fig. 7.24. Section 7.5.3 provides more detailed explanation for selecting compressor categories.

The data used in the aforementioned figures are generalized and there may be cases where a certain manufacturer has a design that will surpass the practical limits shown.

7.5.3 Compressor selection process

After the compression duty has been sufficiently defined, the following steps may be taken for initial selection of the best category of compressor.

Step 1: Convert flow rate to ICFM using Eq. (7.3) (also see Eqs. 7.41, 7.42, and 7.44)

Step 2: Calculate overall pressure ratio, $R_t = P_2/P_1$ (absolute pressure must be used)

Step 3: Calculate discharge temperature, T_2, using Eq. (7.41) and an adiabatic efficiency, $\eta_{ad} = 0.75$ for all types of machines except. Use $\eta_{ad} = 1.0$ for reciprocating compressors (absolute temperature).

Step 4: If the discharge temperature is less than 300°F (149°C) the application can most likely be achieved in one step of compression, without intercooling, with a centrifugal or rotary compressor. For reciprocating units, the discharge temperature for a single cylinder should be less than 300°F (149°C).

Step 5: If Step 4 indicates that intercooling is not necessary, refer to Fig. 7.24 and select the type of compressor. Then proceed to Step 10.

Step 6: If the temperatures cited in Step 4 are exceeded, more than one step of compression will be required (note that the word "step" is used to denote a section of the compression duty to avoid with compression "stages.") Each row of blades in an axial unit or each impeller in a centrifugal compressor is called a "stage," whereas each cylinder of a reciprocating unit is called a stage. The number of steps or sections is estimated by assuming an equal pressure ratio for each step. Use 3% allowance for pressure drop between the steps. By trial-and-error method, use the following equation and Eq. (7.12) to determine the number of steps keeping the discharge temperature at 300°F (149°C) or less.

$$R = \left[\frac{R_t}{(0.98)^{n-1}}\right]^{\frac{1}{n}} \tag{7.54}$$

where:
R = pressure ratio of each step
R_t = overall pressure ratio
n = number of steps; 2,3,4…
n–1 = number of intercoolers

	Local priority			Compressor type											
				Axial			Centrifugal			Recip			Rotary		
	Critical	Import	Not imp	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor
				X			X				X			X	
Efficiency					X			X		X				X	
Capacity turndown-constant speed						X			X	X			X		
Capacity turndown-variable speed					X			X		X			X		
Plot area required				X			X					X	X		
Pulsationless flow				X			X					X		X	
Noise					X			X		X					X
Shaking forces transmitted to surroundings				X			X					X	X		
Foundation size				X			X					X	X		
Pressure ratios > 15						IMPR		X		X					IMPR
Discharge pressure >1000 PSIG						IMPR	X			X					IMPR
Discharge pressure >5000 PSIG						IMPR		X		X					IMPR
Vacuum applications						X		X			X		X		
Handling entrained liquids						X	X				X			X	
Low horsepower applications						X			X	X			X		
Maintenance costs				X			X				X			X	
Sidestream capability						X	X				X				X
Complexity of oil seals when required					X			X		X	(SIMPLE SYSTEM	PACKING USED)			X
Complexity of control system					X			X		X			X		
Suitability for turbine drive				X			X				①			X	
Suitability for motor drive					X			X		X			X		
Suitability for engine drive						X			X	X				X	
Piping design complexity					X			X				X	X		
Unit purchase price 2000 BHP range						X		X			X		X		
Unit purchase price 500 BHP range						IMPR			X		X		X		
Unit purchase price 100 BHP range						IMPR			IMPR	X			X		

IMPR = Impractical

Notes :

① Not common, but has been successful in some upstream application

Fig. 7.25 Selection guide.

Ranges:

ICFM:	Typical: Low: High:	75,000–250,000 30,000 1,000,000
Discharge pressure:	Typical: High:	15–150 psig 550 psig (special design for LNG plant)
Discharge temperature:	Typical: High:	400–650°F 720°F
No. of stages per casing:	Typical: High:	6–15 20 (special to 22)
Adiabatic head per stage, ft:	Typical: High:	4000–5000 6000
Speed, RPM:	2800−12,000	
Bhp per casing:	Typical: High:	6000–50,000 over 100,000

Selection notes:

- Generally used for air service - high volume and relatively low pressure. Typical pressure ratios for air service are 2.5 to 7. High pressure ratio is about 14.
- Is more efficient than centrifugal.
- Is usually physically smaller and lighter in weight than centrifugal for same duty.
- Speed is somewhat higher than that of centrifugal for same duty.
- Two casings can be put in tandem arrangement, but it is seldom done.
- Some designs have provisions for intermediate nozzles for intercooling or sidestreams, but this feature is not often employed.
- Very narrow stable operating range at constant speed—about 12%.
- Some designs utilize one or more centrifugal impellers on the high-pressure end of the rotor. This feature greatly improves stable operating range.
- Can be fitted with variable stator vanes on first through fifth (and higher) stages to widen performance map. Machines so fitted are often operated at constant speed.
- Relatively quiet operation.
- Typical applications: Large air compressors, such as FCC or coker air blowers. Front-end air compressor for combustion gas turbines (not specified separately)

Fig. 7.26 Axial compressor—application ranges and selection notes.

Step 7: Calculate suction and discharge pressure for each step of compression using the R determined in Step 6 and taking a 3% pressure drop between each step.

Step 8: Calculate ICFM for each compression step. The suction temperatures for the first step should be known. If suction temperature for succeeding steps is unknown, use 15°F (−9°C) plus the temperature of available cooling water; or if cooling with air coolers is desired, use 25°F (−4°C) plus the design maximum ambient temperature. If these values are not yet known, use 100°F (38°C) as suction temperature of succeeding steps.

Step 9: Refer to Fig. 7.24 and select the compressor category that will satisfy the ICFM and discharge pressure for all compression stages. Generally, it is desirable that

Ranges:

ICFM:	Typical: Low: High:	1500–100,000 500 180,000 (360,000 for double suction)
Discharge CFM (DCFM):	Low:	250 (can be lower with special designs)
Discharge pressure, psig:	Typical: High:	15–4000 10,000 (one design has been tested at 13,000)
Discharge Temperature, °F	Typical: High:	250–300 350 (with oil seals) 500 with (labyrinth seals)
No. of impellers per casing:	Available: High:	1–10 8–10 (to 20,000 ICFM) 6–7 (20,000–40,000 ICFM) 4–5 (>40,000 ICFM)
Adiabatic head per stage, ft:	Typical: High:	8000–10,000 13,000 (special to 30,000)
Speed, RPM:	Typical: High:	3000–14,000 30,000 (special to over 50,000)
Bhp per casing:	Typical: High:	1000v20,000 over 50,000

Selection notes

- Most versatile type of compressor with wide application range.
- Each lower pressure casing may have up to three pairs of intermediate nozzles (8 nozzles total) for connecting intercoolers. This means that one casing can have as many as four sections of compression, but typically only three sections per casing are used. Note that only one intermediate nozzle is required to introduce or extract each sidestream. Some refrigeration compressors have as many as three sidestreams.
- As many as four casings have been driven in tandem without interposed gear. Two casings in tandem are commonplace, as are two casings separated by a gear.
- Polytropic efficiency varies widely from about 60% at low ICFM to over 80% at very high ICFM. Efficiency also varies inversely with number of impellers in series.
- 1000 ICFM or 1000 bhp are about the minimum economic sizes in API class machines. Refrigeration class machines down to 500 bhp are available.
- Noisy without acoustic treatment
- Stable operating range for one multi-stage casing is usually about 30% at constant speed. Further capacity reduction at constant speed can be done by (1) variable inlet guide vanes on first stage (fairly efficient), (2) suction or discharge throttling (less efficient), or (3) bypass (inefficient). When two or more casings are driven in tandem, the overall stable operating range is reduced. Stable operating range varies inversely with number of impellers in series.
- 10 impellers per casing is generally not recommended. Use a maximum of 8 impellers per casing for initial estimating.
- Typical applications: Process-gas recycle, high-capacity plant/process air systems, pipeline compression, high-capacity refrigeration.

Fig. 7.27 Centrifugal compressor—application ranges and selection notes.

one type of compressor will handle all steps. Refer the initial selection back to the process engineer to ensure the selection is compatible with the process requirements. Occasionally, it may be necessary to use a combination of types such as axial for the low-pressure (LP) step and a centrifugal for the high-pressure (HP) step. Another example would be a centrifugal and a reciprocating for LP and HP steps, respectively.

Step 10: Calculate weight flow using Eq. (7.15) and assume that this value is constant for all compression steps. Calculate adiabatic head and GHP for each step using Eqs. (7.26), (7.31). For the first estimate, it is convenient to use adiabatic relationships for all types of compressors to minimize computations. Also, compressibilities can be neglected in most cases unless the gas deviates widely. Add GHPs to find the total GHP for the compression duty.

Ranges:

ICFM:	Typical: High:	100–3000 up to 7500 per cylinder for vacuum or low pressure service to 20 psig discharge, or 6000 to 50 psig.
Discharge pressure, psig:	Typical: High:	40–6000 up to 15,000 (special to over 35,000)
Discharge temperature, °F:	Typical: High:	250 300 (400 in special cases)
No. of crank throws per frame:	Typical: Maximum:	2–6 8 (10 available)
Speed, RPM:	Typical:	1000 (5-inch stroke) 720 (7-inch stroke) 514 (10-inch stroke) 360 (14-inch stroke) 257 (20-inch stroke)
Bhp:	Typical: High:	150–6000 12,000

Selection notes

- Very high efficiency. At pressure ratios of 2.5–5 per stage, adiabatic efficiency is 85%–89%, and mechanical efficiency is 95% for slow-speed (600 RPM or less) machines rated at 200 HP and higher. At pressure ratios lower than 1.5, valve losses become significant.
- Available in non-lubricated version with minor sacrifice in efficiency and reliability.
- Horizontally opposed configuration requires large plot area which is especially significant on skidded (packaged) units. Vertical, "Y", and "L" configurations reduce plot area, but limit number of cylinders per frame.
- Can usually achieve capacity control down to 25% with five-step unloading (down to 12-1/2% on multi-unit installation, or unit with two first-stage cylinders). Small loss in efficiency at reduced capacity.
- Also can bypass discharge back to suction for more capacity control.
- Typical Applications: Downstream—Hydrogen makeup/booster compression, overhead/flare gas compression, small instrument air compression, refrigeration. Upstream—Gas injection, gas processing, gas-product compression.

Fig. 7.28 Reciprocating compressor—application ranges and selection notes.

Step 11: Review Figs. 7.25–7.29 which may help resolve the choice of the compressor when the application falls into an overlapped area.

Step 12: At this point, some thought should be given to reliability and availability of the compression system. If the service is deemed to be critical, the following number of units is typically used:

Axial/Centrifugal	One 100% unit
Reciprocating	Three 50% or two 100% units
Rotary	Same as reciprocating (Rotary are seldom used in critical service)

For general-purpose service, one 100% unit is the usual choice. In some instances, the capacity may vary widely on a seasonal basis or it may build up over a period of years. In such cases, it may be economical to use smaller sized units.

Step 13: Proceed to the chapter of this text that corresponds to the selection made in the previous steps (*Chapter 9—Centrifugals, Chapter 10—Positive Displacement*). Review this information to verify and refine the selection.

Ranges:		
ICFM:	Typical:	300–2000
	High:	30,000 (For low-pressure and vacuum service)
Discharge pressure, psig:	Typical:	40–150
	High:	550 (usually attained in typical: second of two casings in tandem.
Discharge temperature, °F	Typical:	200–300
	Maximum:	450 (for some designs)
Pressure ratio,	Typical:	2–3
P_2/P_1:	High:	4 (20 is attainable with oil-flooded and liquid ring machines.)
Differential pressure,	Typical:	10–75
(P2 /P1), psi:	High:	170
Speed, RPM:	Typical:	300–3600
	High:	20,000
Bhp:	Typical:	50–2000
	High:	6000

Selection notes

- Available in oil-flooded, drop-lubricated, "dry" (oil-free), and liquid-ring design. When the oil-flooded or liquid-ring version is applied, the discharge temperature is substantially less than that indicated by adiabatic calculations. For example, the actual discharge temperature might be 200°F or less versus a calculated value of 350°F.
- Good efficiency at low-pressure ratios (somewhat lower than that of reciprocating).
- Require inlet and discharge silencers at higher power levels to achieve tolerable noise level.
- Very good for skidding and semi-portable installations due to small size and freedom from vibration.
- Are often two-staged by connecting two casings in tandem. Tandem arrangement allows sidestream (in or out)
- Can have stepless capacity control down to 10%–15% of rated with hydraulically operated slide valve. Efficiency loss at turndown greater than that of reciprocating.
- Capacity control can also be achieved by speed adjustment down to 50% of rated speed.
- Dry (oil-free) units are very noisy.
- Typical Application: Freon, NH_3 refrigeration, plant air.

Fig. 7.29 Rotary compressor—application ranges and selection.

Step 14: If the type of driver is known, make sure that it is compatible with the size and rating of the type of compressor selected. If the type of driver is unknown, use the information contained in *Chapter 12* for preliminary selection. Due to limitations of the driver or size of the compressor train, it may be necessary to use two or more equally sized trains for the required compression capacity. This step should be considered along with Step 12.

Step 15: When the application involves the introduction of sidestreams, for example, refrigeration compressor or flash gas compressors accepting intermediate sidestreams, it is necessary to calculate the properties of the mixture of the two gas streams at the entrance of the next section or stage of compression.

Step 16: Contact at least two vendors for each type of compressor selected. Have each vendor prepare preliminary selections, and submit order-of-magnitude prices and estimated performance information. If it becomes evident that a small design change might save a considerable amount of money, the change should be referred to the process engineer. In some cases the process design can be conservative and a slight modification may allow a substantial savings.

7.5.4 Selection analysis

After the initial selection has been made and estimating information has been received from the vendors, it is advisable to make an analysis of the following factors to confirm the economic feasibility of the compressor:

- Purchase price of compressor and driver
- Cost of auxiliaries
- Space requirements
- Foundation requirements
- Installed cost
- Cost of utilities, for example, cooling water, electricity, steam, and so on.
- Annual compressor load profile, for example, amount of operating time at full and part loads.
- Annual power, steam, or fuel costs corresponding to load profile
- Compressor availability versus production goals
- Economic factors of interest on capital, depreciation, income tax, and escalation
- Local experience, for example, are operating and maintenance personnel familiar with the type of machine selected
- Standardization of units and spare parts

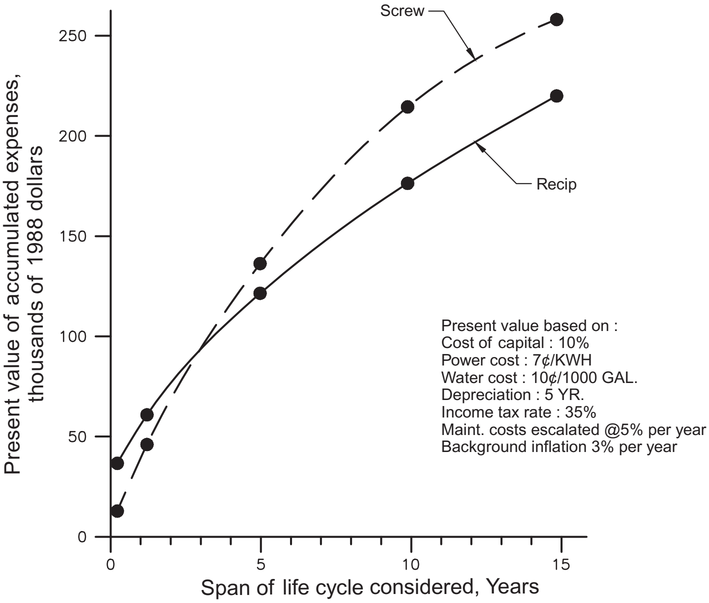
This analysis is partially important when two or more categories or types of compressors are initially selected. A quick look at first costs or installed costs will rarely provide a reliable indication of the overall lifecycle costs of owning and operating a compressor for a period of years. Required payback periods for projects are typically in the range of 2 to 5 years. In some cases, it may be desirable to examine the lifecycle costs for periods of 15 years or more.

7.5.4.1 Example economic studies

The usual approach is to calculate the present value and accumulate them on an annual basis for the desired number of years. Fig. 7.30 shows the results of an economic study for two different categories of air compressors for 500 ACFM. As shown in Fig. 7.30, the installed cost of the reciprocating compressor is 50% higher than that of the screw unit, but the accumulated costs are equal when considering a period of longer than 2-1/2 years. (Note also that using unit costs such as $/HP or $/ICFM in terms of installed costs would be very misleading.) Efficiency directly affects power costs which is the dominant component of the overall accumulated expenses.

Fig. 7.31 shows the total expenses for two lifecycle periods for three categories of compressors in a large air compression duty. In this comparison, the reciprocating and centrifugal units are even in costs for a five-year cycle, despite the fact that the installed cost of the reciprocating unit is twice that of the others. But when considering the 15-year case, efficiency takes over, making the reciprocating compressor the least costly.

Figs. 7.31 and 7.32 do not show any particular unit costs or trends for air compressors or compressors for other gases. Their only purpose is to demonstrate the need for



Other data	Recip	Screw
	Years	
$/HP (installed)	338	170
$/ICFM (installed)	74	46
BHP/100 ICFM	21.9	26.9

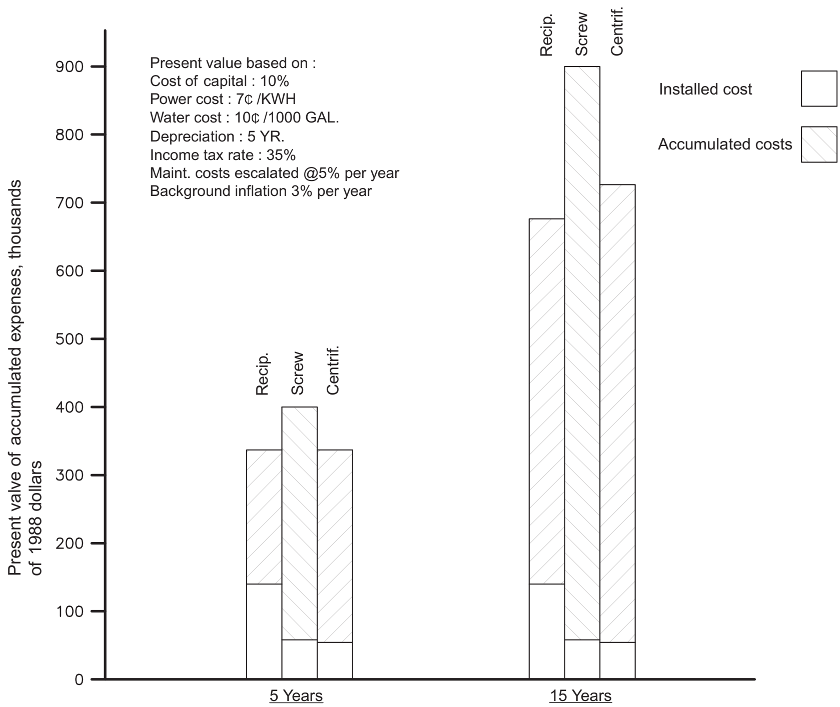
This graph displays the total compressor costs (purchase, installation, energy consumption, and maintenance), when totaled for different assumed project lives. The total costs are displayed in terms of "present value." This is the amount that would be needed up front to pay all the costs over the project life, assuming, the money compounded at the cost of capital (10%), and were spent to pay the respective costs.

This chart shows that the reciprocating compressor costs more initially, but when you consider the future power consumption, the screw compressor will actually cost more. This is because the reciprocator is more efficient, and will consume less power over the years.

Fig. 7.30 Lifecycle costs for light-duty reciprocating and rotary screw compressors for a 500 ICFM (nominal) 125 psig air compressor.
For illustration purposes only. Do not use for design.

an economic analysis. For critical services, the requirements for reliability or safety may overrule the choice determined by the other economic factors, such as purchase price, energy, and so on.

It is not uncommon, economic factors, or the methods for determining them, will have already been established by the sponsors for a given project. If this is the case, the economic study will be simplified. In some cases, the project may rule that a study is not required. At any rate, the economics of the proposed installation should be reviewed with the appropriate operating personnel.



Other data	Recip	Screw	Centrif.
$/HP (Installed)	433	191	176
$/ICFM (Installed)	83	49	45
BHP/100 ICFM	19.1	25.8	23.8

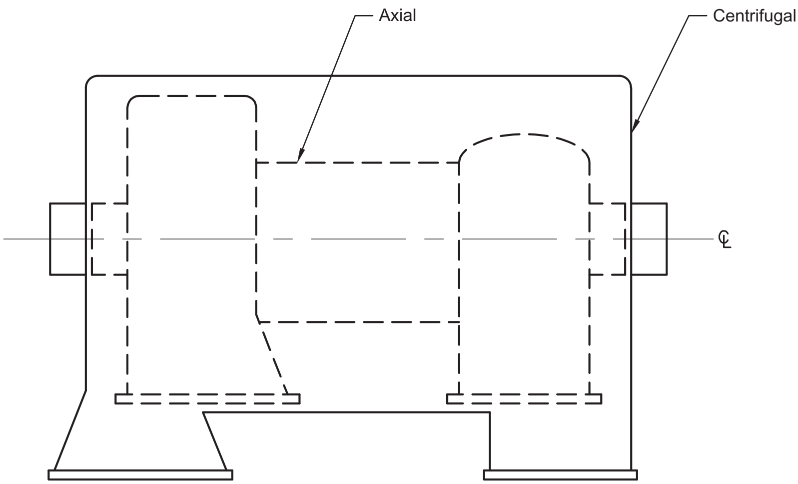
Note: This graph depicts two cases of "present value" for assumed project lives of 5 and 15 years. (Present value is explained in Figure 100–17.)

This chart shows that: 1) installed costs are a minor part of the total compressor expense, even at a very short project life of 5 years, and 2) the reciprocating compressor becomes more attractive as longer project lives are assumed. Like Figure 100–17, this is because the higher efficiency of the reciprocator pays off in power consumption over the operating life of the compressor (despite typically higher maintenance expense).

Fig. 7.31 Lifecycle for heavy-duty reciprocating, rotary screw and centrifugal compressors for a 1500 ICFM (nominal) 125 psig air compressor.
For illustration purposes only. Do not use for design.

7.5.4.2 Typical dimensional charts

Figs. 7.32–7.35 provide further information on the physical sizes of the various types of compressors. Figs. 7.31 and 7.32 do not show any particular unit costs or trends for air compressors or compressors for other gases. Their only purpose is to demonstrate the need for an economic analysis. For critical services, the requirements for reliability



	Axial	Centrifugal
ICFM	10,000	10,000
No. of stages	14	7
Approx. speed, RPM	6000	3200
Total weight, LBS	75,000	135,000
Relative cross-sectional Area of casing at suction end	0.85	1.0

Fig. 7.32 Comparison of axial and centrifugal compressor size.

Dimensions for horizontally- and vertically-split centrifugal compressors					
Horizontally split[a]					
ICFM	**Width, ft**	**Length, ft**		**Weight, lbs**	
		3-stg	**8-stg**	**3-stg**	**8-stg**
20–40K	8	7	11	24 K	40 K
9–20K	6	6	9	13 K	20 K
4–9K	4	5	6	7 K	10 K
0.5–4K	4	4	6	6 K	9 K
Vertically split[a,b]					
Weights @ casing rating,	**Width, ft**	**Length, ft**	**750 psi**	**3000 psi**	**6000 psi**
9–20K	6	7	29	35	50
4–9K	5	6	22	33	41
0.5–4K	4	5	18	22	25

a Width dimensions shown are across support feet.
b Based on 5-stage casing.

Fig. 7.33 Typical plot dimensions and weights of centrifugal compressor casings.

Maximum ICFM	3 Wheels L x W x H, Ft.	Add per wheel Lbs.	Inches	Estimated maximum skid weights for 3 wheels with motor driver, Kips	Add skid weight per each add'l wheel, Kips
Horizontally split units					
2600	4.2 x 3.5 x 2.0	300	4	10.8	0.2
5850	5.5 x 4.5 x 3.5	1300	6	13.1	0.2
13,000	6.3 x 5.3 x 4.0	2000	7	15.0	0.3
23,000	5.5 x 7.0 x 5.8	1850	7	16.5	0.3
35,000	7.4 x 6.5 x 6.0	4000	9	18.6	0.4
Vertically split units					
5000	5.0 x 4.3 x 4.6	1000	4	16.8	0.2
	5.0 x 5.0 x 5.0	1400	4	16.8	0.2
	6.0 x 5.2 x 5.8	2200	5	19.5	0.3
	6.4 x 6.0 x 7.0	4500	6	21.7	0.4
10,000	5.5 x 5.6 x 6.0	2000	5	18.0	0.3
	6.2 x 6.2 x 6.3	3400	6	20.1	0.4
	6.9 x 7.3 x 7.2	6400	6	21.7	0.4
23,000	6.7 x 6.7 x 6.3	2500	8	18.6	0.4
	8.0 x 7.5 x 7.2	4200	8	22.4	0.5

Fig. 7.34 Typical dimensions for centrifugal compressors—horizontal and vertically split units.

Frame horsepower	No. of cylinders	RPM	Average footprint[a]	Average weight[a,b]
500	2	900	14′W x 5′L	13,500 lbs
1000	2	514	14′W x 9′L	26,000 lbs
5000	4	327	22′W x 14-1/2′L	120,000 lbs
10,000	6	257	26′W x 20′L	210,000 lbs

a Does not include driver
b Frame plus average size cylinders

Fig. 7.35 Approximate plot areas and weights of typical reciprocating compressors.

or safety may overrule the choice determined by the other economic factors, such as purchase price, energy, and so on.

Figs. 7.32–7.35 provide further information on the physical sizes of the various types of compressors.

7.5.4.3 Packaging considerations

Packaging is a technique used to minimize construction time and labor costs at the installation site. It consists of placing the compressor, driver, auxiliaries, and control system on one or more skids (baseplates). Piping, tubing, and wiring are routed between a minimum number of terminal points on the perimeter of the skid. These terminal points facilitate connecting the package to the system at the jobsite.

The size and complexity of packaged equipment range from a simple air compressor package rated at a few hundred horsepower to a large and complicated package containing a gas compressor driven by a large mechanical drive gas turbine.

When packaging is applied on a large scale to a major project, the concept is known as modularization or modular construction. This approach has particular application to projects in remote areas where the availability of skilled labor is either low or costly and difficult to implement. Modular construction takes advantage of the availability of skilled labor at major industrial centers throughout the world. In addition to potential labor cost reductions, this approach provides the opportunity to improve the overall project schedule.

Modular construction is used extensively for offshore platforms and onshore facilities such as those on Alaska's North Slope and in jungle and desert locations. Typical compressor module weights are 25 to 3000 tons. Module or package size and weight are limited by factory handling capability, transportation constraints, and the capacity of jobsite lifting or moving apparatus.

Console-mounted lube- and seal-oil systems for compressors are good examples of smaller packages. Fig. 7.36 provides some general weights and dimensions which may be used for early estimates.

Figs. 7.37 and 7.38 list some estimating weights and dimensions for reciprocating compressors (with and without coolers) with various drivers. Fig. 7.39 shows typical dimensions and weights for integral gas engine-driven reciprocating compressors. Figs. 7.40 and 7.41 provide similar information for centrifugal compressors including turbine drivers and enclosures.

Nominal oil flow, gpm	L x W x H, ft	Dry wt, Kips	Operating wt, Kips
25	14 x 8 x 7	8.5	10.0
50	18 x 9 x 8	12.0	15.9
75	19 x 9 x 8	14.0	18.5
100	20 x 10 x 8	16.0	22.0
150	22 x 11 x 8	18.0	27.0
20	24 x 12 x 8	20.0	32.0
300	32 x 12 x 8	35.0	60.0
400	40 x 12 x 8	52.0	92.0

Fig. 7.36 Typical packaged seal/lube oil system weight and dimensions.

Driver	HP	L x W x H ft	Weight Kips	MMSCFD
Engine	1200	34 x 13 x 16	100	19.1
Engine	565	35 x 12 x 14	60	2.3
Engine	500	30 x 12 x 14	47	4.7
Engine	450	30 x 18 x 18	55	1.8
Engine	415	30 x 12 x 13	48	5.2
Motor	350	20 x 12 x 14	51	2.0

Fig. 7.37 Typical reciprocating packages with coolers weight and dimensions (900 rpm separable compressors).

Driver	HP	L x W x H ft	Weight Kips	MMSCFD
Engine	2600	38 x 12 x 10	106.0	13.2
Engine	1000	35 x 13 x 12	71.0	9.5
Turbine	2000	35 x 12 x 12	100.0	17.0
Turbine	1000	50 x 12 x 12	80.0	11.6
Motor	2000	22 x 12 x 10	89.5	12.5
Motor	900	26 x 12 x 8	49.0	12.5

Fig. 7.38 Typical reciprocating packages without coolers weights and dimensions (900 rpm separable compressors).

HP	LxWxHft	Weight Kips
1000	23 x 14 x 12	100
1500	27 x 14 x 12	140
2000	33 x 16 x 15	190
2400	33 x 16 x 15	210

Fig. 7.39 Typical integral compressor packages weights and dimensions.

Maximum ICFM	3 Wheels L x W x H, ft.	Add per wheel lbs.	Add per wheel Inches	Estimated maximum skid weights for 3 wheels with motor driver kips	Add skids weight per each add'l wheel, Kips
Horizontally split units					
2600	4.2 x 3.5 x 2.0	300	4	10.8	0.2
5850	5.5 x 4.5 x 3.5	1300	6	13.1	0.2
13,000	6.3 x 5.3 x 4.0	2000	7	15.0	0.3
23,000	5.5 x 7.0 x 5.8	1850	7	16.5	0.3
35,000	7.4 x 6.5 x 6.0	4000	9	18.6	0.4
Vertically split units					
5000	5.0 x 4.3 x 4.6	1000	4	16.8	0.2
	5.0 x 5.0 x 5.0	1400	4	16.8	0.2
	6.0 x 5.2 x 5.8	2200	5	19.5	0.3
	6.4 x 6.0 x 7.0	4500	6	21.7	0.4
10,000	5.5 x 5.6 x 6.0	2000	5	18.0	0.3
	6.2 x 6.2 x 6.3	3400	6	20.1	0.4
	6.9 x 7.3 x 7.2	6400	6	21.7	0.4
23,000	6.7 x 6.7 x 6.3	2500	8	18.6	0.4
	8.0 x 7.5 x 7.2	4200	8	22.4	0.5

Fig. 7.40 Typical dimensions for centrifugal compressors horizontally split units and vertically split units.

ISO HP	Model	Speed, rpm	Turbine-compressor skid dimensions L x W x H, ft	Ancillary equipment dimensions L x W x H, ft
4250			34.5 x 8.0 x 20.0	
16,000			59. x 10.0 x 24.0	
26,500			61.0 x 10.0 x 24.0	
4900	Single		30.0 x 8.0 x 8.0	21.4 x 7.8 x 10.7
4900	Tandem		34.0 x 8.0 x 8.0	21.4 x 7.8 x 10.7
10,600		8140	48.5 x 8.0 x 11.8	38.0 x 18.0 x 15.1
3830		15,700	27.2 x 7.8 x 8.3	21.4 x 7.8 x 10.0
1165		22,300	23.1 x 5.8 x 7.3	12.0 x 6.0 x 4.8
4900	Single		26.0 x 8.0 x 8.0	
4900	Tandem		29.0 x 8.0 x 8.0	
2500	Single		28.0 x 8.0 x 8.5	
2500	Tandem		33.0 x 8.0 x 8.5	
1875			25.0 x 8.0 x 8.6	
ISO HP	**Approximate weights, kips**	**Skid weight (kips) per additional foot**	**With enclosure type add kips**	**Engine control cab add-Kips**
4250	33	0.8		
16,000	96	1.6		
26,500	105	1.8		
4900	74			
4900	81			
10,600	107.5	1.2		
3830	52.1	0.6		
1165	18.5	0.4		
4900	55.0		Open side - 4.1 Total encl. 5.3	0.5
4900	66.0		Open side - 4.1 Total encl. 5.3	0.5
2500	33.0		Open side - 3.4 Total encl. 5.0	0.5
2500	48.0		Open side - 3.4 Total encl. 5.0	0.5
1875			Total encl. 5.3	0.5

Fig. 7.41 Typical gas turbine-driven centrifugal compressor dimensions and weights.

7.6 Exercises

(1) Natural gas with the following properties and conditions is to be compressed by a centrifugal compressor

Specify gravity = 0.620
Ratio of specific heats = 1.26
Suction temperature = 99°F
Suction pressure = 256 psia
Flow rate = 45 MMSCFD
Discharge pressure = 665 psia

Determine the following:

(a) Compression ratio
(b) Discharge temperature
(c) Polytrophic head
(d) Flow rate at suction conditions in actual cubic feet per minute
(e) Brake Horsepower required to compress this gas assuming 75% adiabatic efficiency and 95% mechanical efficiency
(f) Approximate brake horsepower using the "Approximate" BHP Formula
(g) Approximate brake horsepower using the GPSA "Quick Look" Approximate curve

(2) A reciprocating compressor has been selected for gas lift services. The following gas properties are known:

$Q_g = 20\text{MMSCFD}$
$T_s = 150^{o}\text{F}$
$P_s = 75$ psig
$P_d = 1000$ psig
$S = 0.6$
Elevation = 5000 feet above sea level

Determine the following:

(a) Number of stages
(b) Required Horsepower using GPSA approximation graph
(c) Type of compressor
(d) Number of compressors

(3) Natural gas with the following properties is to be compressed with a centrifugal compressor:

(a) Suction pressure = 585 psig
(b) Atmospheric pressure = 15 psig
(c) Discharge pressure = 1200 psig
(d) Operating temperature = 100°F
(e) Ratio of specific heats = 1.26
(f) Gas specific gravity = 0.65
(g) Polytrophic efficiency = 0.8

Determine the following:

(a) Compression ratio
(b) Discharge temperature
(c) Number of stages
(d) Approximate BHP. Use GPSA Graph and "Quick Look" Formula
(e) Adiabatic head
(f) ACFM, if the desired flow rate is 175 MMSCFD
(g) BHP required assuming a 75% Adiabatic efficiency and a 95% mechanical efficiency

7.6.1 Solution

1. Compression ratio (assume one stage)

$$R = [P_d/P_s]1/n$$

$$R = [665/256]1/1$$

$$R = 2.6$$

2. Discharge temperature

$$T_d = T_s \left[\frac{P_d}{P_s}\right]^{\frac{k-1}{kn_p}}$$

$$T_d = (460+99)\left[\frac{665}{256}\right]^{\frac{1.26-1}{1.26} x \frac{1}{0.8}}$$

$$T_d = 715^{\circ}\mathrm{R}$$

$$T_d = 255^{\circ}\mathrm{F}$$

3. Polytrophic head

Determine the compressibility factors at suction and discharge conditions—use compressibility curves for 0.60 and 0.65 gas specific gravity—take the average.

	Z_s	Z_d
SG = 0.60	0.97	0.96
SG = 0.65	0.95	0.94
Z_s average	0.96	
Z_d average		0.95

$$H_{\mathrm{AD}} = 53.3[Z_{sv}]\left[\frac{T_s}{S}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{kn_p}} - 1\right]$$

$$H_{\mathrm{AD}} = 53.3\left[\frac{0.96+0.95}{2}\right]\left[\frac{460+99}{0.62}\right]\left[\frac{(1.26)(0.8)}{1.26-1}\right]\left[\left(\frac{665}{256}\right)^{\frac{1.26-1}{(1.26)(0.8)}} - 1\right]$$

$$H_{\mathrm{AD}} = 49{,}675 ft - lbllb$$

4. Flow rate in ACFM at suction conditions

$$\mathrm{ACFM} = 19.6\frac{Q_g Z_s T_s}{P_s}$$

$$\text{ACFM} = 19.6\frac{(45)(0.96)(460+99)}{256}$$

$$\text{ACFM} = 1.849 \text{ MMSCFD}$$

5. Horsepower

$$\text{BHP} = 0.0857\left[Z_{AVG}^{\frac{I}{k}}\right][Z_s]^{\frac{k-1}{k}}\left[\frac{(Q_g)(T_s)}{E_m E_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{kn_p}}-1\right]$$

$$\text{BHP} = 0.0857\left[\frac{0.97+0.95}{2}\right]^{\frac{1}{1.26}}[0.96]^{\frac{1.26-1}{1.26}}\left[\frac{(45)(460+99)}{(0.95)(0.79)}\right]$$

$$X\left[\frac{(1.26)(0.8)}{1.26-1}\right]\left[\left(\frac{665}{256}\right)^{\frac{1.26-1}{1.26X0.8}}-1\right]$$

BHP = 2972

6. Horsepower from approximate equation:

$$\text{BHP} = 22RnFQ_g$$

$$\text{BHP} = 22\left(\frac{665}{256}\right)(1)(1)(45) = 2572$$

$$\text{BHP} = 2572$$

7. Horsepower from GPSA curve:

$$R_t = \frac{664}{256} = 2.60$$

$$\text{BHP} = (59)(45) = 2655$$

7.6.2 Solution

1. Calculate overall compression ratio:

$$P_s = 75 + 12.2 = 87.2$$

$$P_d = 1000 + 12.2 = 1012.2$$

$$R_t = \frac{P_d}{P_s} = \frac{1012.2}{87.2} = 11.6 \text{ Greater than } 5\text{, so too high.}$$

2. First guess for R

$$R = (11.6)^{1/2} = 3.41$$

Assume $n = 2$

3. Calculate discharge temperature

Assume $k = 1.24$, $n_p = 1$ for reciprocating compressor

$$T_d = T_s\left(\frac{P_d}{P_s}\right)^{\frac{1-1}{kn_p}}$$

$$T_d = 610\left(\frac{1012.2}{87.2}\right)^{\frac{1.24-1}{(1.24)(1)}}$$

$$T_d = 773^{\circ}\mathrm{R} = 313^{\circ}\mathrm{F}$$

4. Select more stages as this discharge temperature is too high (greater than 300°F). Other alternatives would be to use lower R for first stage and interstage cool to lower suction temperature for second stage, or cool inlet gas.

Assume $n = 3$

$$R = (11.6)^{\frac{1}{3}} = 2.264$$

$$T_d = (150 + 460)(2.264)\frac{1.24 - 1}{(1.24)(1.0)}$$

$$T_d = 715^{\circ}\mathrm{R} = 255^{\circ}\mathrm{F}$$

5. Estimate horsepower from graph

$$\mathrm{BHP/MMSCFD} = 170(3\ \text{stages},\ k = 1.24)$$

$$\text{Total BHP} = (170)(20) = 3400$$

6. Calculate inlet flow

$$\mathrm{ACFM} = 19.6\frac{Z_s T_s}{P_s}Q_g$$

$$Z_s(\text{from charts}) = 0.98$$

$$\mathrm{ACFM} = \frac{(19.6)(0.98)(150 + 4)(20)}{87.2}$$

$$\mathrm{ACFM} = 2687$$

If two units are used

$$ACFM = 1344$$

If three units are used

$$ACFM = 896$$

7. Possible selections include:

(a) If one unit is used either a centrifugal or reciprocating can be used. For this horsepower consider integral rather than high-speed separable.
(b) If two 1700 HP units are used either a centrifugal or reciprocating. At this level a high speed is possible.
(c) If three 1133 HP units are used, than only reciprocating can be used. Choose high-speed separable.

8. Final choice depends on length of service, design curve of demand, and availability of units in stock.

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Dynamic compressors

8

8.1 Engineering principles

8.1.1 Background

Dynamic compressors are subdivided into centrifugal and axial compressors. Centrifugal compressors are by far the most widely used dynamic compressor and thus will be emphasized in this chapter. Axial compressors are similar in operation to that of a centrifugal compressor but are rarely used in upstream operations. A separate section in this chapter will discuss axial compressors in detail.

Operation of a dynamic compressor is based on the basic principles of thermodynamics. In the polytropic compression process, work is done on a fluid so as to raise its pressure. Dynamic compressors are better suited for constant operating conditions due to their narrow operating range.

Compression is achieved by applying inertial forces to the gas by the bladed impellers. Velocity energy (acceleration) is added to the gas by the rapidly rotating impeller. Part of this energy, approximately two-thirds, results in a static pressure rise of the gas in the impeller. After leaving the impeller, the gas enters a diffuser, which is a stationary component, where it slows down (decelerates) resulting in an additional pressure increase. After leaving the diffuser, the gas either exits the compressor case after single-stage compression or enters the eye of the next impeller for multistage compression.

Dynamic compressors are more reliable than other types of compressors because they do not have reciprocating components that experience cyclic stress while in service. Generally, one should select centrifugal compressors unless there is a specific reason not to do so, since they:

- Have the widest range of operation
- While less efficient than positive displacement compressors, they are more reliable
- Run unspared in many high-value processing facilities and plants
- Are the baseline compressor selection

Typical centrifugal compressor applications include:

- Associated gas gathering
- Gas plant compression
- Pipeline compression
- High-capacity refrigeration
- High-capacity plant/process air systems
- Process gas recycle
- Gas lift/injection

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00008-9

Table 8.1 summarizes typical application ranges of centrifugal compressors. As presented in Table 8.1, the centrifugal compressor is the most versatile type of compressor with the widest application range. Some specific selection notes are listed as follows:

- Each lower pressure casing may have up to three pairs of intermediate nozzles (8 nozzles total) for connecting intercoolers. This means that one casing can have as many as four sections of compression, but typically only three sections per casing are used. Note that only one intermediate nozzle is required to introduce or extract each sidestream. Some refrigeration compressors have as many as three sidestreams.
- As many as four casings have been driven in tandem without interposed gear. Two casings in tandem are common as are two casings separated by a gear.
- Polytropic efficiency varies widely from about 60% at low ICFM to over 80% at very high ICFM. Efficiency also varies inversely with number of impellers in series.
- 1000 ICFM or 1000 BHP is about the minimum economic sizes in API machines. Refrigeration class machines down to 500 BHP are available.
- Noisy without acoustic treatment
- Stable operating range for one multistage casing is usually about 30% at constant speed. Further capacity reduction at constant speed can be done by (1) variable inlet guide vanes on first stage (fairly efficient), (2) suction or discharge throttling (less efficient), or (3) bypass (inefficient). When two or more casings are driven in tandem, the overall stable operating range is reduced. Stable operating range varies inversely with number of impellers in series.

Table 8.1 Typical application ranges of centrifugal compressors

ICFM	Typical	1500–100,000
	Low	500
	High	180,000 (360,000 for double suction)
Discharge CFM (DCFM)	Low	250 (can be lower with special designs)
Discharge pressure, psig (bar)	Typical	15–4000 (1–276)a
	High	10,000 (one design has been tested at 13,000)
Discharge temperature, °F (°C)	Typical	250–300 (121–149)
	High	350 (with oil seals) 500 (with labyrinth seals)
No. of impellers per casing	Available	1–10
	High	8–10 (to 20,000 ICFM) 6–7 (20,000–40,000 ICFM) 4–5 (>40,000 ICFM)
Adiabatic head per stage, ft	Typical	8000–10,000
	High	13,000 (special to 30,000)
Speed, RPM	Typical	3000–14,000
	High	30,000 (special to over 50,000)
BHP per casing	Typical	1000–20,000
	High	over 50,000

- 10 impellers per casing are generally recommended. Use a maximum of 8 impellers per casing for initial estimating.

The remainder of this section covers the fundamentals of centrifugal compressors, describing the gas flow path, conversion of velocity to pressure, thermodynamic relationships, and the effect of component geometry on compressor performance.

These fundamental principles provide a basic foundation for troubleshooting performance problems, making rerating or initial selection estimates, evaluating vendor proposals, engineering compressor applications, and assisting with overall process design.

8.1.2 Gas flow path through a centrifugal compressor

An understanding of the gas flow path through a centrifugal compressor will provide a better understanding of the compression process. Unfortunately, there is much confusion concerning the term “stage” when applied to a centrifugal compressor. The process engineer thinks of a stage as a compression step made up of an uncooled section, usually consisting of several impeller/diffuser units. On the other hand, the mechanical engineer usually defines a stage as one impeller/diffuser set, and a section as a single compressor casing containing several stages. In this text the following definitions are used:

- *Stage* is defined as one impeller/diffuser set
- *Process stage* is defined as an uncooled section, or casing, containing several impellers/diffusers.

Based on the earlier definitions, a centrifugal compressor is made up of one or more stages; each stage consisting of a rotating component or impeller, and the stationary components which guide the flow into and out of the impeller. Fig. 8.1 shows the flow path through a section of a typical multistage unit.

8.1.3 Conversion of velocity energy to pressure

The pressure in a centrifugal compressor is increased by transferring energy of the gas and accelerating it through the impeller. All work on the gas is done by the impeller. The stationary components only convert the energy added by the impeller. Part of this energy is converted to pressure in the impeller while the remainder is converted to pressure as it decelerates in the diffuser. A typical pressure-velocity profile across a stage is shown in Fig. 8.2.

Since the kinetic energy is a function of the square of the velocity, the head, not pressure, produced is proportional to the square of the impeller tip speed as shown in Eq. (8.1):

$$H = K\frac{U^2}{g} \tag{8.1}$$

Fig. 8.1 Gas flow path through a centrifugal compressor. Courtesy of Elliot Company.

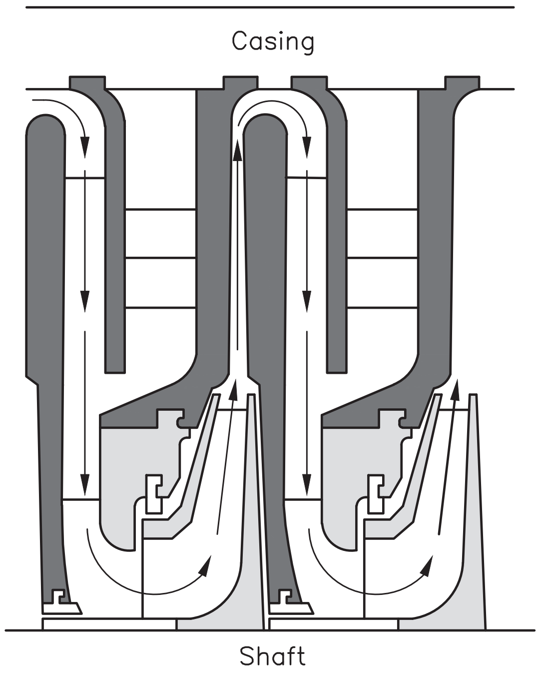
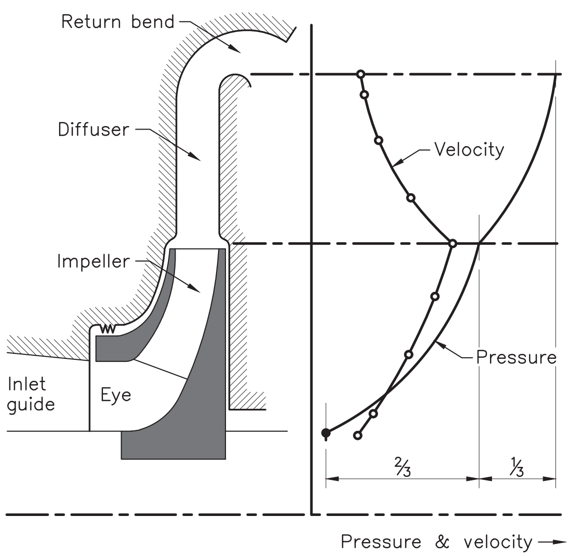


Fig. 8.2 Typical pressure-velocity profile across a stage of a centrifugal compressor.



where:
H = head, (ft-lb$_f$)/lbm, (N/m)/kg
U = impeller tip speed, ft/sec, m/sec
K = constant
$g = 32.174$, (ft-lb$_m$)/(lb$_f$ sec^2), 1 m kg/N sec^2

The term "head" is used to describe the work input to a compression process. The units of head are foot-pounds (force) divided by pounds (mass) [newton meter divided by kilograms]. In general, head is usually taken as "feet" or "meters."

Manufacturers typically define performance of individual impellers in terms of:

- *Head coefficient (μ)*: a function of actual work input and stage efficiency
- *Flow coefficient (θ)*: a nondimensional function of volume flow and rotational speed

Fig. 8.3 shows a typical individual impeller curve performance of a centrifugal compressor. The head coefficient typically varies from about 0.4 to 0.6. The surge line in Fig. 8.3 is discussed in Section 8.3.4

Using the head coefficient, the head can now be shown as:

$$H = \frac{\mu U^2}{g} \tag{8.2}$$

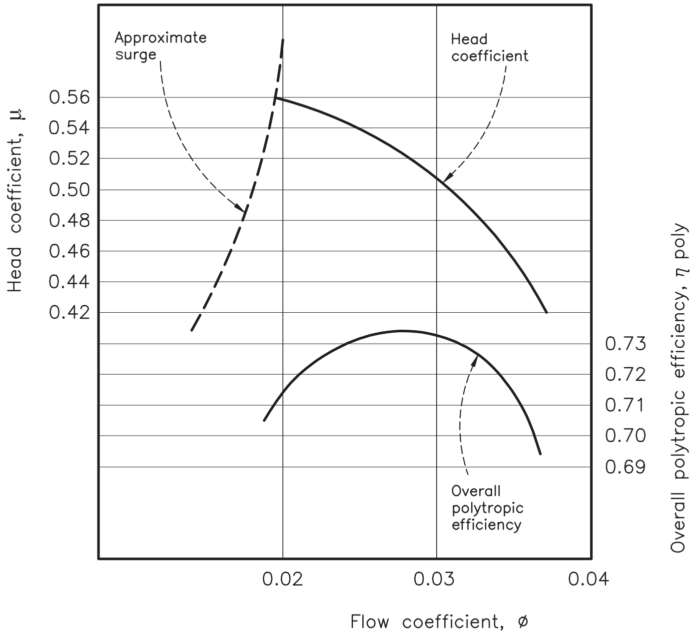


Fig. 8.3 Typical individual impeller curve performance of a centrifugal compressor.

8.1.4 Thermodynamic relationships

As shown in Chapter 7, the thermodynamic head relationships may now be equated as shown in Eqs. (8.3), (8.4).

$$H_{\mathrm{poly}} = \frac{\mu U^2}{g} = Z_{\mathrm{avg}} R T_1 \frac{\left[r^{\frac{n-1}{n}} - 1 \right]}{\frac{n-1}{n}} \tag{8.3}$$

where:

$$Z_{\mathrm{avg}} = \frac{Z_1 + Z_2}{2}$$

As mentioned in Chapter 7, the polytropic process is typically used for centrifugal compressors. Using the relationship for, k, n, η_p, polytropic efficiency is determined as follows:

$$\eta_p = \frac{\frac{k-1}{k}}{\frac{n-1}{n}} \tag{8.4}$$

8.1.5 Component geometry effects performance

The effects resulting from the geometric shape of the principal components of the compressor are shown in Fig. 8.4. Variables such as the impeller configuration and blade angle, inlet guide vane angle, diffuser size and shape, and so on, can be adjusted by the manufacturer for optimum performance under a specified set of operating conditions. Fig. 8.5 shows impeller vector diagrams for various blade angles while Fig. 8.6 illustrates the effects of impeller blade shapes on the compressor's head–capacity performance curve.

Blade shapes are classified as:

- Forward leaning
- Radial
- Backward leaning

Impellers with backward leaning blades are the most commonly used geometric shape impeller used for most centrifugal compressors because they:

- Produce the greatest operating range
- Offer the greatest stable operating range (Fig. 8.7)
- Offer the best efficiency

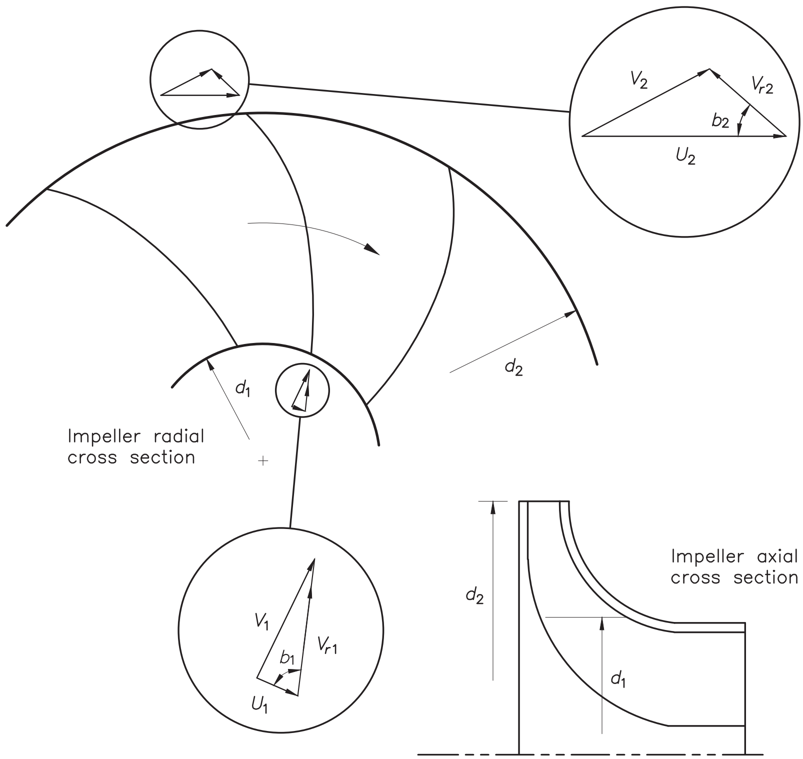


Fig. 8.4 Impeller inlet and outlet flow vector triangles. Adjustments made to impeller, blade angle, inlet guide vane angle, diffuser size and shape, and so on are adjusted to obtain optimum performance under specific operating conditions.
Courtesy of Compressors: Selection and Sizing, by Royce Brown, Gulf Publishing Company, Houston, TX, 1986.

Forward and radial blades are seldom used in upstream and petrochemical applications. Machine output is always affected by combined losses, such as:

- Mechanical loss.
- Aerodynamic loss.
- Friction and shock loss.

Mechanical losses, such as those from journal and/or thrust bearings, affect the power input required, but do not influence the head-capacity curve. Aerodynamic losses that do not influence the shape of the curve consist mainly of wall friction, fluid shear, seal losses, recirculation in flow passages, and shock losses. Shock losses are the result of expansion, contraction, and change of direction associated with flow separation, eddies, and turbulence. Friction and shock losses are the predominant sources of the total aerodynamic losses.

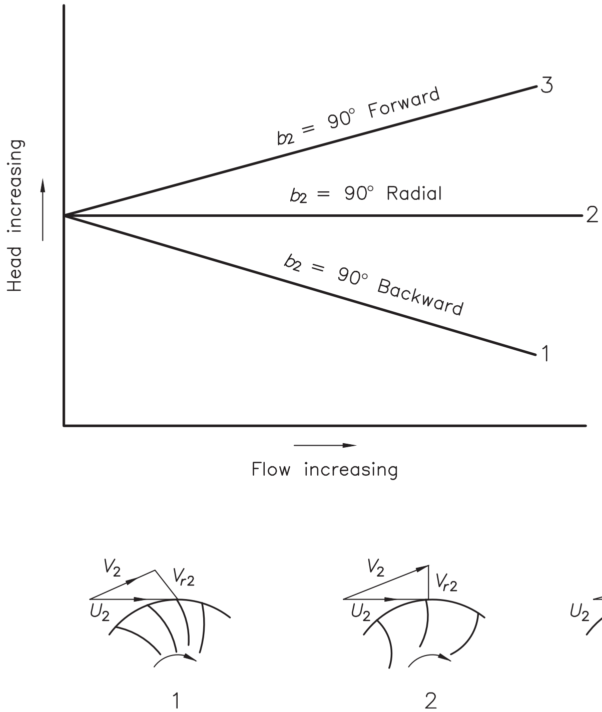


Fig. 8.5 Forward, radial, and backward curved blades.
Courtesy of Compressors: Selection and Sizing, by Royce Brown, Gulf Publishing Company, Houston, TX, 1986.

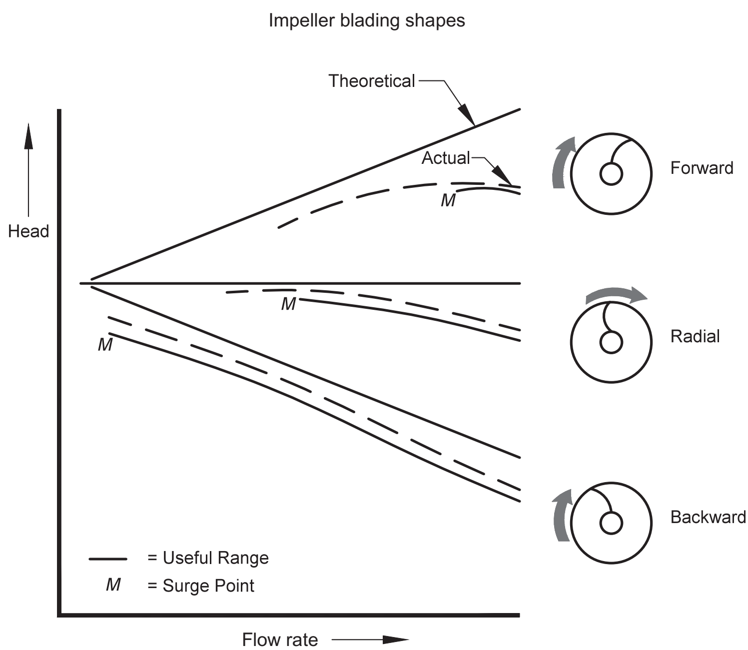


Fig. 8.6 Impeller blade shapes.

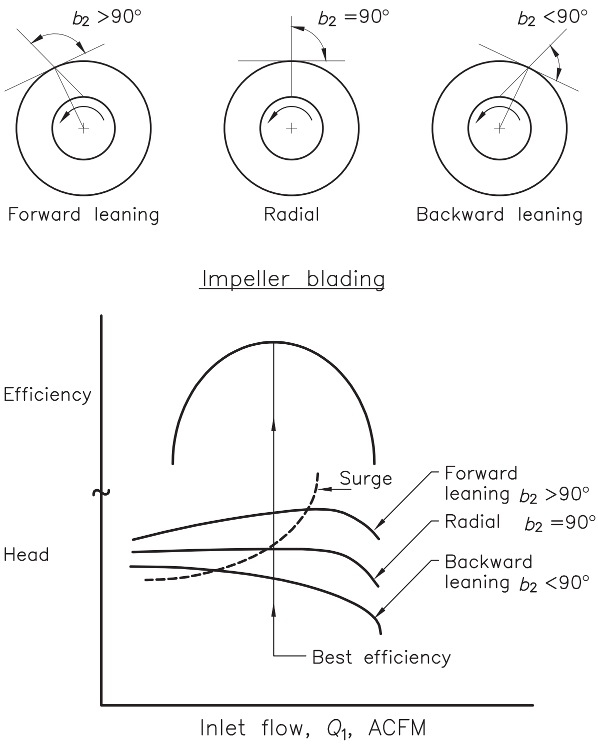


Fig. 8.7 Effect of blade angle on stability.

Fig. 8.8 illustrates the effect of these combined losses in reducing the theoretical head.

Friction losses can be reduced by improving surface finishes. Shock losses may sometimes be mitigated by further streamlining of flow passages. These techniques will improve efficiency and tend to reduce the surge point, but they are costly and there is a point of diminishing returns. Most company specifications do not allow the manufacturer's quoted performance to include efficiency improvements due to impeller polishing.

8.2 Major components

8.2.1 Introduction

Centrifugal compressors are made up of a casing with stationary internals, containing a rotating element, or rotor, supported by bearings. Shaft end seals are provided to contain the process gas. Centrifugal compressors are available in a variety of designs and arrangements. Fig. 8.9 shows a typical multistage compressor and identifies the basic components (refer to Fig. 8.1 for details of the gas flow):

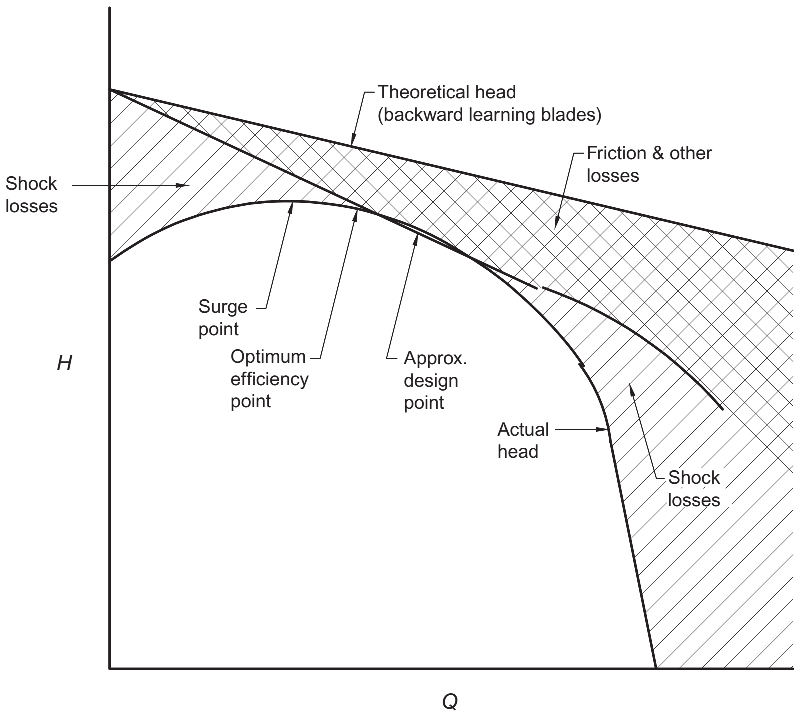


Fig. 8.8 Typical combined losses in reducing compressor head.

- *Housing or case*—contains the rotor assemble and connects last stage diffuser to outlet nozzle
- *Rotor assembly*—supported by bearings and consists of shaft and impellers
- *Impellers*—imparts kinetic energy to the gas, closed and open styles
- *Stationary components*
  - *Diffusers*— reduces the velocity head into pressure
  - *Diaphragms*—stationary element located between stages
  - *Inlet guide vanes*—evenly distributes gas into impeller eye at a certain angle
- *Seals and bearings*—used between rotating and stationary parts to minimize gas leakage between areas of unequal pressures.

8.2.2 Housing or case

The housing or case is the pressure-containing component that encloses the rotor assembly and internal stationary components. The case provides the inlet and outlet flow passages or nozzles. Case construction includes cast, forged or fabricated by welding. There are two types of compressor cases, defined by either:

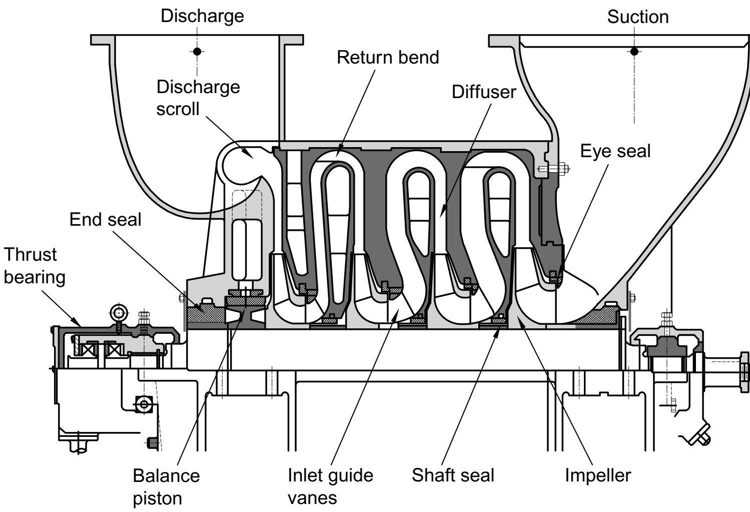


Fig. 8.9 *Top*: Major components of a centrifugal compressor; *Bottom*: Cross-section of a multistage centrifugal compressor.
Courtesy of Demag Delaval.

- Axial, or horizontally split (Fig. 8.10)
- Radial, or vertically split (Fig. 8.11)

API 617, Centrifugal Compressors, requires the use of the vertically split casings when the partial pressure of hydrogen exceeds 200 psi (13.8 bar). Other factors that influence the horizontal/vertical split joint include the absolute operating pressure of the service and ease of maintenance for a particular facility layout.

Horizontal-split compressors have a split along the rotor assembly. This type of casing is identified by noting the large horizontal bolted joint at the compressor centerline. The top half of the horizontally split casing (refer to Fig. 8.12) has the advantage of allowing removal of the top half for access to the rotor without having to remove major process piping. The stationary diaphragms are installed individually in the top and bottom half of the casing. The main process connections may be located either in the top or bottom half. This type of casing is generally selected when a

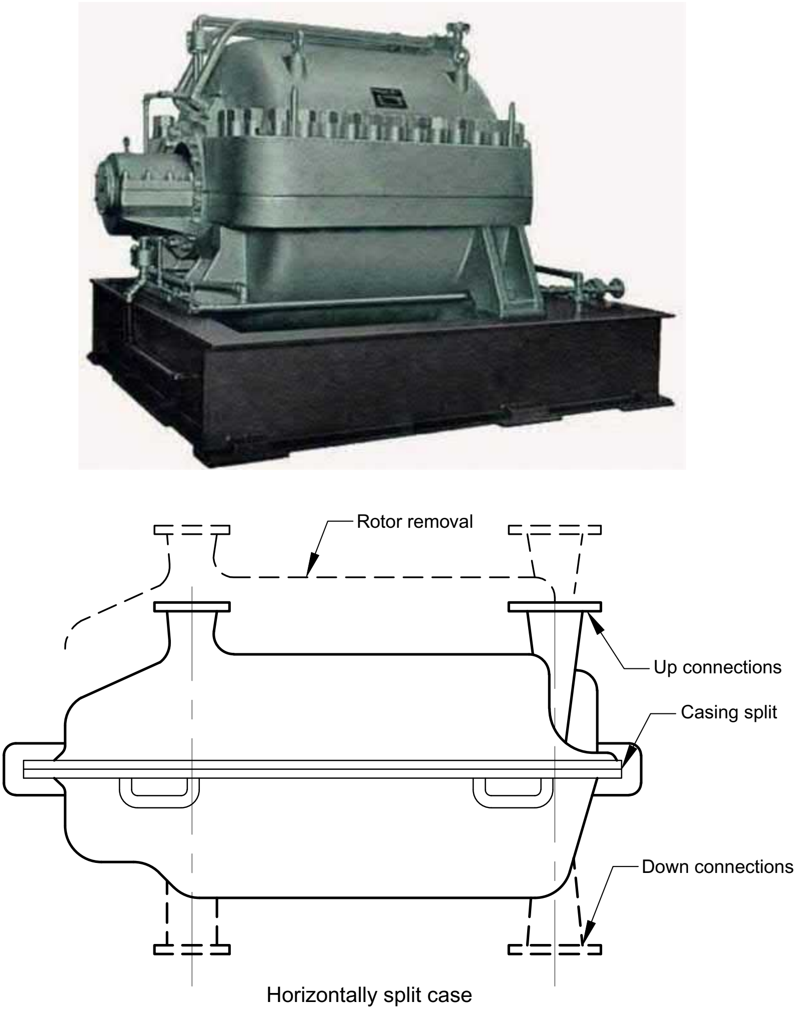


Fig. 8.10 Horizontally split centrifugal compressor. *Top*: Assembled; *Bottom*: Joint Construction.

vertically split casing is not required. The disadvantages of this type of joint include the possibility of leaks between the stages and the inability to handle high pressure due to sealing limitations at the split (Fig. 8.13).

Vertically split or "barrel" compressors have a complete cylindrical outer casing. This type of casing is identified by noting a large vertical bolted joint at one or both ends of the casing. This type of casing is generally selected when:

- The partial pressure of hydrogen in the gas stream exceeds 200 psi (low molecular weight services)

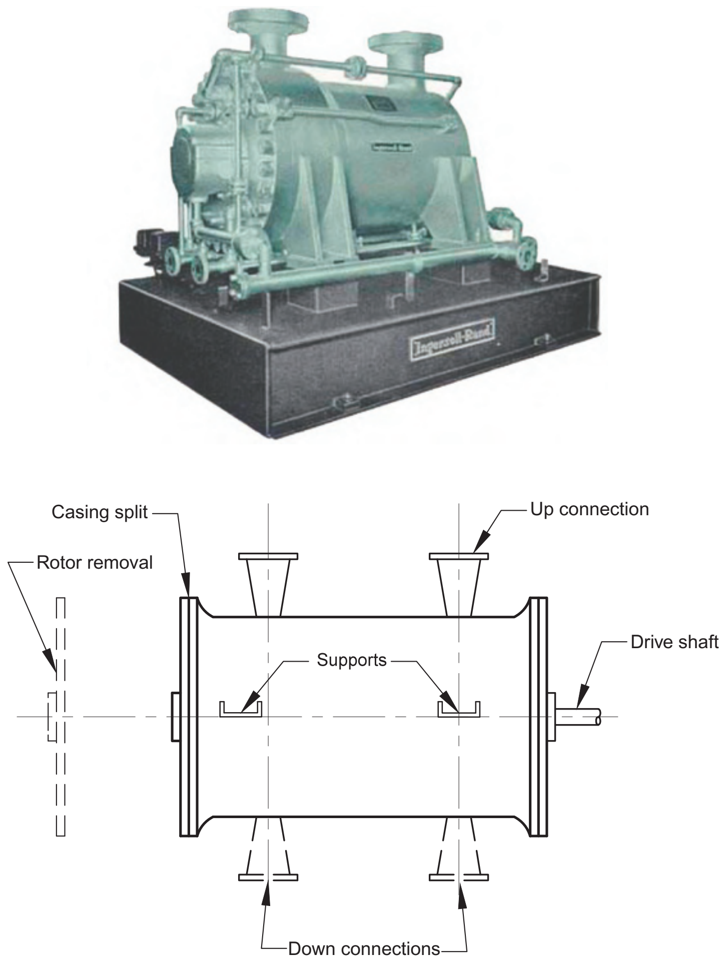
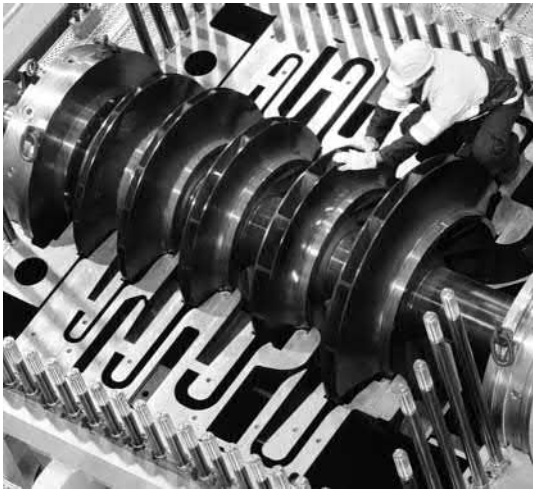


Fig. 8.11 Vertically split (barrel) centrifugal compressor. *Top*: Assembled; *Bottom*: Joint construction.

- The service pressure rating requires the use of the heavy wall casing (high-pressure services)
- Maintenance considerations may have a small influence in the type of casing selected

The stationary diaphragms are assembled around the rotor to make up an inner casing, and installed inside the outer casing as a unit, contained by heads or end closures at each end (Fig. 8.14). Maintenance of the rotor and other internal parts, other than bearings and shaft end seals, involves removal of at least one head, and then dismantling of the inner casing to expose the rotor. The inner casing and rotor can be removed from either the up- or down-connected vertically split outer casing without disturbing process piping. The advantage of the vertically split joint is its high-pressure capability due to the one-piece construction and conventional, proven flange type seals at each end (Fig. 8.15).

Fig. 8.12 Horizontally split casing illustrating accessibility when the top half is removed.



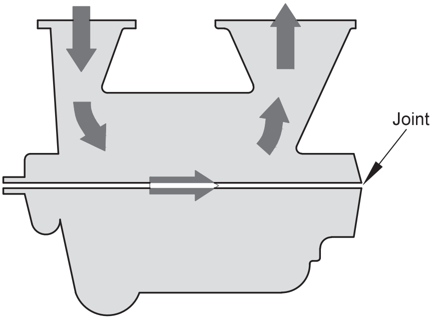


Fig. 8.13 Flow path in a horizontally split casing illustrating joint construction limiting its pressure handling capability.

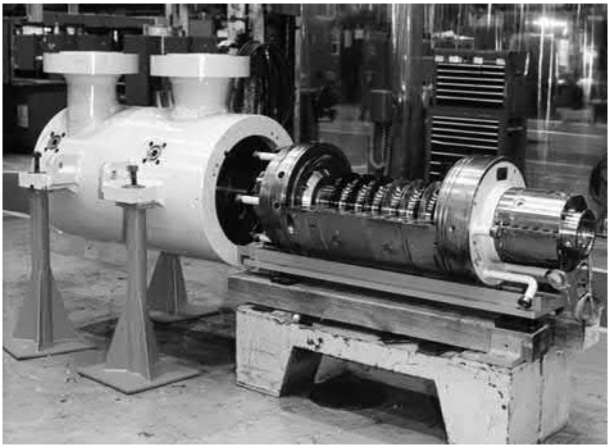


Fig. 8.14 Vertically split or barrel compressor showing complete cylindrical outer casing.

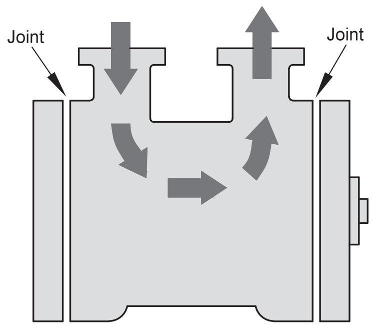


Fig. 8.15 Flow path in a vertically split casing allowing higher pressures since the joint is not in contact with the pressure.

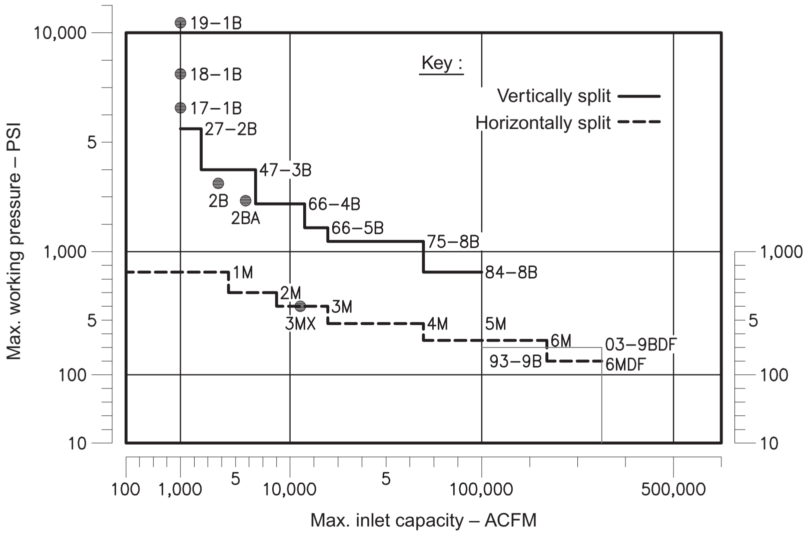


Fig. 8.16 Pressure-capacity comparison chart for horizontally/vertically split compressors. Courtesy of Ingersoll Dresser.

Both the horizontally split and vertically split casing designs allow removal of bearings and shaft end seals for maintenance without disassembly of major casing components.

Fig. 8.16 shows a comparison of pressure versus capacity for multistage horizontally and vertically split casing construction. The size/rating comparisons are general and specific pressure-capacity ranges and casing configurations vary between manufacturers.

8.2.2.1 Casing materials

The following is a summary of casing materials and their applications.

Cast iron

- Limited to low-pressure applications for nonflammable, nontoxic gases
- Limited in location and size of main and sidestream connections to available patterns.

Cast steel

- Quality is difficult to obtain
- X-ray inspection requirements increase costs
- High rejection rate or involved repairs can extend deliveries

Fabricated steel

- Used for both horizontally and vertically split casings
- Improved quality control possible
- Delays associated with rejection or repair of castings are avoided
- Variable stage spacing provides minimum bearing span for required stages.
- Main and sidestream nozzle size and location are not limited by pattern availability

Forged steel

- Used for small vertically split casing sizes where application involves very high pressures.

All centrifugal compressor casings used to be cast, but due to problems associated with quality control on large castings, coupled with improved fabrication techniques and costs, many manufacturers converted to fabricated steel casings, especially on the larger frame sizes.

8.2.2.2 Nozzles

Inlet and outlet nozzles are available in a variety of configurations, depending on the manufacturer. They are normally flanged. (Typical arrangements are shown later in this section.) API 617 covers requirements for flange type and ratings of main and auxiliary connections.

The increased use of fabricated cases has provided additional flexibility in nozzle orientation. If the installation permits, the following should be considered:

- Horizontally split units with process connections in the lower half (down-connected) allow removal of the top half, and internals including rotor, without disturbing the process piping.
- If overhead process piping is required, the use of vertically split barrel compressor casings still allow removal of the inner casing and access to the internals without removing process piping. Fabricated casing design makes the vertically split unit a cost-effective alternative for larger medium pressure applications.

8.2.2.3 Stage

The heart of the centrifugal compressor is the impeller “stage.” The stage is made up of the following parts:

- Inlet guide vanes
- Impeller
- Diffuser

- Return passageway (return bend or crossover)
- Return channel

The stage can be separated into two major elements:

- The impellers which are mounted on the shaft as part of the rotor
- The stationary components including the inlet nozzle and other components mentioned before

8.2.3 Rotor assembly

The rotor assembly is composed of a shaft, impeller(s), impeller spacers, thrust collar, and the balancing drum. It may include shaft sleeves, floating seal parts, and thrust collar. The shaft is commonly referred to as the rotor. Fig. 8.17 shows a typical multistage rotor assembly. The two basic rotor designs used are (refer to Fig. 8.18)

- Solid-shaft rotor
- Modular-shaft rotor

The solid-shaft rotor is built with the impellers keyed and/or press-fit onto a solid center shaft—commonly referred to as a "stiff" shaft configuration. The modular rotor is constructed by stacking each impeller, and spacers when required, on each other and clamping together with a center bolt—commonly referred to as a "*built-up*" or flexible shaft.

Shaft sleeves position the impeller along the shaft and function as wearing surfaces for interstage seals. They isolate the shaft from the process gas.

If a rotor always operates below the lowest critical speed, it is known as a *stiff-shaft* rotor. In contrast, a rotor with a normal operating range above one or more of its critical speeds is a flexible shaft. Most multistage centrifugal compressors have *flexible-shaft* rotors; and therefore, must pass through at least one critical during start-up

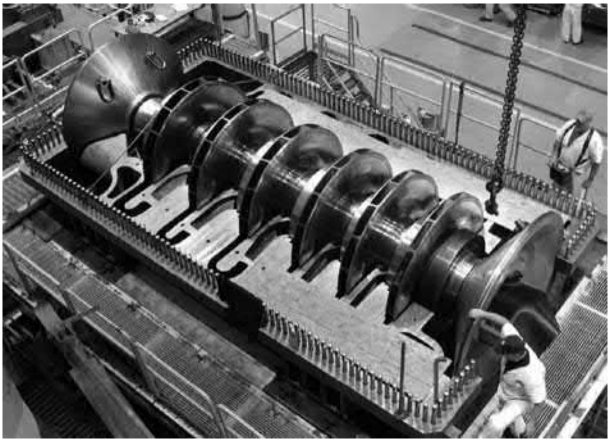


Fig. 8.17 Typical multistage rotor assembly.

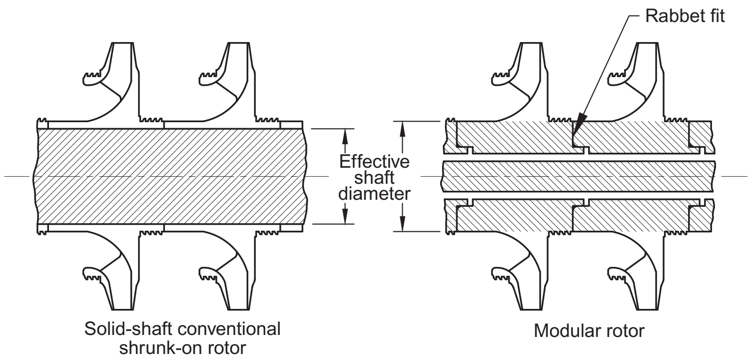


Fig. 8.18 Solid and modular rotor designs.

or shutdown. From an operational point of view, stiff shafts would be preferable. However, it is not practical since the shafts would become prohibitively large.

Shafts are made from alloy steel forgings, finished by grinding or honing to produce the required finish. Special requirements are detailed in *API 617* for balancing and concentricity during rotor assembly. Impellers are normally mounted on the shaft with a shrink fit with or without a key, depending on the particular manufacturer and compressor frame size. Most manufacturers use shaft sleeves to both locate impellers and provide protection for the shaft in the event of contact with internal labyrinth seals.

8.2.4 Impellers

Impellers are the components in a centrifugal compressor that perform work on the gas to be compressed. Since they are rotating parts with fairly high vane tip velocities, they are also the most highly stressed components in a centrifugal compressor. As such, impeller construction is a subject of interest. The impeller(s) is the part of the rotor assembly that adds velocity to the gas. Centrifugal compressor performance is dependent upon the impeller(s). The shape of the impeller affects the head versus flow characteristic "performance" curve as shown previously in Fig. 8.6. The open impellers are classified as:

- *Open wheel*: Have vanes positioned in a radial direction and have no enclosing covers on either the front or back sides (Fig. 8.19). The open wheel is used for medium flow rates and high discharge heads (30,000 to 60,000 ft.) [10,000 to 20,000 m]. They are only used in single-stage compressors. They require close clearance between impeller and inlet shroud. Natural frequencies of the blades can be a problem. Not typically used in the process industry except for air compression or occasionally as the first stage of a multistage compressor. Open impeller wheels can be cast and machined, or in many cases are fully machined from solid material.
- *Semienclosed (closed) wheel*: Usually have the vanes positioned in a radial or backward leaning direction and have a cover on the backside which extends to the periphery of the vanes. The radial blade, semiopen impeller provides for a maximum amount of flow and head in a single stage, even in large diameter impellers. The semienclosed wheel is used

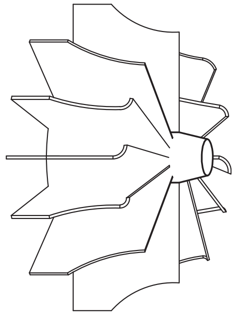


Fig. 8.19 Open impeller.

for high flow rates. This type of wheel can be used to equip multistage or single-stage compressors (refer to Fig. 8.20A–C).

- *Enclosed (closed) wheel*: Have enclosing covers on both the front and backside (Fig. 8.21). Originally, the covers on enclosed impellers were riveted. Riveted impellers had obvious problems with potential failure of the rivets. Welded covers were more rugged, but the weld fillet reduced the flow area of the impeller. Very small impellers could not be fillet welded. The enclosed wheel is mainly used in multistage compressors.

Impellers come in enclosed (closed), semienclosed (open), and open configurations. By saying that the impeller is closed, one means that there is a shroud on the face of the impeller so the blades are exposed. In a closed impeller, the gas flow path can be more easily sealed by maintaining a close clearance at the ring surface near the eye of the impeller. The gas path in open impellers is sealed by maintaining a tight clearance between the blade face and the casing.

Open and semiopen impellers are generally found in smaller higher running speed compressors, where it would be difficult to manufacture the small gas passage through a closed impeller of corresponding size. As an example, integrally geared centrifugal air compressors have semiopen impellers. Open impellers are often used for large flow rates in single-stage compressors. Semiopen impellers are used for large flow rates in single-stage compressors and as the first stage in a compressor. The closed impeller is used mainly in multistage compressors. The backward leaning, closed impeller is the most commonly used impeller because of its wide flow range capability.

Single-inlet impellers take the gas in an axial direction, on one side of the impeller only, and discharge the gas in a radial direction.

Double flow impellers take the gas in an axial direction, on both sides of the impeller, and discharge the gas in a radial direction. They are, in effect, the equivalent of two single-inlet impellers placed back to back and, in general, will handle twice the flow at the same head as a single-inlet impeller of the same diameter operating at the same speed.

Some impeller designs utilize a three-dimensional blade or vane configuration, which varies the inlet blade angle from hub to outside diameter, thereby providing


(A) (B)

Fig. 8.20 (A) *Top*: A fabricated semienclosed impeller; *Bottom*: A semienclosed impeller with machined blading integral with the back plate. (B) *Top*: Semiopen impeller installed on a multistage centrifugal compressor; *Bottom*: Damaged semiopen impeller on the first stage of a multistage centrifugal compressor.
(A) Top (Courtesy of GE Energy); Courtesy of Dresser-Rand.

optimum aerodynamic geometry and improved performance over that of two-dimensional designs.

Centrifugal compressor impellers discharge gas radially, but the gas enters in an axial direction. An axial flow element called an inducer is sometimes incorporated into the impeller. This combination is called a mixed flow impeller. This configuration results in increased efficiency in high-flow applications.

Impellers may be fabricated (riveted, brazed, or welded) or machined from precision castings. Open impellers can be cast and machined, or in many cases are fully machined from solid material (refer to Fig. 8.22).

Originally, the covers on closed impellers were often riveted together, the vanes riveted to the back plate/hub and shroud. Riveted impellers had obvious problems with potential failure of the rivets. This limited the tip speed. Some of these impellers can still be found in older plants. Today, closed impellers are either cast and welded or fully machined and welded. In either case, the vanes and back plate/hub are cast or fully machined as one piece and the shroud is welded to the vanes. Welded covers are more rugged, but the weld fillet reduced the flow area of the impeller. In addition, very small impellers with gas passages less than 5/8” wide normally cannot be fillet

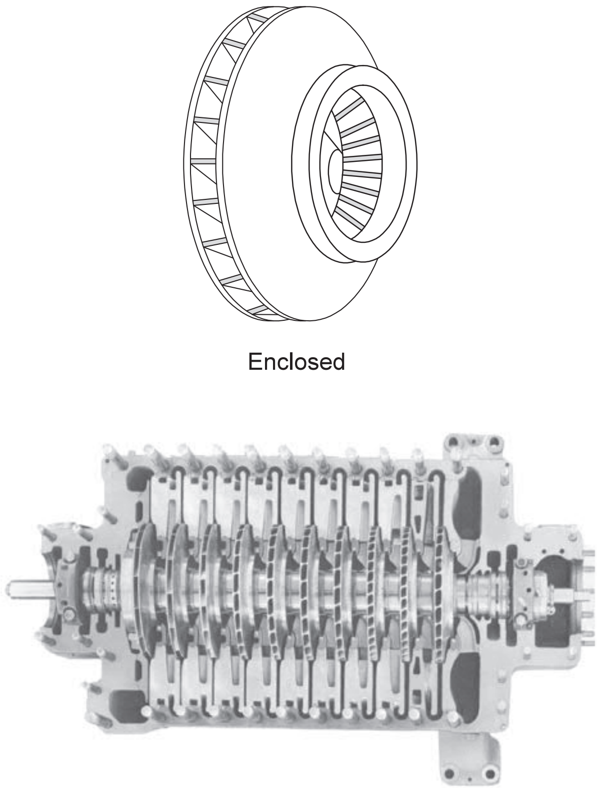


Fig. 8.21 *Top*: Enclosed Impeller; *Bottom*: Multistage rotor assembly with enclosed impellers.

welded. An alternative solution was to mill a slot in the impeller cover and then place the cover over the impeller vanes. The slot is then filled with weld material from above. Closed impellers are limited to 10,000 to 14,000 lb_f-ft/lb_m per impeller. Natural gas compressor impellers are commonly made from stainless steel, but in general, the impeller construction will be a function of the type of service.

8.2.5 Stationary components

As shown in Fig. 8.23, the internal stationary components consist of:

- Diaphragms
- Guide vanes
- Turning vanes

During compressor assembly, the diaphragms and guide vanes are bolted together to form what is referred to as stators. The stator forms a diffuser passage for the gas after each impeller. The turning and directing of the gas flow through the compressor is achieved by inlet and exit guide vanes and walls of the casing. Guide vanes located between impellers are sometimes called "turning vanes." The stationary parts are normally cast of ductile iron or steel.

Fig. 8.22 *Top*: Welded, riveted, and milled impellers; *Center*: Left-riveted impellers, Right-damaged riveted impeller; *Bottom*: Fillet-welded closed impellers.

8.2.5.1 Single stage

A single-stage centrifugal compressor is defined as the combination of one impeller with its associated inlet guide vane and diffuser (Fig. 8.24). Velocity is converted into pressure within the diffuser and then further increased as the gas passes through

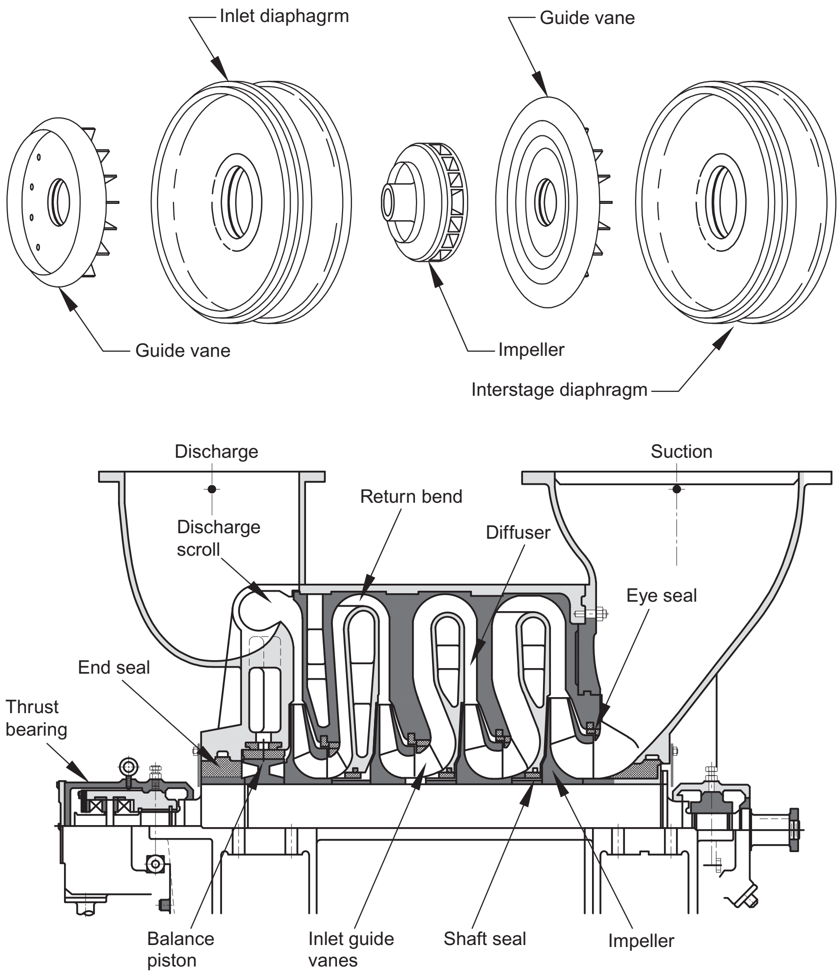


Fig. 8.23 Stationary components of a centrifugal compressor.

the volute. As shown in Fig. 8.25, the velocity head that a compressor develops represents the height a column of gas is lifted.

8.2.5.2 *Multistage*

A multistage centrifugal compressor consists of several stages (Fig. 8.26). The number of stages is limited by the case size and the stability of the rotor assembly. As shown in Fig. 8.27, as gas leaves the first impeller, it gains some velocity. The increased velocity is partially converted into pressure within the diffuser. As the gas leaves the diffuser, it enters the return passageway which guides it into the eye of the next impeller.

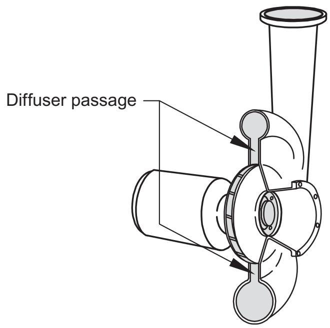


Fig. 8.24 Schematic illustrating a single-stage compressor consisting of a single impeller and diffuser

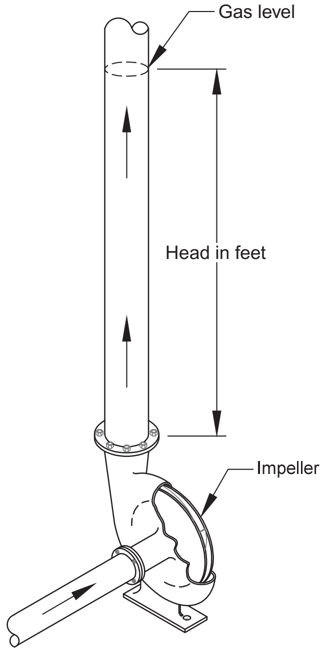


Fig. 8.25 Schematic illustrating the head developed (height a column of gas is lifted) by a centrifugal compressor which represents the height a column of gas is lifted.

8.2.5.3 Diaphragm

As shown in Fig. 8.28, the diaphragm is the stationary element inside the casing located between stages. The diaphragm includes a diffuser for the gas as it leaves the impeller, and a channel to redirect the gas through the return bend and return channel or passageway into the next stage. Diaphragms can be either cast or fabricated, with cast diaphragms normally made from iron.

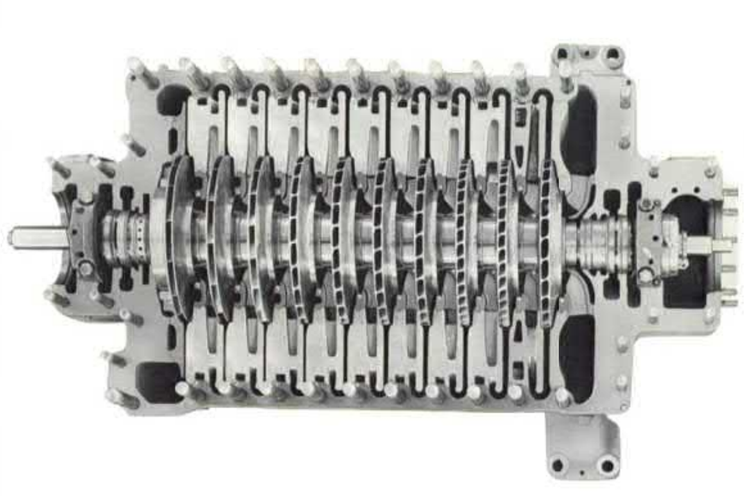


Fig. 8.26 Cutaway of a multistage centrifugal compressor.

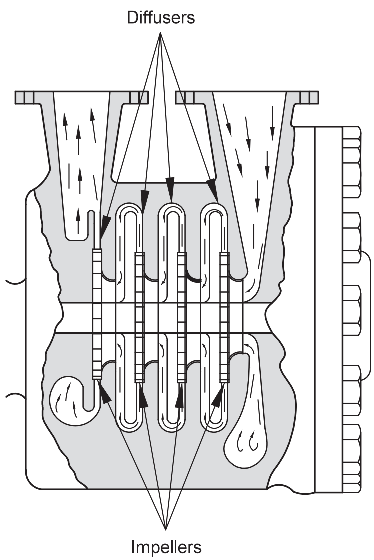


Fig. 8.27 Cutaway of a four-stage centrifugal compressor showing the impellers, diffusers and return passageways.

Normally, diaphragms are not exposed to high-pressure differentials, and therefore are not highly stressed. Diaphragms should be made of steel where high differentials may exist, such as back-to-back impellers. The diaphragm may also include guide vanes for directing gas to the next impeller. Some diaphragms are liquid cooled as shown in Fig. 8.29. The inside of the diaphragm consists of an intake and discharge tube through which coolant (water) can flow.

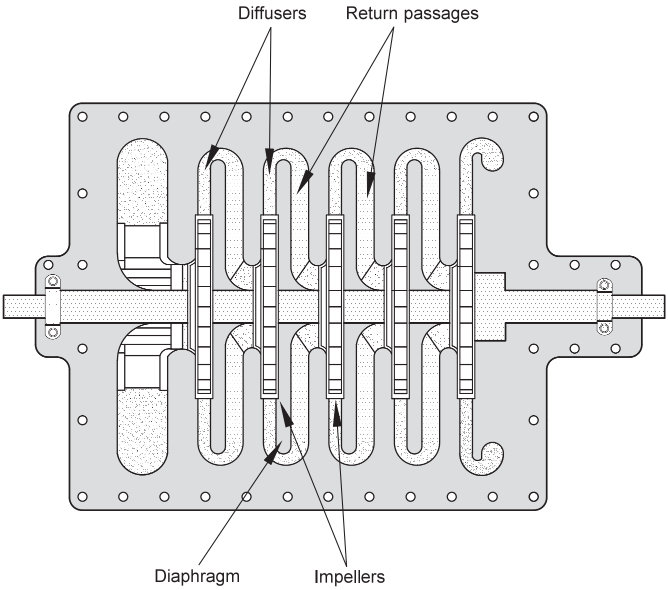


Fig. 8.28 Cutaway of a multistage centrifugal compressor illustrating diffusers, return passages, diaphragm, and impellers.

Adjacent walls of the diaphragm form the diffuser (refer to Fig. 8.30) where velocity energy imparted by the impeller is converted to static pressure.

8.2.5.4 Inlet guide vanes

Inlet guide vanes may be either fixed or variable. They are usually located ahead of each impeller eye (Fig. 8.31) and are designed to guide the flow of gas efficiently into the eye of the impeller. Most process compressors use fixed since most vanes usually corrode/foul and jam. Many air compressors use variable on the first stage.

Adjustable inlet guide vanes, shown in Fig. 8.32, change the angle of gas flow into the eye of the impeller. They are used to change the slope of the compressor performance curve. The effect of inlet guide vanes is shown in Fig. 8.33. For example, guide vanes that add "preswirl" counter to the impeller rotation will increase head and reduce the slope of the curve while guide vanes adding "preswirl" with the impeller rotation will decrease head and increase the slope of the curve.

8.2.6 Seals and bearings

8.2.6.1 Shaft seals

Centrifugal compressors use shaft end seals to:

- Restrict or prevent leakage of air or oil vapors into the process gas stream.
- Restrict or prevent leakage of process gas from inside the compressor.

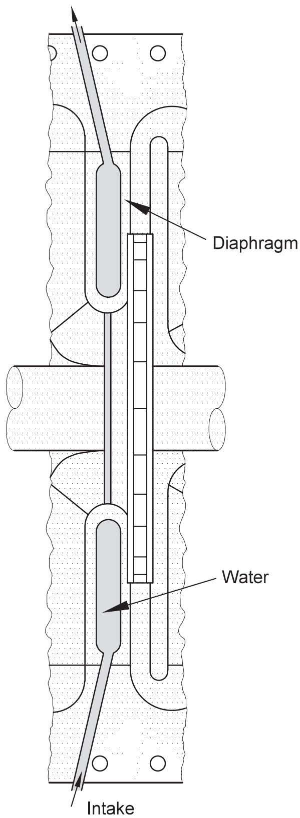


Fig. 8.29 Cutaway of a liquid-cooled diaphragm.

Various types of seals are used, depending on the gas being compressed, the pressures involved, safety, operating experience, power savings, and process requirements.

Shaft end seals are separated into broad categories:

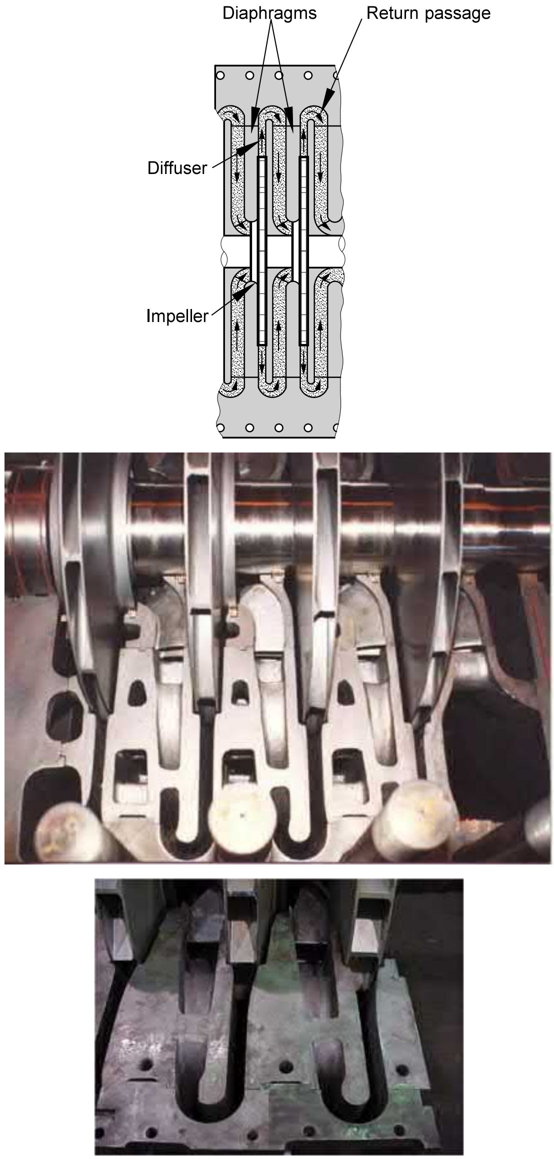
- *Restrictive seal*—restricts but does not completely prevent leakage
- *Positive seal*—designed to prevent leakage

Restrictive seals are usually *labyrinths*. They are generally limited to applications involving nontoxic, noncorrosive, abrasive-free gases at low pressures. In some cases, ports for injection or withdrawal of the gas is used to extend the range of effectiveness.

Another form of the restrictive seal is the *dry carbon ring seal* or *"restrictive" seal ring*, often used on overhung single-stage compressors where maximum sealing and minimum axial shaft spacing are important. Since this seal can be held to close clearances, leakage is less than with the labyrinth seal. Also, less axial shaft space is required.

Positive seals, while varying somewhat in design between manufacturers, are either *liquid-film* or *mechanical contact type*.

Fig. 8.30 *Top*: Cutaway illustrating the adjacent walls of a diaphragm forming the diffuser; *Middle*: Horizontal split bottom case with rotor; *Bottom*: Diffuser/return bend.



Shaft seals are devices used between rotating and stationary parts to minimize gas leakage between areas of unequal pressures. Pressure sealing around the rotor shaft is accomplished by the combination of the following:

- Restrictive seals
  - Labyrinth
  - Dry carbon “restrictive” seal rings

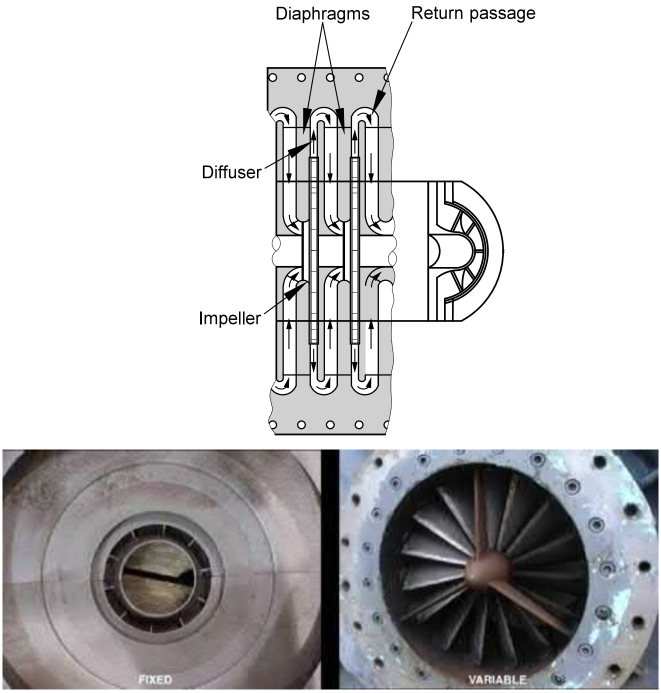


Fig. 8.31 *Top*: Cutaway showing inlet guide vanes located ahead of each impeller eye; *Bottom*: Left-fixed and right-variable guide vanes.

- Positive seals
  - Liquid film
  - Mechanical contact

8.2.6.1.1 Labyrinth seal

The labyrinth seal is a controlled leakage seal used to prevent high-pressure gas leaking out of the case to the atmosphere and minimizes internal leakage. The labyrinth seal consists of one or more sharp-edged ridges machined on the outside diameter of a rotating shaft or on the inside diameter of a sealing passage as shown in Fig. 8.34.

The labyrinth seal may be either stationary or rotating. Sealing is achieved by the pressure drop across each ridge as a small amount of gas is allowed to leak. The sealing action is a result of flow resistance by throttling across the labyrinth teeth as shown in Figs. 8.35 and 8.36.

Labyrinth seals are used as:

- *Eye seals*: Reduces internal leakage from impeller tip back to the eye
- *Shaft seals*: Reduces leakage from impeller eye back to upstream stage and between stages
- *End seal*: Contains process gas within the case

Internal seals are installed on multistage centrifugal compressors to prevent leakage between stages, thereby improving performance. Labyrinth seals are commonly used,

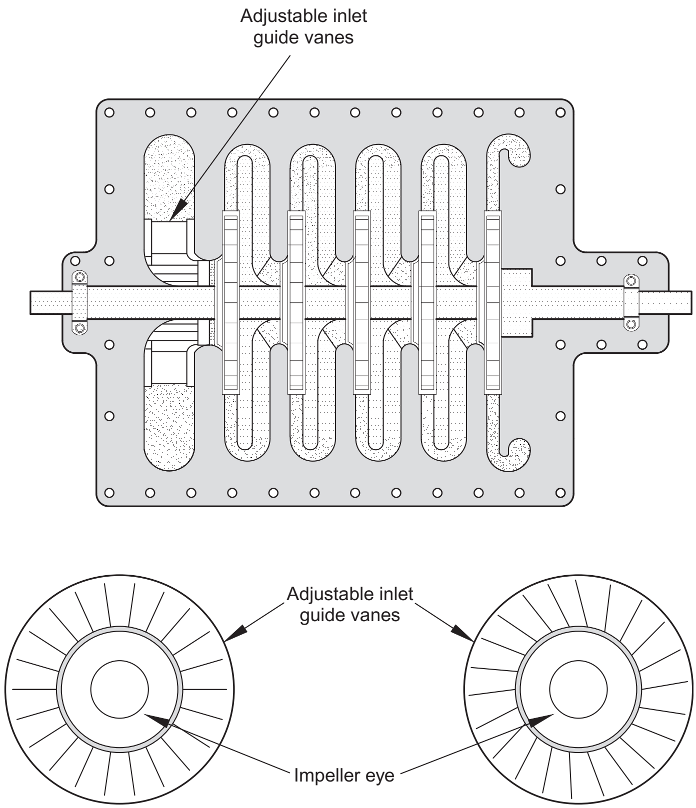


Fig. 8.32 Cutaway of a centrifugal compressor illustrating adjustable guide vanes.

being located at the impeller eye and at the shaft between stages. At high pressures, an interlocking labyrinth seal is used. The shaft has teeth that interlock with the seal (Fig. 8.37). Seals not only reduce leakage between the shaft and diaphragm but also between the shaft and casing as shown in Fig. 8.38. When handling toxic gases the seal must be ported and flushed with inert gas (Fig. 8.39). The port is placed on the seal between the process gas and the atmosphere. The inert gas is forced into the labyrinth of the seal.

8.2.6.1.2 Dry carbon "restrictive" seal rings

Dry Carbon "Restrictive" seal rings consist of flat rings mounted on a stuffing box. The rings are held in position around the shaft by stationary ring cups as shown in Fig. 8.40. Leakage through the ring is prevented by the vertical contact between the ring and the ring cup. Rings do not actually contact the shaft, thus some leakage will occur. Restrictive seal rings are only used when the gas is clean and free of debris.

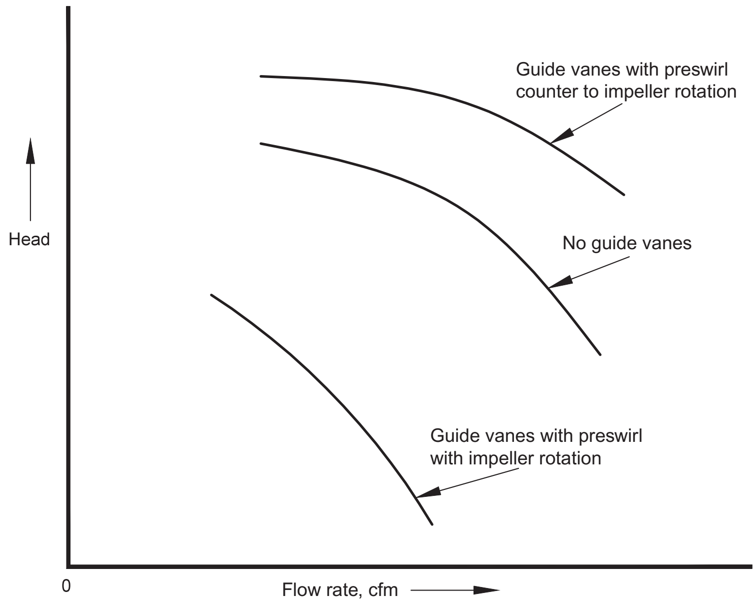


Fig. 8.33 Effect of inlet guide vanes.

The seal rings are made out of low friction material such as carbon. Their design can be either one piece or segmented. If segmented they are held together with a garter spring (Fig. 8.41).

8.2.6.1.3 Liquid film seals

Liquid film seals are often used in combination with the labyrinth seal. The liquid film seal consists of inner and outer floating carbon seal rings that do not touch each other or the shaft (Fig. 8.42). Sealing is accomplished by injecting oil between the rings at a pressure slightly higher than the gas pressure in the case (Fig. 8.43). Sealing oil is fed to the seal from an overhead tank located at an elevation above the compressor set to maintain a fixed, typically 5 psi, differential above "seal reference" pressure (seal reference pressure is very close to suction pressure).

The oil enters between the seal rings and flows in both directions to prevent inward leakage to the process gas or outward leakage of the gas to the atmosphere. "Buffer ports" are often available for injection of an inert gas to further ensure separation of the process from the sealing medium. The liquid film seal requires a continuous supply of high-pressure oil and a system for circulating the oil and maintaining pressure. They are typically suitable for sealing pressures in excess of 3000 psi.

8.2.6.1.4 Mechanical contact seals

Mechanical seals are typically used for pressures up to 1000 psi. They have the added feature of providing more positive sealing during shutdown. They rely on the continuous contact between a rotating and fixed seat that are separated by a carbon ring (refer

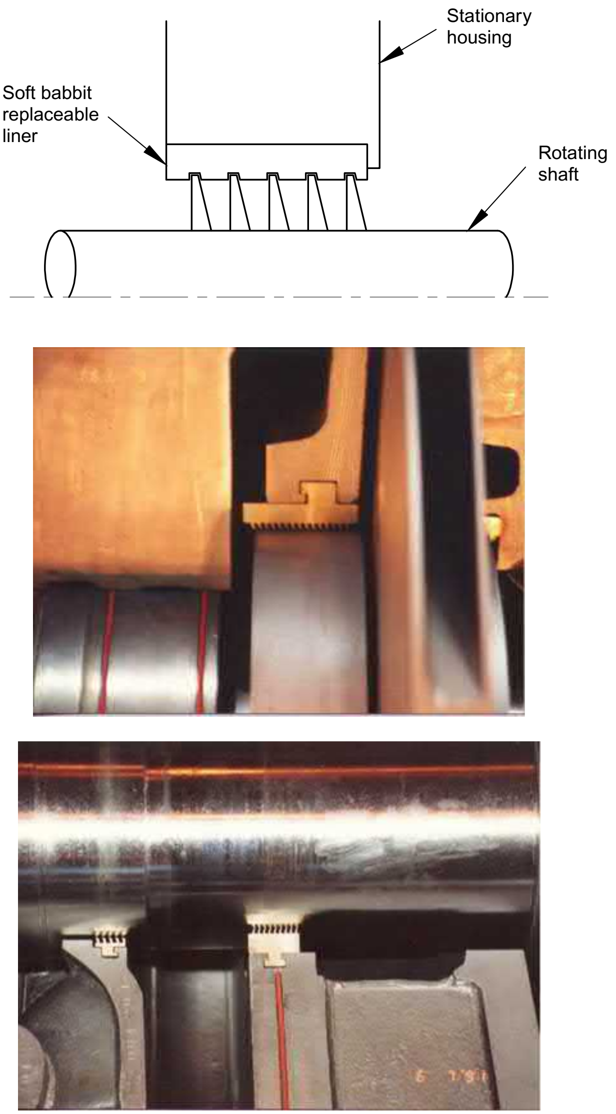


Fig. 8.34 *Top*: Schematic of a labyrinth seal; *Middle*: Labyrinth seal used as an eye seal; *Bottom*: Labyrinth seal used as a shaft seal.

to Fig. 8.44). Sealing is accomplished by vertical contact between the carbon ring and the rotating and stationary seat (Fig. 8.45). Seal faces require lubrication to reduce friction and thus to heat and cool the metal. The seal oil functions primarily as a coolant. Lubrication is accomplished by injecting oil through a hole in the seal housing at a pressure of about 40 psi higher than the gas pressure in the seal zone. Seal oil differential is controlled by a regulator rather than an overhead tank.

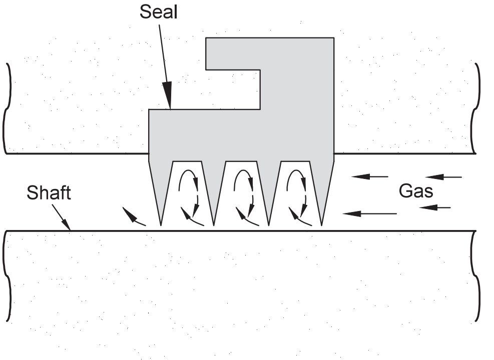


Fig. 8.35 Cutaway illustrating the sealing action of a labyrinth seal.

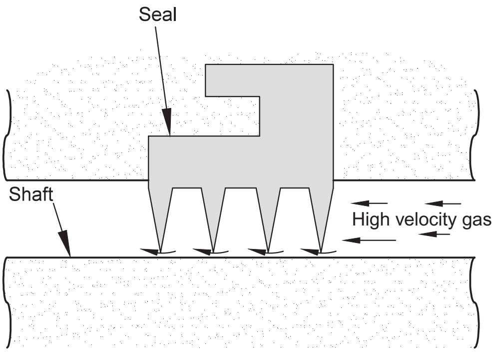


Fig. 8.36 Cutaway of an labyrinth seal leaking at high gas velocities.

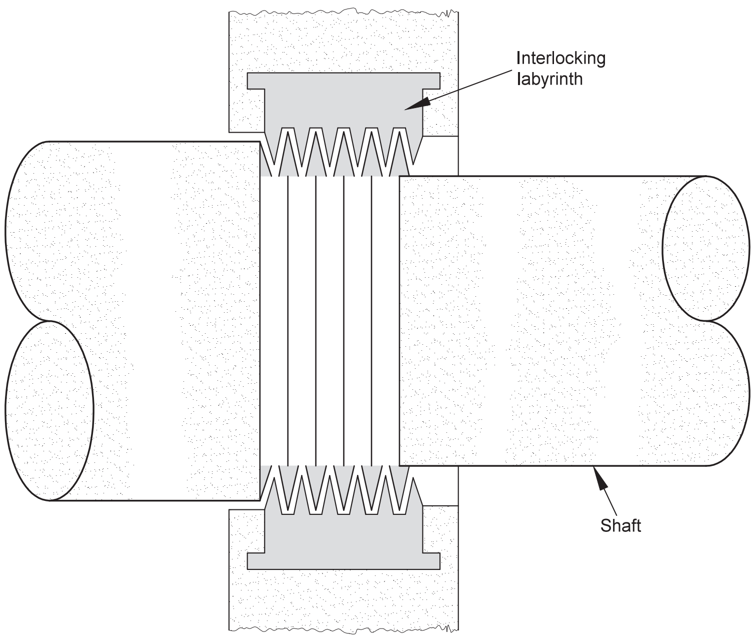


Fig. 8.37 Cutaway of an interlocking labyrinth seal used in high-pressure service.

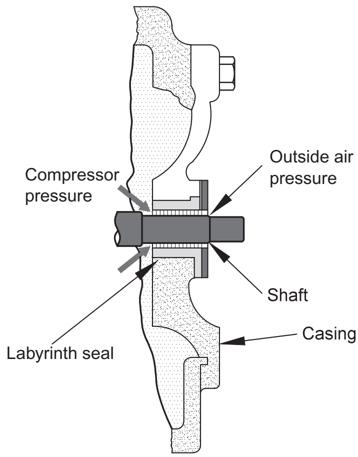


Fig. 8.38 Cutaway illustrating a labyrinth seal used to reduce gas leakage between shaft and casing.

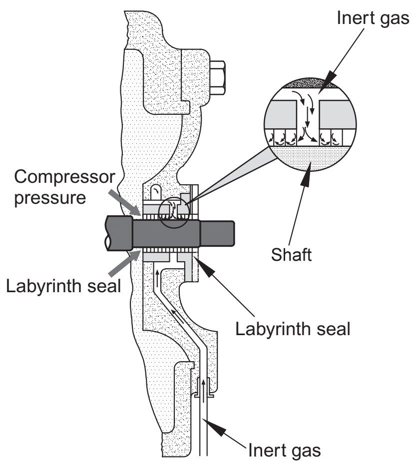


Fig. 8.39 Cutaway of a labyrinth seal used in toxic service.

8.2.6.2 Seal oil and buffer gas systems

The seal oil system prevents gas from leaking out of the compressor case. Fig. 8.46 is a schematic of a typical seal oil system working in combination with a buffer gas system. These systems require auxiliary equipment such as pumps, regulators, filters, and so on for proper operation. The oil that flows across the outboard ring mixes with the

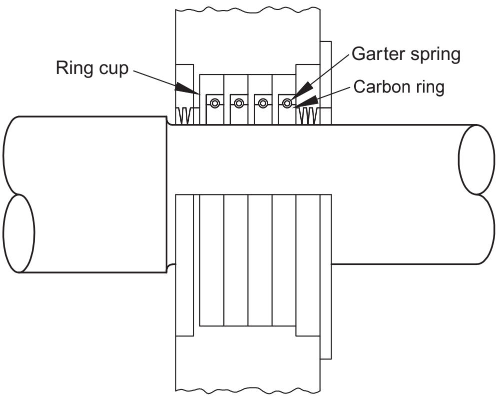


Fig. 8.40 Cutaway of a restrictive seal ring held in position by stationary ring cups.

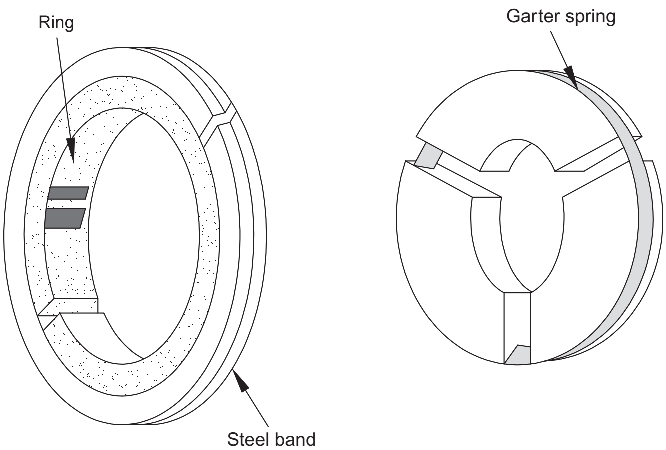


Fig. 8.41 One-piece and segmented seal ring design.

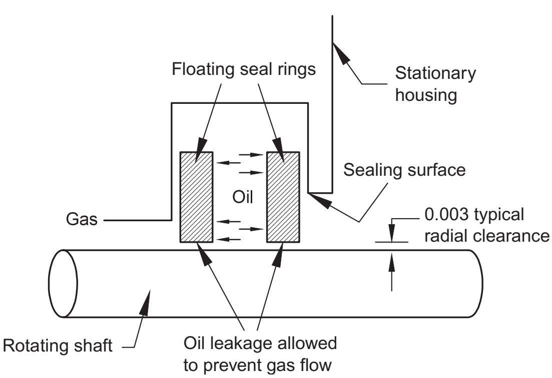


Fig. 8.42 Cutaway illustrating a liquid film seal incorporating carbon seal rings.

Fig. 8.43 Three-dimensional view illustrating a liquid film assembly.

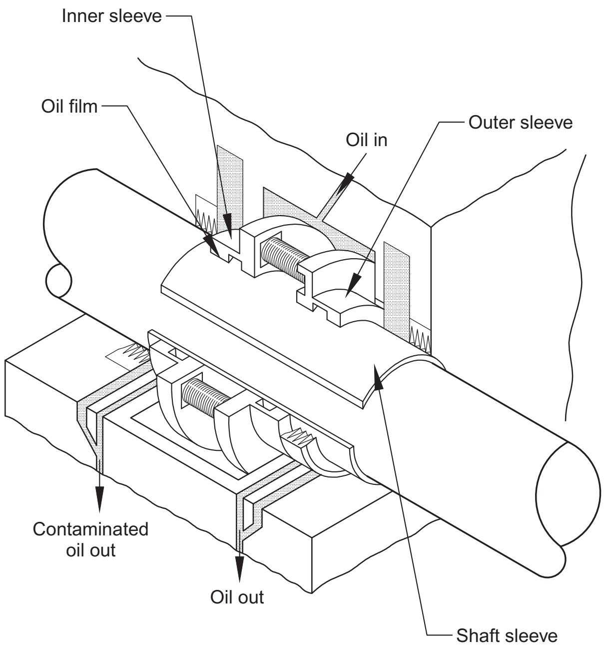
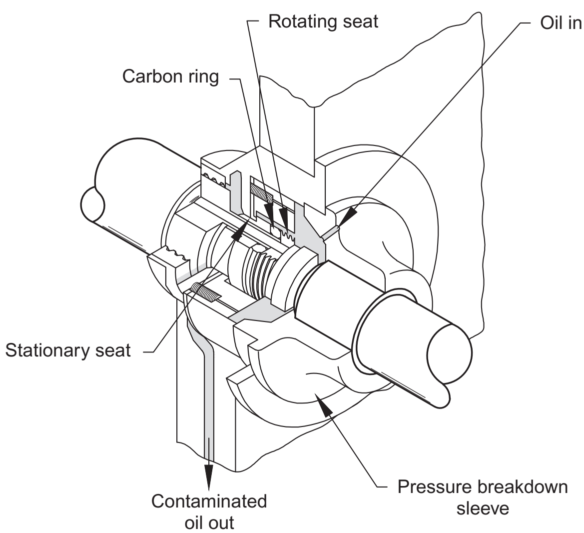


Fig. 8.44 Three-dimensional view of a mechanical seal.



bearing lube oil and returns to a lube oil tank. On the inboard side of the rings, the seal oil mixes with the buffer gas. Since the pressure at each end of the compressor case is at or near suction pressure and maintaining the buffer gas at a pressure above the compressor suction pressure and seal oil pressure above buffer gas pressure, leakage of gas from the case and leakage of seal oil into the case are prevented.

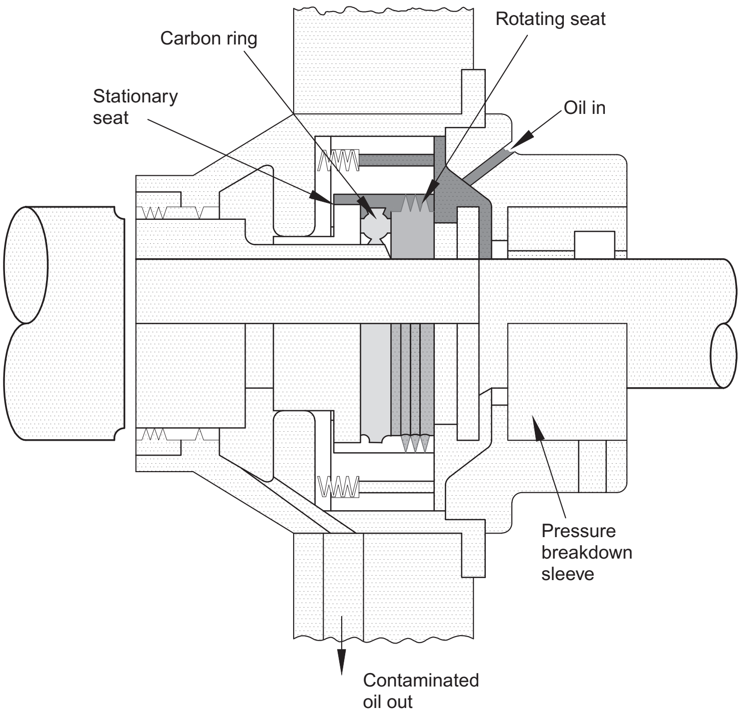


Fig. 8.45 Cutaway of a mechanical seal illustrating the vertical contact between the carbon ring, rotating and stationary seals.

8.2.6.3 Lube oil system

The compressor lube oil system is often common to the driver lube system but in some cases, separate lube systems are supplied. Fig. 8.47 shows a typical common gas turbine compressor lube oil system.

8.2.6.4 Balancing drum (piston)

The balancing drum is used to equalize the axial thrust in the direction of the suction end of the compressor that is created by the impellers. Fig. 8.48 illustrates the differential pressure across a multistage centrifugal compressor. By using a balancing drum, both ends of the compressor case are near the suction pressure. The balancing drum (piston) is a drum larger in diameter than the rotor shaft, with a series of labyrinth seals to allow discharge pressure on one side to leak to suction pressure on the other side (refer to Fig. 8.49).

The balancing drum (piston) is installed at the discharge end of the rotor where one side of the drum is at discharge pressure and the other side is vented to the inlet pressure. The pressure on the vented end is the same as the suction pressure. Pressure on

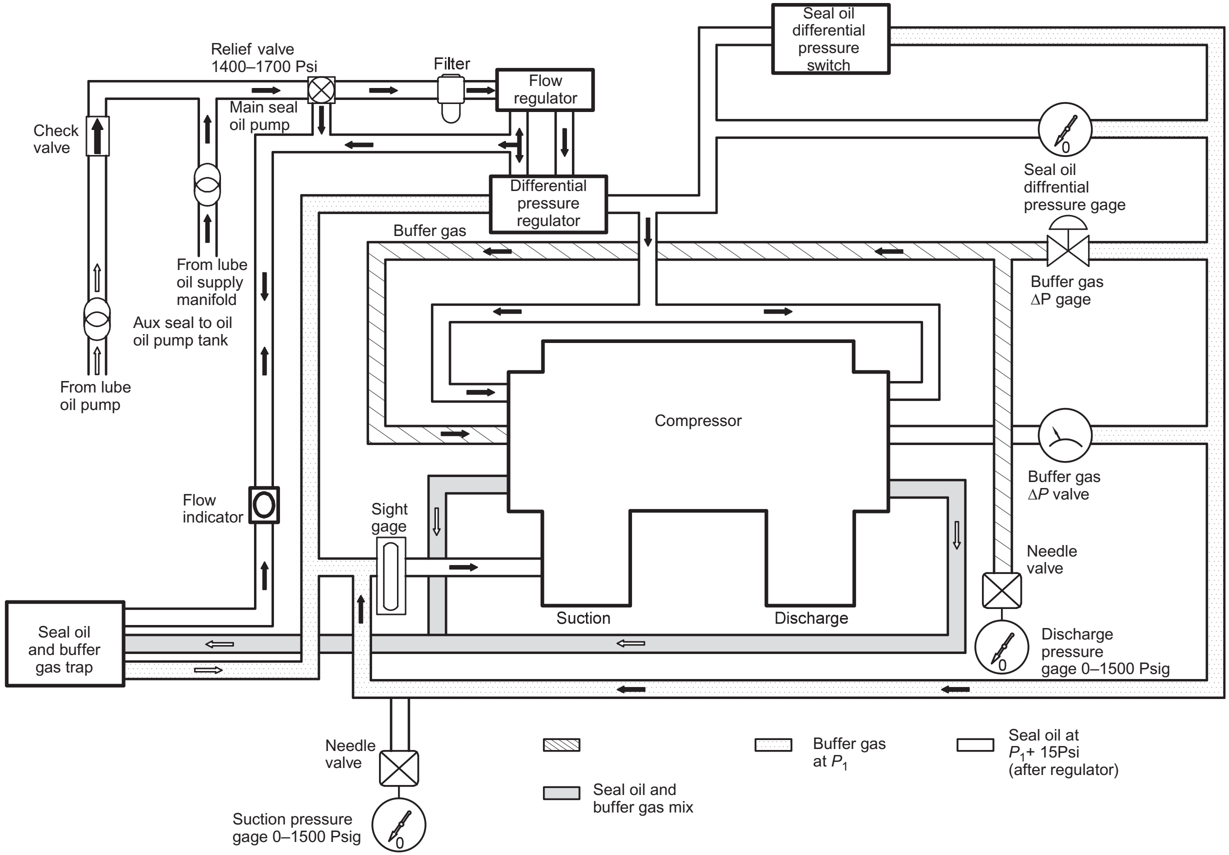


Fig. 8.46 Schematic of typical seal oil and buffer system.

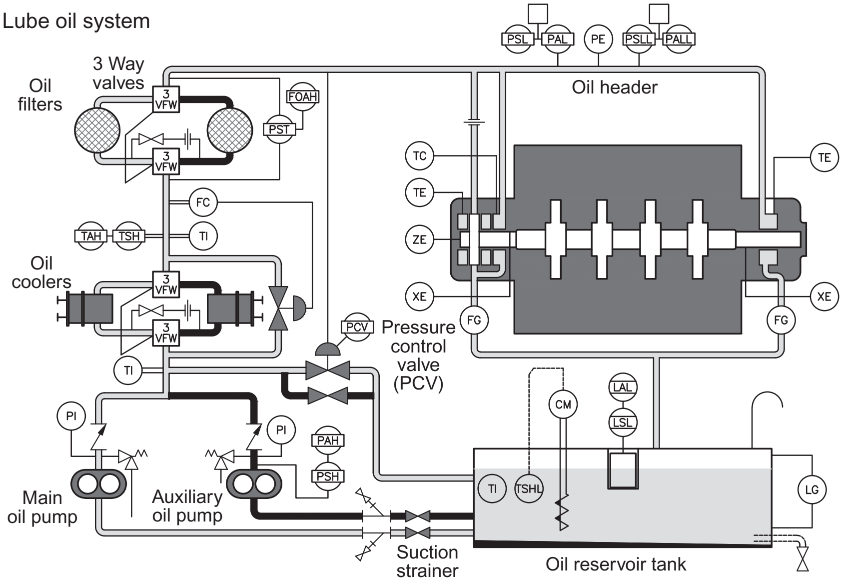


Fig. 8.47 Typical common gas turbine compressor lube oil system.

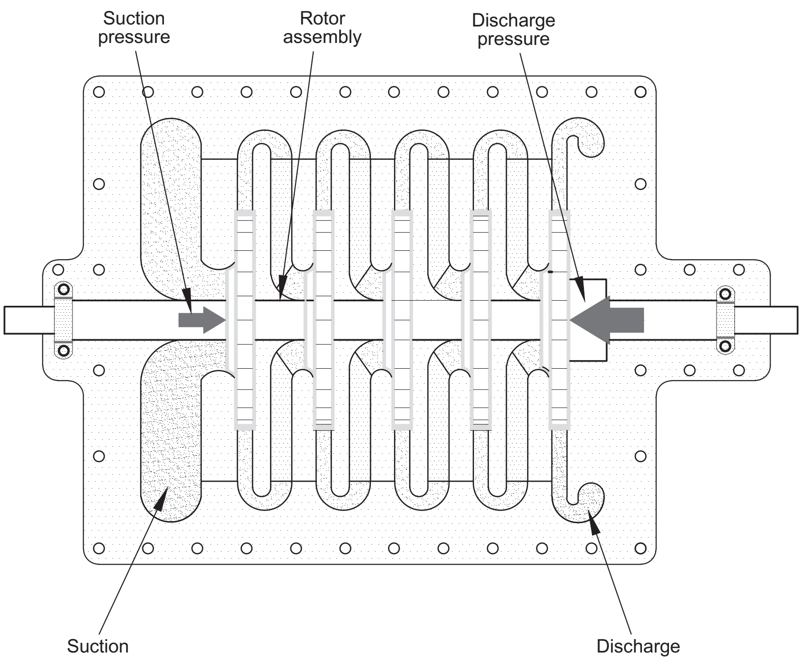


Fig. 8.48 Cutaway of a multistage centrifugal compressor illustrating the differential pressure across the rotor.

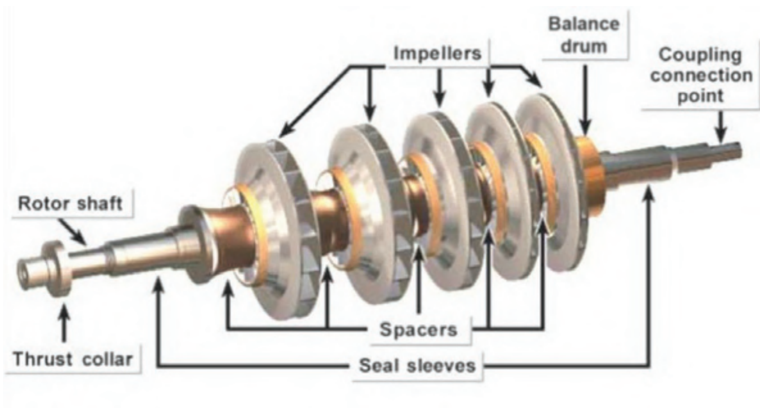


Fig. 8.49 Multistage rotor assembly showing the balancing drum on a compressor's rotor.

the nonvented side of the drum is exposed to the gas at the discharge pressure. As shown in Fig. 8.50, the differential pressure across the drum creates a net force that opposes the axial thrust of the rotor. The amount of counter-thrust produced is determined by the area of the balancing drum. The forces balance each other because they are exerted in opposite directions (refer to Fig. 8.51).

8.2.6.5 Bearing system

The bearing system often is contained in the same housing (integral) as the seal system. The rotor assembly is supported in the case on both ends by journal or radial bearings. Fig. 8.52 illustrates the three ways an unsupported shaft can move.

8.2.6.5.1 Thrust bearing system

Thrust loads on the bearing system are minimized or in some cases eliminated by the use of a balancing drum (piston) at the discharge of the rotor. Thrust bearings are used to restrict axial motion. The thrust bearing transmits the residual axial thrust of the rotor to the shaft bearing housing. As shown in Fig. 8.53 (top), a thrust bearing consists of a stationary thrust surface, thrust shoes (pads), and a revolving thrust collar that rotates with the shaft (refer to Fig. 8.53, bottom). Axial motion is prevented by thrust shoes (pads). Under normal operations, the thrust collar and thrust shoe are separated by a thin film of oil.

The tilting pad is the most common thrust bearing used in centrifugal compressors. The flat land and tapered land bearings are used less frequently. Fig. 8.54 shows a tilting-pad bearing, consisting of a thrust collar (collar disk) attached to the rotor shaft and a carrier ring which holds the pads. A button on the back of the pad allows the pad to pivot freely, thus allowing adjustment to varying oil velocity at the basic design is the self-equalizing bearing shown in Fig. 8.54. An equalizing bar design allows the bars to rock until all pads carry an equal load.

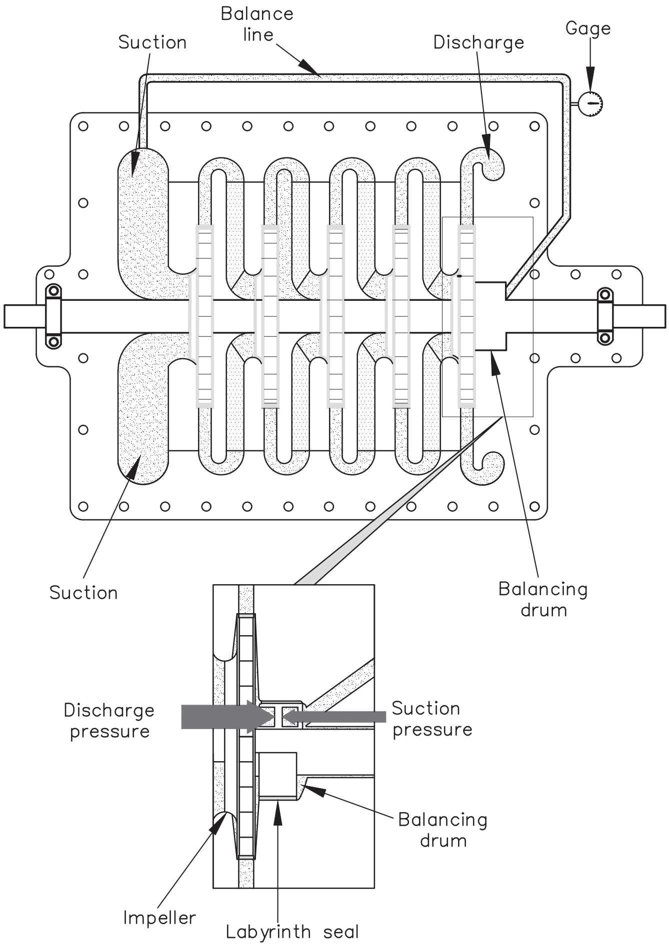


Fig. 8.50 Cutaway of a centrifugal compressor illustrating the differential pressure across the balancing drum creates a net force that opposes the axial thrust of the rotor.

8.2.6.5.2 Flat plate thrust bearing

The flat plate thrust bearing consists of a stationary flat plate and a rotating flat plate that are mounted on the shaft (Fig. 8.55). The stationary flat plate incorporates radial grooves where oil is injected. Oil flows circumferentially and radially out of the bearing.

8.2.6.5.3 Kingsbury thrust bearing

The Kingsbury bearing consists of several pads that are mounted radially and tilt circumferentially (refer to Fig. 8.56). Oil is fed into the bearing near the center where it flows radially outward. When the shaft is stationary the pads lie with their surfaces

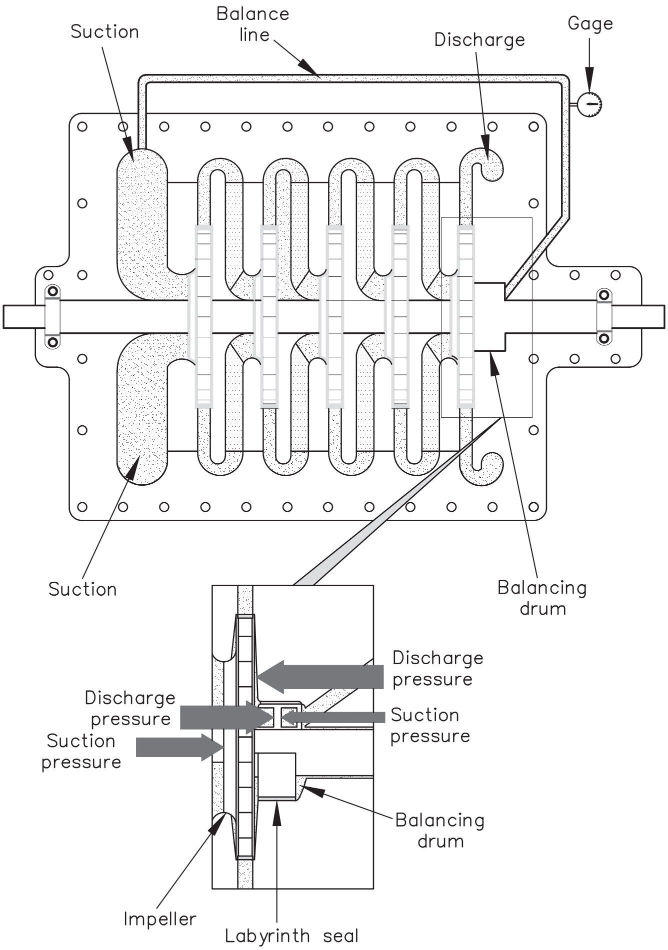


Fig. 8.51 Cutaway illustrating the forces across the balancing drum balance each other.

parallel to the shaft face. When the bearing is started an oil film is created between the pad and shaft. Each pad tilts to an angle that assures the proper distribution of film pressures within the bearing.

8.2.6.5.4 Radial bearings

Radial "journal" bearings on centrifugal compressors are usually pressure lubricated. For ease of maintenance, they are horizontally split with replaceable liners or pads. The liners or pads are usually steel backed with a thin lining of Babbitt.

Since centrifugal rotors are relatively light, bearing loads are low. This often leads to instability problems which must be compensated for by the bearing design. Due to instability, the straight-sleeve bearing is used only in some slow-speed units with

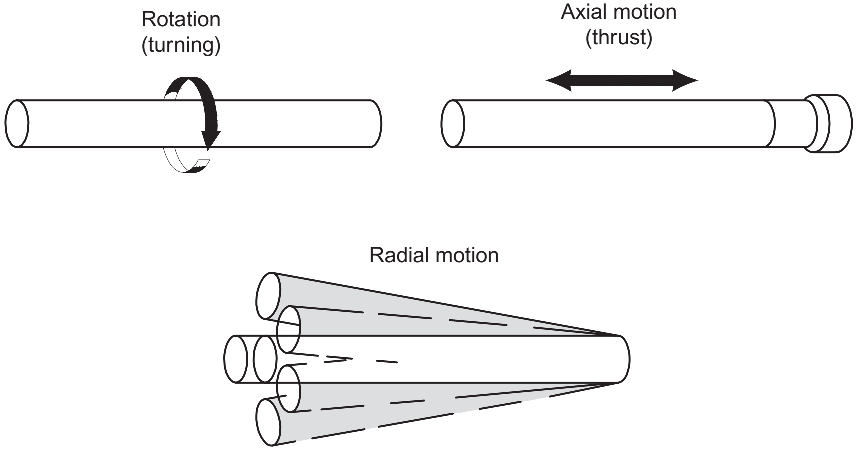


Fig. 8.52 Three ways an unsupported shaft can move.

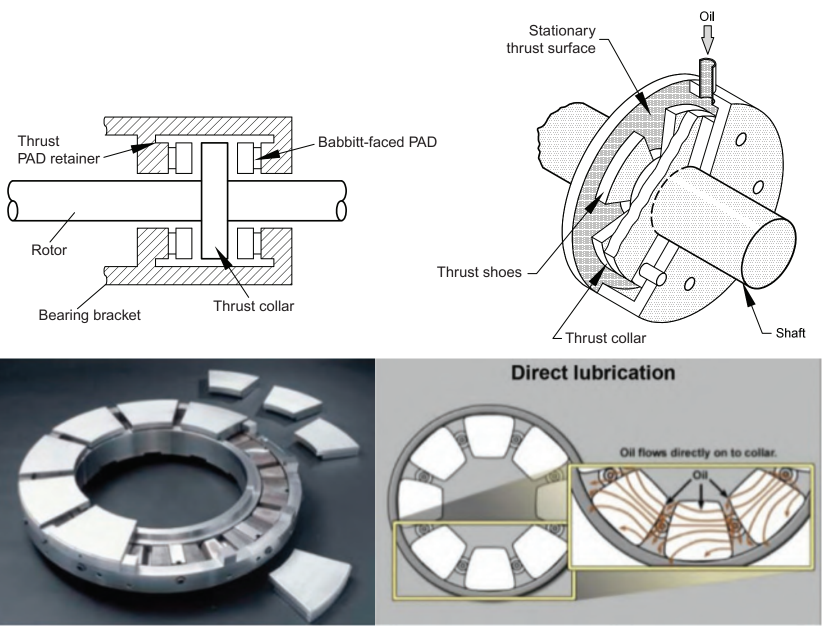


Fig. 8.53 *Top left*: Schematic illustrating the components of a thrust bearing; *Top right*: Three-dimensional cutaway of a thrust bearing; *Bottom left*: Thrust bearing; and *Bottom right*: Direct lubrication of thrust bearing.

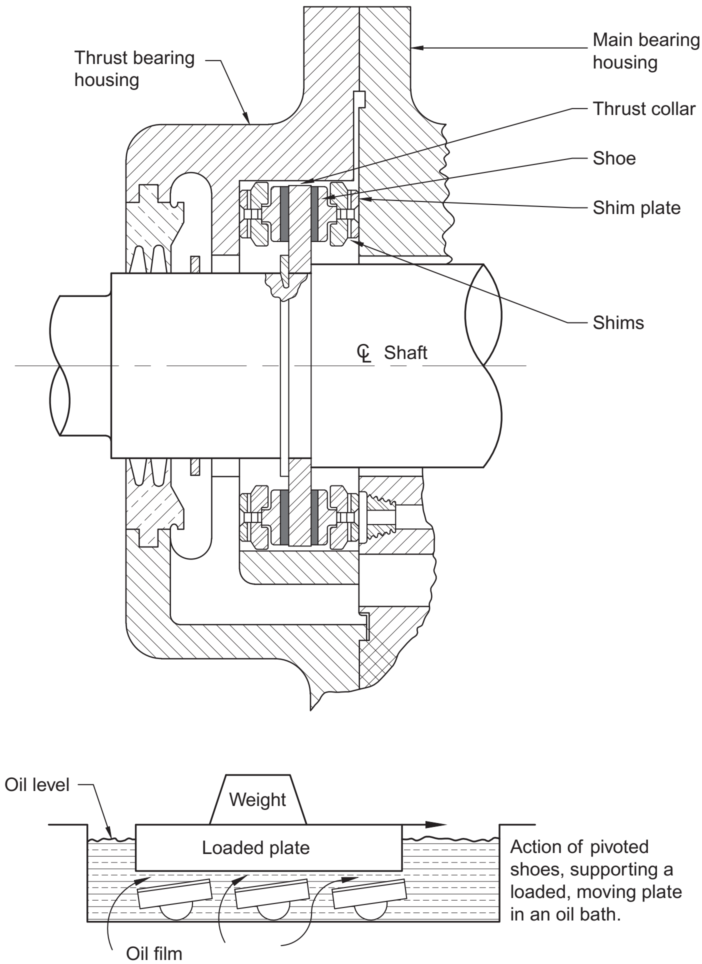


Fig. 8.54 Button-type tilting-pad thrust bearing.
Courtesy of the Elliott Company.

relatively short bearing sans. The pressure-dam sleeve bearing and the tilting-pad bearing are two commonly used designs which improve rotor stability.

The top half of the pressure-dam design is relieved as shown in Fig. 8.57, creating a pressure point where the dam ends. This conversion of oil velocity into pressure adds to rotor stability by increasing the bearing load.

The tilting-pad bearing shown in Fig. 8.58 is usually made up of five individual pads, each pivoted at its midpoint. By adjustments to the shape of the pads and bearing clearance, bearing stiffness and damping characteristics can be controlled. This bearing is successful in applications where the pressure-dam design is inadequate.

Fig. 8.59 shows a schematic of an oil circulation system.

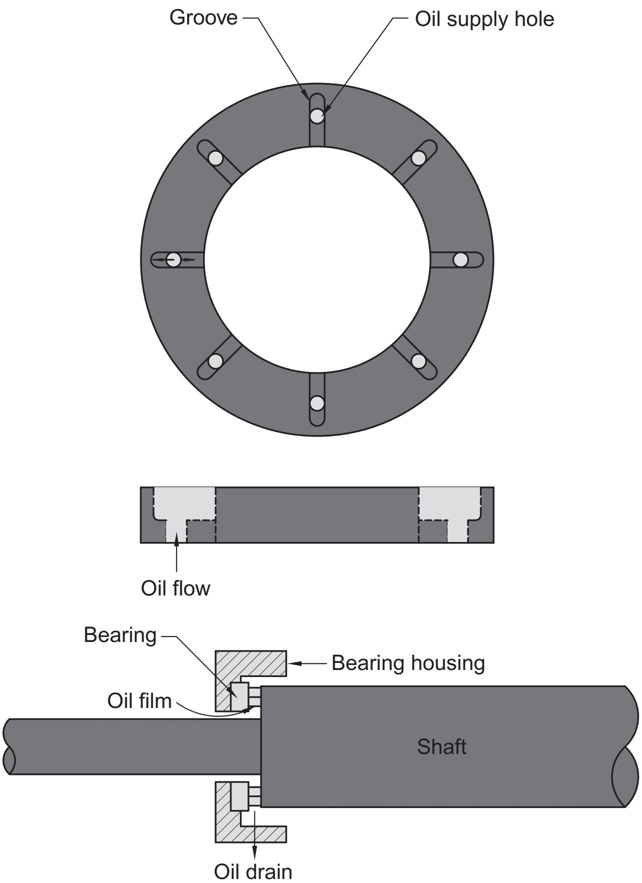


Fig. 8.55 Flat plate thrust bearing.

8.2.7 Configurations

Configuration refers to the relationship between the inlet, discharge, and sidestreams to the mechanical arrangement of the compressor. This section will be clarified by illustrative examples. Common configurations include:

8.2.7.1 Straight-through

Fig. 8.60 shows a typical cross-section of a "straight-through" multistage centrifugal compressor because flow goes in one end and out the other.

8.2.7.2 Out and in

Fig. 8.61 shows an "out and in" or sometimes called "compound" configuration. This arrangement allows removal of the total gas stream for intercooling, power savings or gas processing, and reentry for additional compression, which is commonly used in

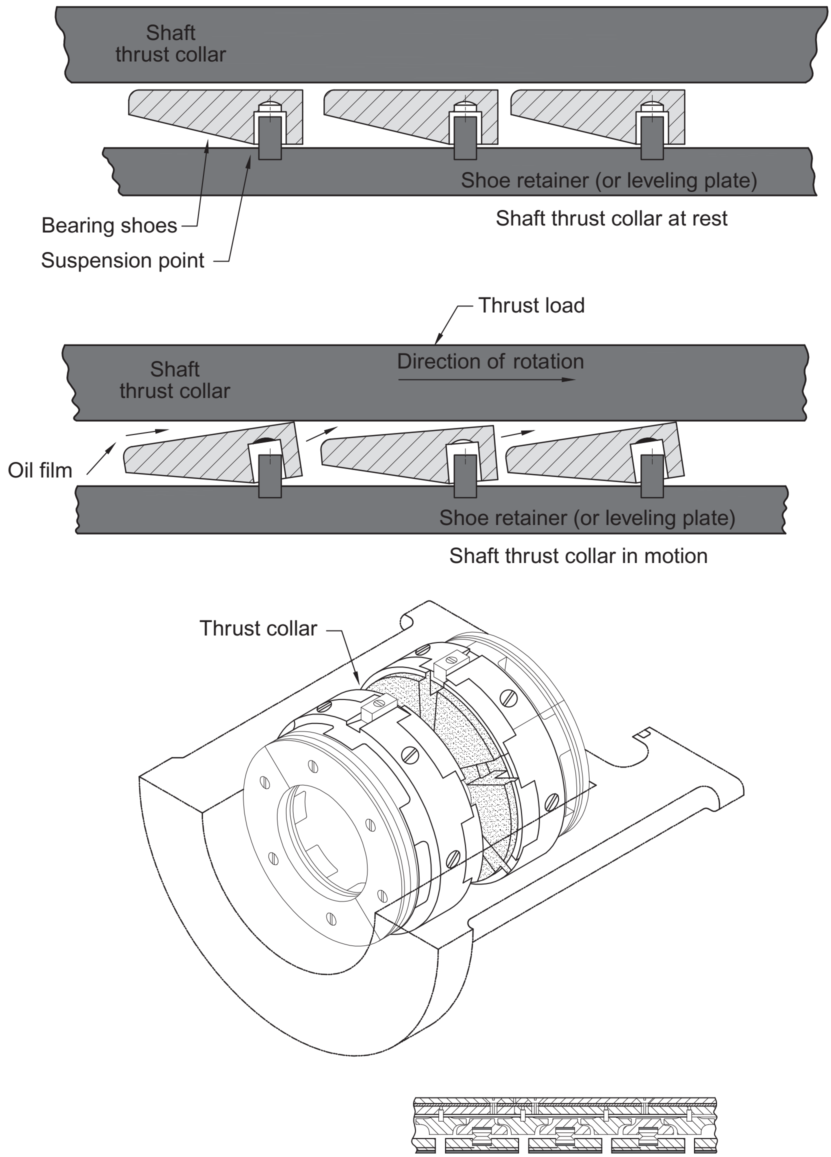


Fig. 8.56 *Top*: Kingsbury thrust bearing; *Bottom*: Self-equalizing tilting-pad thrust bearing. Courtesy of the Elliott Company.

multistage separation. Note the additional spacing required for flow extraction and reentry. Although some designs can minimize the effect, this reduces the maximum number of impellers available for compression.

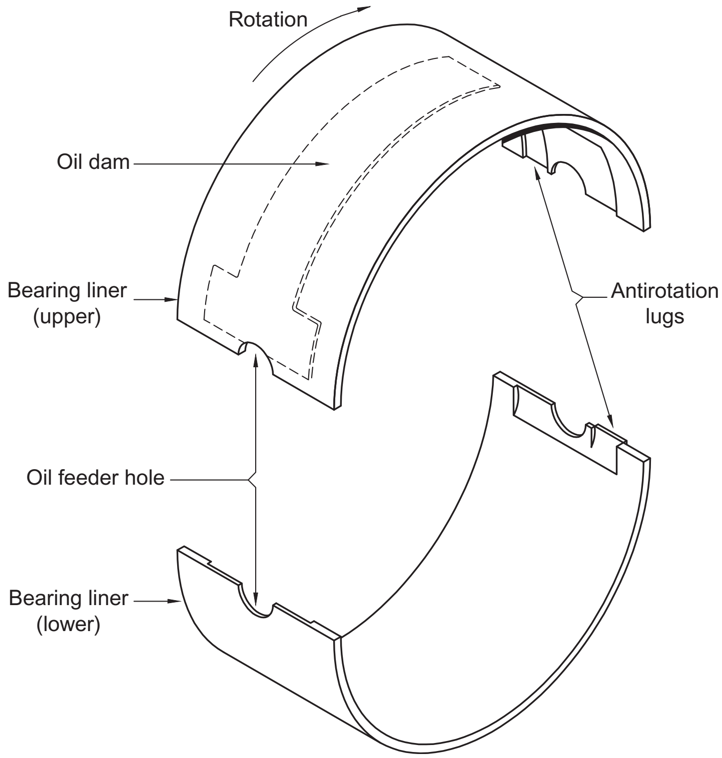


Fig. 8.57 Pressure-dam sleeve bearing liner.
Courtesy of the Elliott Company.

8.2.7.3 Sidestream compressor

Fig. 8.62 shows a "sidestream" compressor that allows the introduction or extraction of partial flows at intermediate levels to satisfy various process requirements. The number of sidestreams in a single casing is limited only by available spacing. This arrangement adds the complexity or requiring mixed temperature calculations to determine impeller performance downstream of sidestream inlets.

8.2.7.4 Double flow

Fig. 8.63 shows a "double-flow" configuration which effectively doubles the capacity of a given frame size. The compressor is divided into two sections, the inlet flow entering at either end and discharging through a common discharge nozzle at the center of the casing. The impellers in each section face in opposite directions, achieving thrust balance at all operating conditions. While flow is doubled, the number of stages available for increasing head is cut in half. The use of the double-flow option should be carefully evaluated against other alternatives.

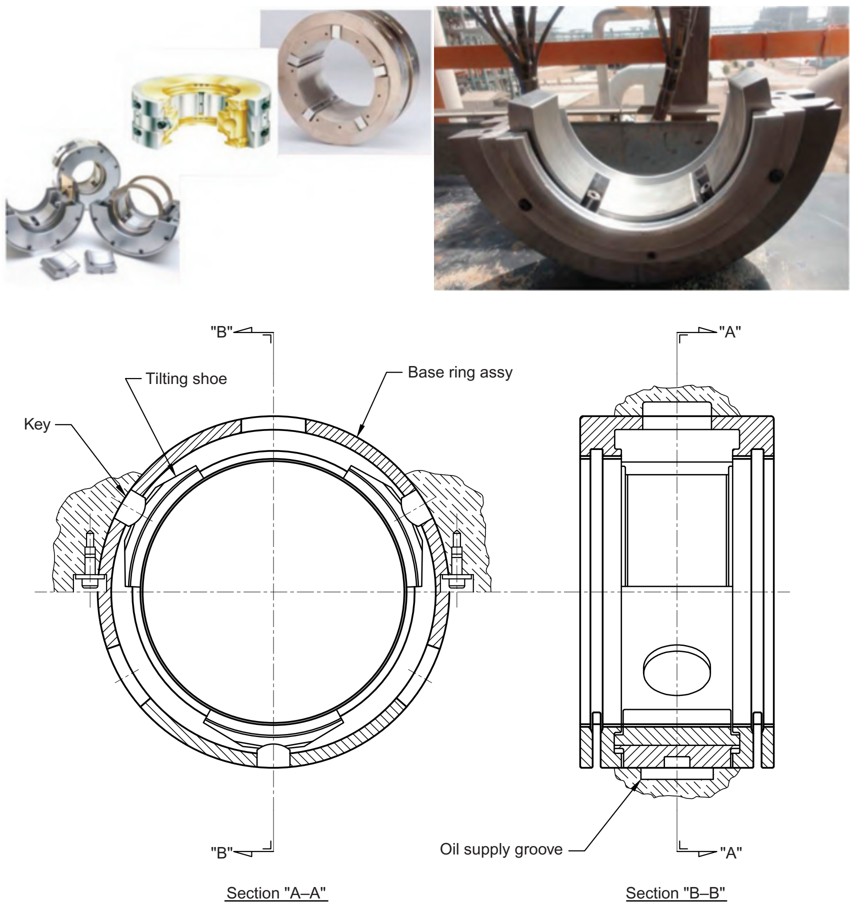


Fig. 8.58 *Top left*: Radial bearings; *Top right*: Section of a radial bearing; *Bottom*: Schematic of a tilting-pad or pivoted shoe radial journal bearing.
Courtesy of the Elliott Company.

8.2.7.5 Back to back

Fig. 8.64 shows the "back-to-back" impeller arrangement. This type has advantages in high pressure-rise applications where thrust balancing becomes difficult using a conventional thrust bearing and balancing drum. Since the back-to-back impellers produce opposing thrust forces, the net thrust is significantly reduced, eliminating the need for a balance drum (piston) to provide thrust compensation. This arrangement must, however, be carefully reviewed with respect to division wall flow disturbances, being span and seal design on rotor stability.

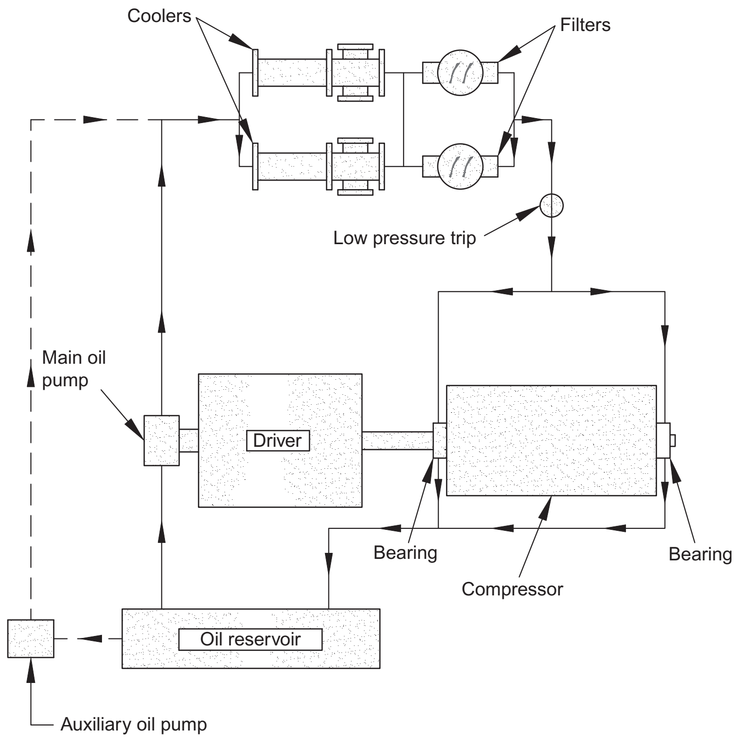


Fig. 8.59 Schematic of an oil lubrication system.

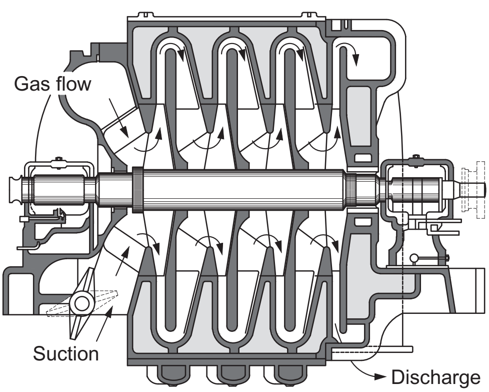


Fig. 8.60 "Straight-through" centrifugal compressor.
Courtesy of the Elliot Company.

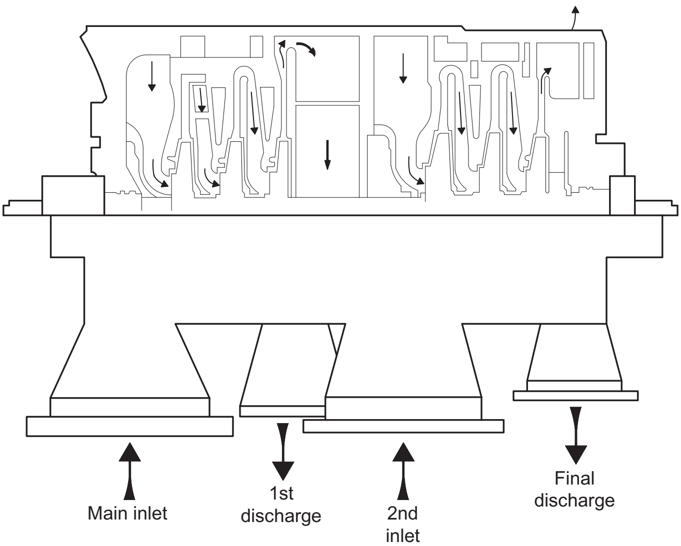


Fig. 8.61 Compound or "out-and-in" centrifugal compressor.
Courtesy of Dresser-Rand.

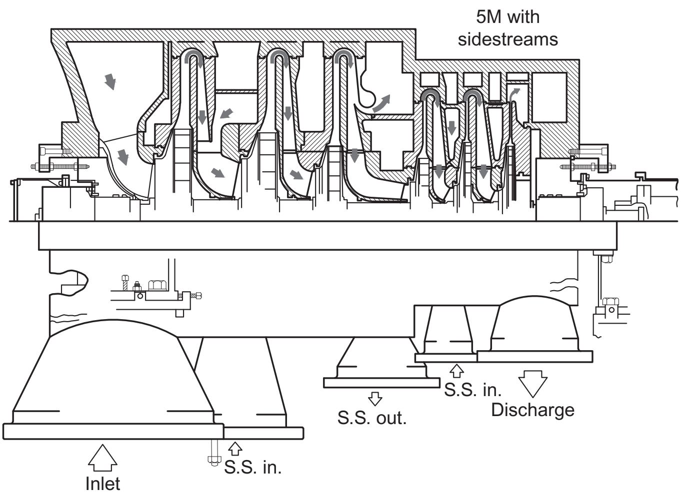


Fig. 8.62 Centrifugal compressor with "sidestream" connections.
Courtesy of Dresser-Rand.

8.2.7.6 Series/parallel

Fig. 8.65 shows a combination of "series/parallel" unit. It is sometimes used in booster-compression service. It offers good performance and flexibility switching back and forth in order to obtain higher flows or discharge pressure, as needed for system operation.

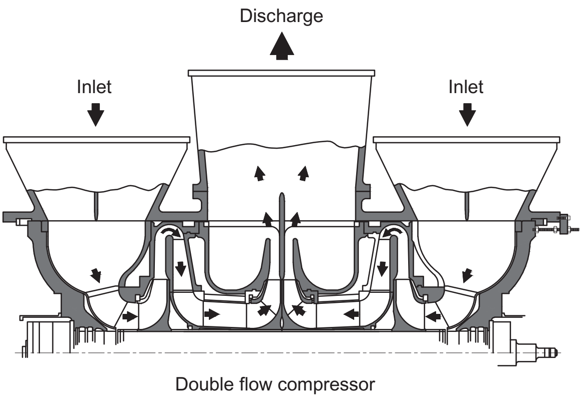


Fig. 8.63 "Double-flow" centrifugal compressor. Courtesy of Dresser-Rand.

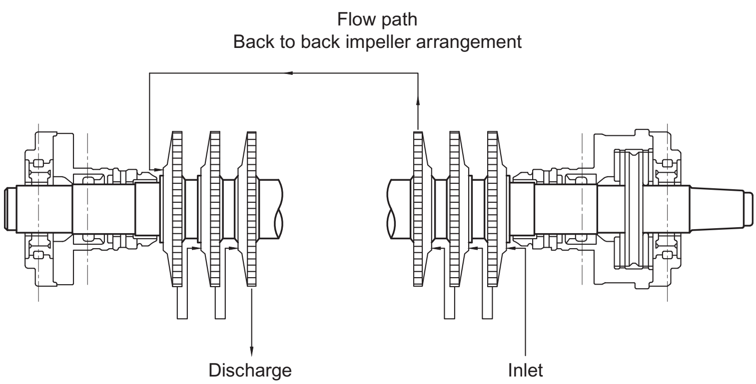


Fig. 8.64 "Back-to-back" impeller arrangement. Courtesy of Dresser-Rand.

8.3 Compressor performance curves

8.3.1 General considerations

Fig. 8.66 presents a centrifugal compressor characteristic or "performance" map (curve), using API 617 nomenclature. The family of curves shows the compressors' performance at various speeds where "N" represents rpm, and:

- *Vertical axis—Head*: polytropic head, pressure ratio, discharge pressure, or differential pressure; and
- *Horizontal axis—Inlet capacity*: called "Q" or "Q_I" shown as actual inlet volume per unit of time ACFM or ICFM where "A" is actual or "I" is inlet.

The inlet flow volume, or capacity, is based on a gas with a particular molecular weight, specific heat ratio, and compressibility factor at suction pressure and temperature.

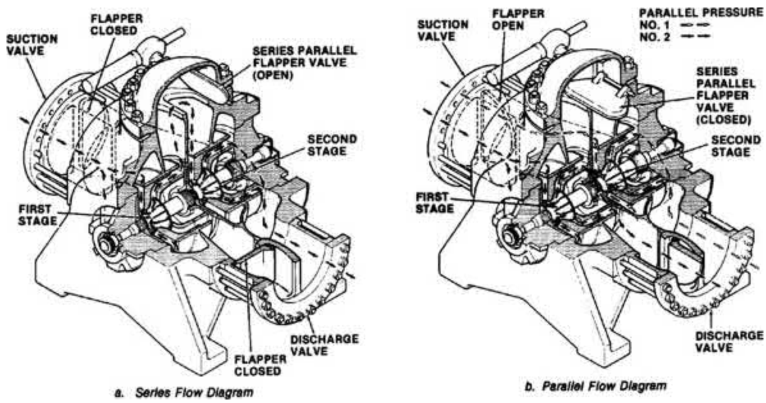


Fig. 8.65 "Series/parallel" compressor. (A) Series flow diagram. (B) Parallel flow diagram.

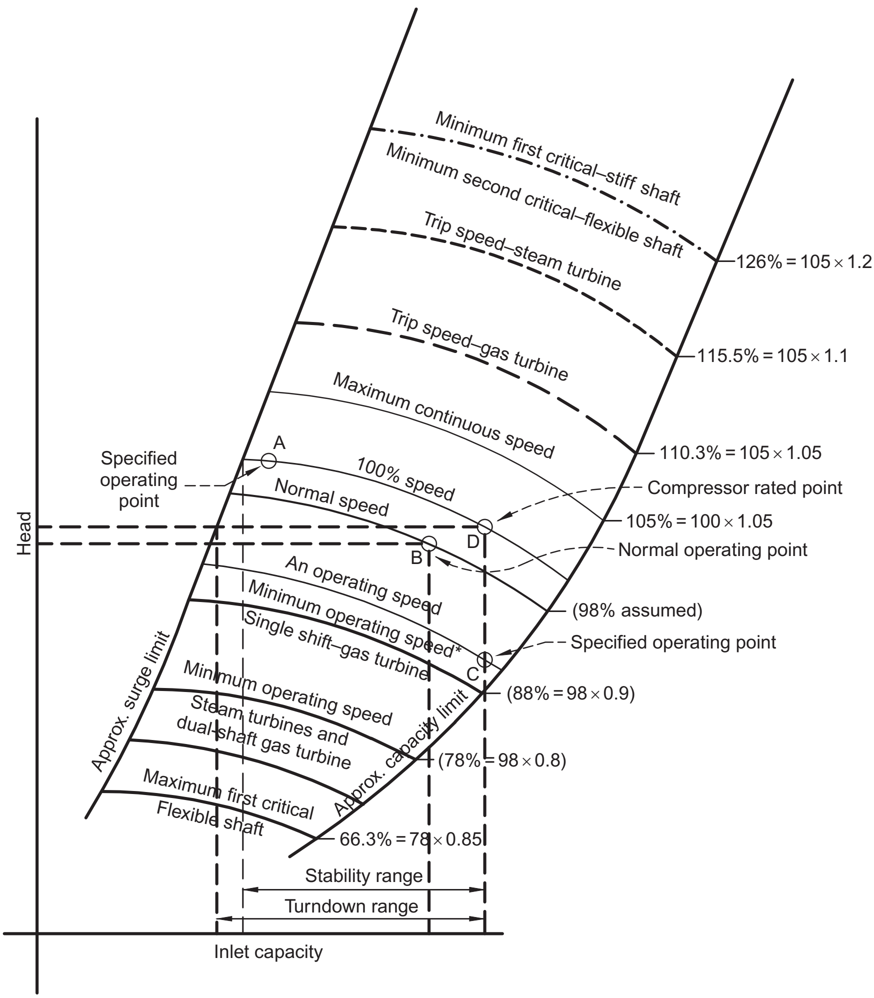


Fig. 8.66 Typical centrifugal compressor performance map.
Courtesy of the American Petroleum Institute.

The curve on the left represents the "*surge*" limit. Operation to the left of this line is unstable and usually harmful to the compressor. The "*stonewall*" or capacity limit or overload curve is shown on the other side of the map. The area to the right of this line is commonly known as "choke." Operation in this area is, in most instances, harmless mechanically, but the head-producing capability of the compressor falls off rapidly, and performance is unpredictable.

It is important one not to confuse surge and stonewall. Although compressor performance is seriously impaired in either case, they are entirely different phenomena. These are covered in more detail later in this section.

Terms frequently used to define performance are "stability range" and "percent stability." Referring again to Fig. 8.66, the rated stability range is taken as $Q_D - Q_S$ where Q_D is the rated point along the 100% speed line. The percent stability expressed as percentage is:

$$\%\text{stability} = \frac{Q_D - Q_S}{Q_D} \times 100 \tag{8.5}$$

8.3.2 Impeller performance curves

Manufacturers usually base the performance of individual impellers on an air test. Fig. 8.67 represents a typical curve which characterizes a certain impeller design. The vertical axis is usually called the head coefficient, "μ," and the horizontal axis is called the flow coefficient, "θ." (refer to Section 8.1.3 for definitions of "μ" and "θ.") In this way, impeller performance data are concisely cataloged and stored for use by designers. When a compressor is originally sized, the designer translates the impeller or "wheel" curve data into ACFM, discharge pressure, and rpm in wheel-by-wheel calculations to select a set of wheels that satisfy the purchaser's requirements.

Theoretically, an impeller should produce the same head, or feet of the fluid, regardless of the gas weight. However, in practice, an impeller will produce somewhat more head, than theoretical, with heavy gases, and less with lighter gases. Gas compressibility, specific heat ratio, aerodynamic losses, and several other factors are responsible for this deviation. Manufacturers should apply proprietary correction factors when the effect is significant. This effect contributes to variance from the well-known "fan laws" or affinity laws covered later in this chapter.

As illustrated in Fig. 8.67, the heavier gas causes surge at a higher Q/N, that is, it reduces stability. The opposite is true of a lighter gas. Similar nonconformance can sometimes be observed when the impeller is run at tip speeds considerably higher or lower than an average design speed. The higher tip speed would surge at higher Q/N, and the lower tip speed would surge at lower Q/N.

Fig. 8.68 illustrates the effects of using movable inlet guide vanes. One will notice as the head or discharge pressure is reduced, the surge volume, shown by the dashed line, is also reduced. The effect is similar to that of speed reduction on a variable speed machine. Inlet throttling, although less efficient, will produce similar curves.

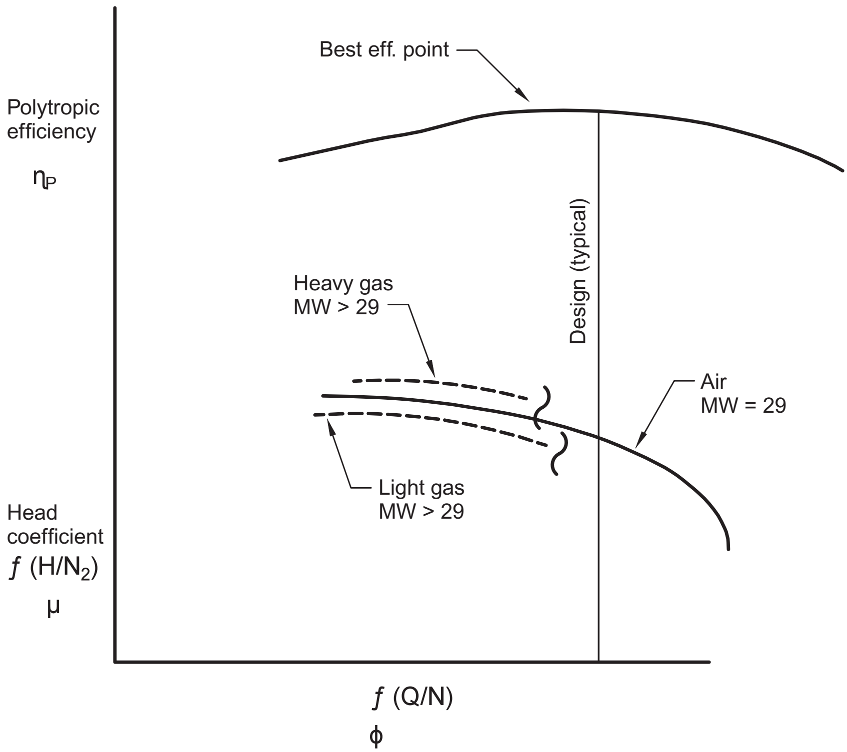


Fig. 8.67 Individual impeller performance curve.

Centrifugal compressors recognize actual inlet cubic feet per minute (ACFM or ICFM) at inlet conditions. Performance curves are most commonly plotted using ACFM. This means that a curve is drawn for a specific set of suction conditions, and any change in these conditions will affect the validity of the curve.

In many cases, performance curves plot discharge pressure on the vertical axis and flow (ACFM) on the horizontal axis. To estimate performance for varying suction pressures, the curves should be converted to pressure ratio on the vertical axis. This can be done by dividing the discharge pressures on the vertical axis by the suction pressure on which the original curve was based. The effect of a small variation in suction temperature can be estimated by using a ratio of absolute temperatures with the original temperature in the denominator. This ratio is used to correct the inlet capacity on the "X-axis" by multiplying inlet capacities by the temperature ratio.

For a rough estimate for molecular weight changes of less than 10%, the pressure ratio on the curve can simply be multiplied by the ratio of the new molecular weight over the original. Unless there are gross changes in the gas composition causing large changes in specific heat ratio, this estimating method will only have an error of 1%–2% for pressure ratios between 1.5 and 3. For more accurate estimates, a curve with polytropic head on the vertical axis must be obtained.

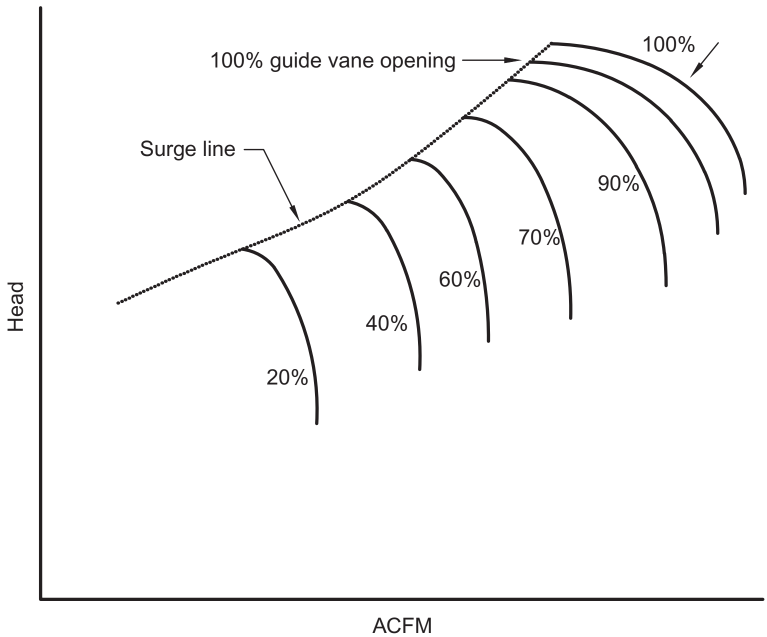


Fig. 8.68 Constant speed compressor with variable inlet guide vanes.

One must remember that any change that increases the density of the gas at the inlet will increase the discharge pressure and the horsepower. In addition, the unit will tend to surge at a slightly higher inlet volume.

8.3.3 Affinity "fan" laws

The affinity or "fan" laws can be used in many cases to estimate performance for small changes in speed and flow, but care and judgment must be used. Using these laws is risky and should be done cautiously.

The fan laws state that inlet volume is proportional to speed and that head is proportional to the speed squared. These laws are based on the assumption that the fluid is noncompressible. Fan laws may be inaccurate when testing the performance level of multistage compressors at "off-design" speeds. Fig. 8.69 illustrates this error. Similar errors could be incurred in estimating surge volumes using the fan laws.

As an example, let us assume a 10% mass flow reduction to the first stage. If all other inlet conditions remain the same, volume flow will also be reduced by 10%. Since mass flow was reduced by 10%, the second stage will also see a 10% flow reduction. (Fig. 8.66 shows that flow reduction results in an increased discharge pressure from the first stage.) Since volume is inversely proportional to pressure, the volume to the second stage will be reduced further in proportion to the increased discharge pressure from the first stage. The second stage will have a similar effect on the third

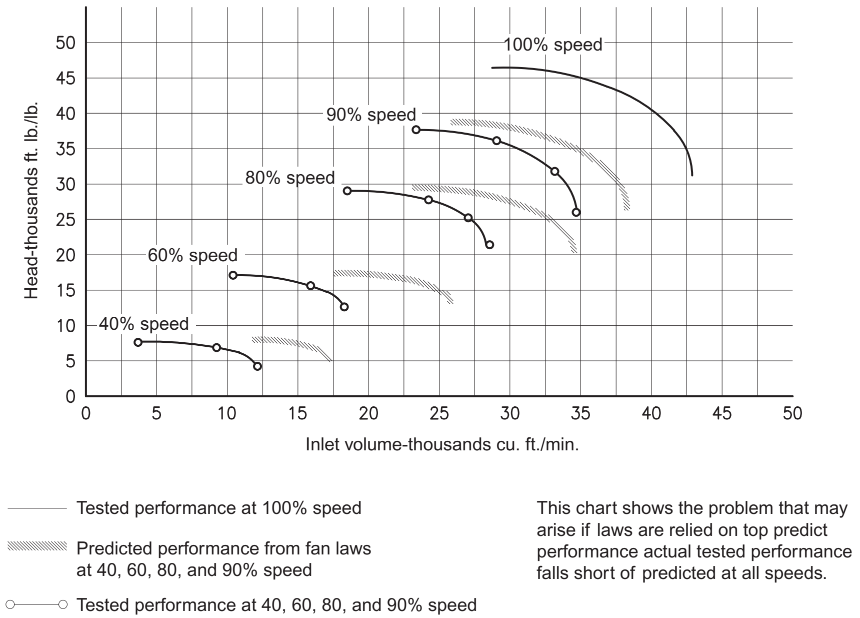


Fig. 8.69 Errors in fan laws.

stage, and so on. Deviation from the ideal gas laws will increase significantly as the number of compressor stages increases.

Performance for speeds other than the rated speed can be estimated from the following equations:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \tag{8.6}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \tag{8.7}$$

$$\frac{HP_1}{HP_2} = \left(\frac{N_1}{N_2}\right)^3 \tag{8.8}$$

where:
Q = Flow rate
H = Head
N = Impeller speed
HP = Horsepower
1,2 = Initial and final conditions

When speed differs from the compressor design speed, the accuracy of these laws declines. Accuracy of the fan laws is reduced as the number of stages increases. Generally, the fan laws will provide acceptable accuracy up to +/− 10% of the rated speed.

8.3.4 Surge

8.3.4.1 Description

Surge is an operating point at which unstable operation occurs that can destroy a compressor. Surge can occur very rapidly, 5–6 times per second (approximately 200 ms). Surge can be very damaging because it reverses the bending stress on many components and usually causes the rotor to shift back and forth rapidly. It is a critical factor in the design of a compressor and its control system. It is also a critical operating limit. As shown in Fig. 8.70, it is a low-flow limit of a compressor at a particular operating speed. To the left of the surge limit point, a compressor's head-making capability decreases as flow decreases. Every centrifugal compressor has an operating point of minimum flow and corresponding maximum head, and, if exceeded, will no longer perform in a stable manner. The limits are exceeded when the backpressure on the system cannot be overcome by the head produced. When this happens, a back-and-forth flow motion (surging) occurs inside the compressor that can be very damaging to the rotating components. At surge, continuous "forward" flow is interrupted. For this reason, operating a compressor at its surge point should always be avoided.

8.3.4.2 Flow through the impeller and diffuser

During normal flow conditions a uniform film, or "boundary layer," of gas is formed on both sides of the impeller vane passage (refer to Fig. 8.71). When the boundary layer is continuous, the entire diameter of the impeller is effectively used for producing head. As the flow is reduced or pressure ratio across the impeller increases, this boundary layer breaks down, or "stalls," at the impeller tip causing recirculation.

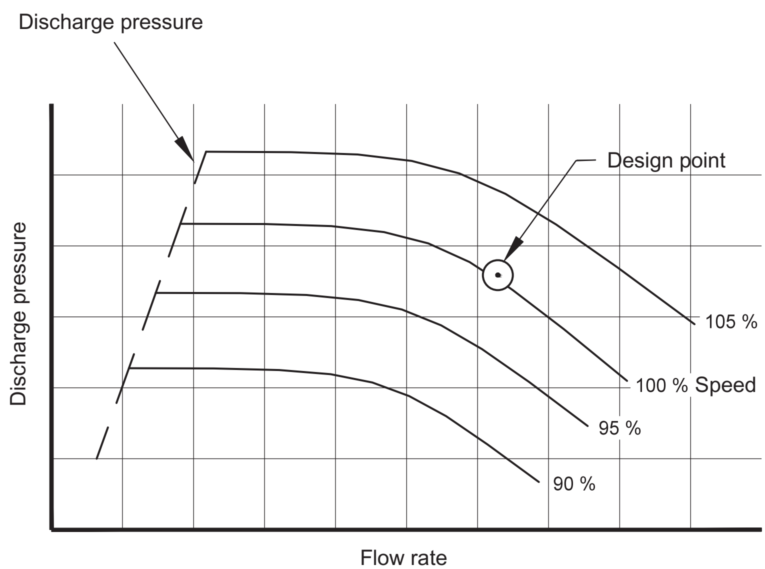


Fig. 8.70 Typical centrifugal compressor curves showing surge.

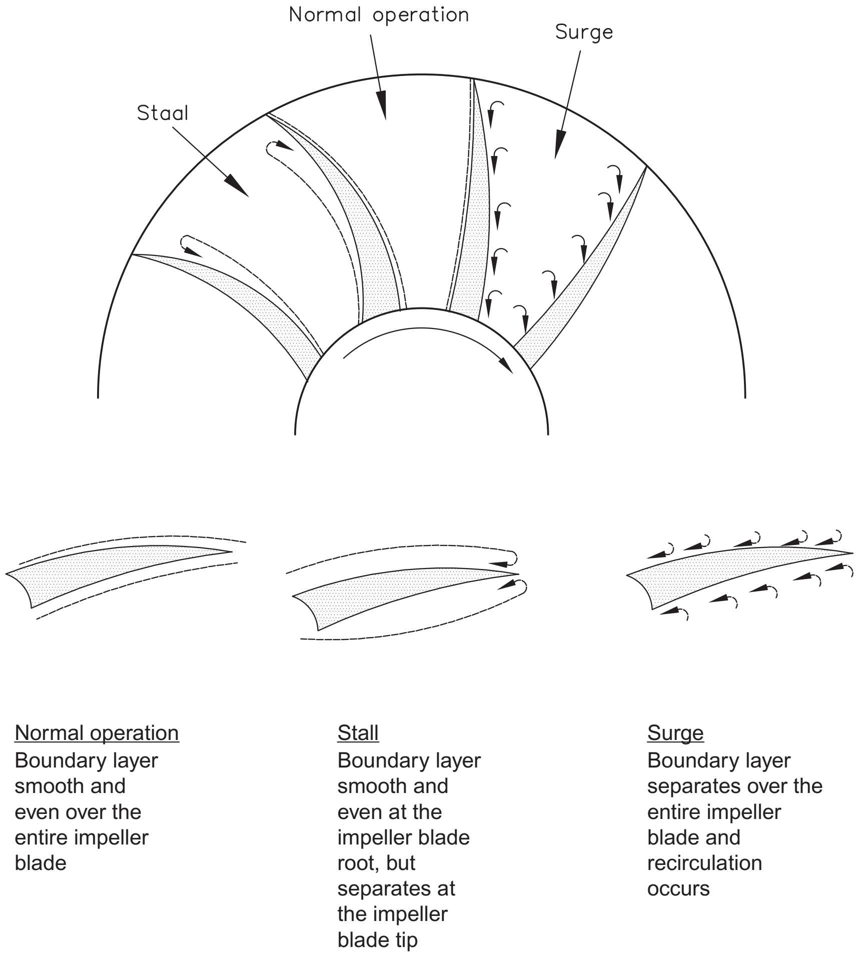


Fig. 8.71 Impeller boundary layer control.

Recirculation reduces the impeller surface area used to accelerate the gas, and the energy of the gas cannot overcome the pressure in the diffuser. The resulting backflow occurs until the diffuser pressure is equalized with the gas leaving the impeller. The boundary layer then reforms, and the pressure builds until it is again too high causing the boundary layer to breakdown. The rapid breakdown and subsequent reforming of the boundary layer characterizes violent surge.

While surge is caused by aerodynamic instability in the compressor, interaction with the system sometimes produces violent swings in flow, accompanied by pressure fluctuations and relatively rapid temperature increase at the compressor inlet. Surge affects the overall system and is not confined to only the compressor. Therefore an understanding of both the external causes and the machine design is necessary to apply an adequate antisurge system.

The compressor surge region was previously identified in Fig. 8.66. In Fig. 8.72 lines depicting three typical system operating curves have been added. The shapes of these curves are governed by the system friction and pressure control in the particular system external to the compressor.

A compressor will operate at the intersection of its curve and the system curve. To change the point at which the compressor operates

- Change the speed or variable geometry of the compressor, thus relocating the compressor curve, or
- Change the system curve by repositioning a control valve or otherwise altering the external system curve.

8.3.4.3 Typical surge cycle

A typical surge cycle is represented by the circuit between points B, C, D and back to B (Fig. 8.72). If events take place that alter the system curve to establish operation at point B, the pressure in the system will equal the output pressure of the compressor. Any transient can then cause reverse flow if the compressor discharge pressure falls below the downstream system pressure.

For reverse flow to occur, compressor throughput must be reduced to zero at point C that corresponds to a pressure called the "shutoff head." When the system pressure has decreased to the compressor's shutoff head at C, the compressor will reestablish forward flow since the flow requirement of the compressor is satisfied by the backflow gas (compressor capability now greater than system requirements).

Now that the compressor has sufficient gas to compress, operation will immediately shift to the right in approximately a horizontal path to point D. With the compressor now delivering flow in the forward direction, pressure will build in the system, and operation will follow the characteristic speed curve back to points B and C. The cycle will rapidly repeat itself unless the cause of the surge is corrected, or other favorable action taken, such as increasing the speed.

Several internal factors combine to develop the surge condition. From the surge description, one can see the domed shape of the head-capacity characteristic curve is fundamentally responsible for the location of the surge point at a given speed. On the right side of the performance map (Fig. 8.72) the slope of the curve is negative. As inlet flow is reduced, the slope becomes less negative until it reaches zero at the surge point. As flow is reduced further to the left of the surge point, the slope becomes increasingly positive.

8.3.4.4 Surge frequency

The surge frequency cycle varies inversely with the volume of the system. For example, if the piping contains a check valve located near the compressor discharge nozzle, the frequency will be correspondingly much higher than that of the system without a check valve. The frequency can be as low as a few cycles per minute up to 15 or more cycles per second. Generally, the higher the frequency, the lower the intensity.

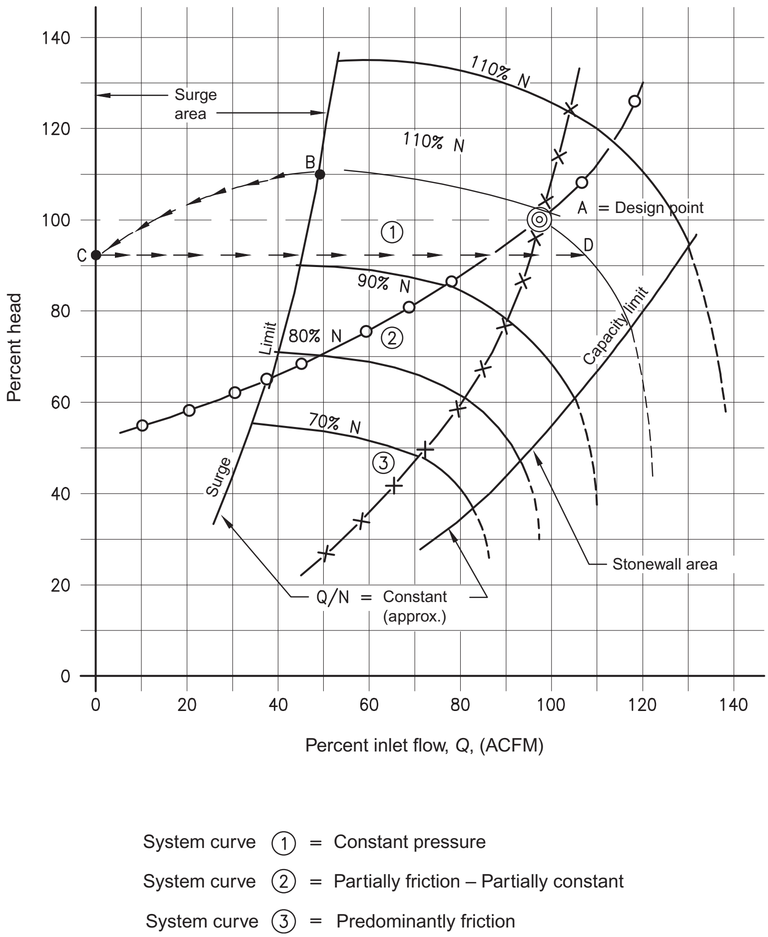


Fig. 8.72 Typical centrifugal compressor performance map showing surge cycle.

The intensity or violence of surge tends to increase with increased gas density which is directly related to higher molecular weights and pressures, and lower temperatures. Higher differential pressure generally increases the intensity.

8.3.4.5 Design factors affecting surge

A greater number of impellers in a given casing will tend to reduce the stable range. Similarly, so does the number of sections of compression, or the number of casings in series.

The large majority of centrifugal compressors use vaneless diffusers, which are simple flow channels with parallel walls, without elements inside to guide the flow. The trajectory of a particle through a vaneless diffuser is a spiral of about one half the

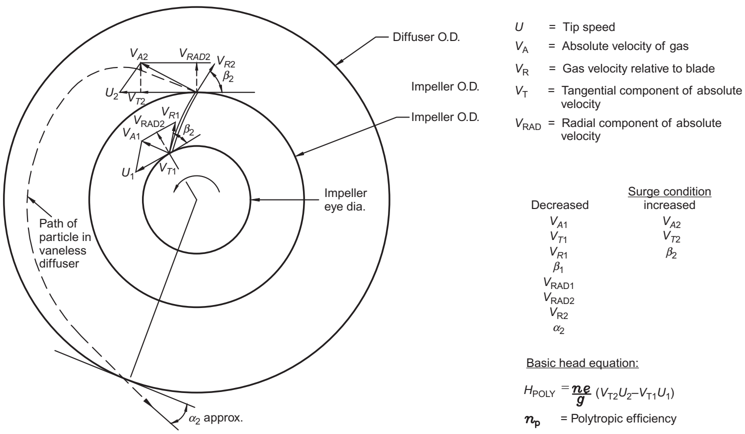


Fig. 8.73 Design condition triangles. The trajectory of a particle through a vaneless diffuser is a spiral of approximately one half the circumference distance around the diffuser.
Courtesy of the Turbo-machinery Laboratory, from Proceedings of the 12th Turbo-machinery Symposium, Texas A&M University, College Station, TX, 1983.

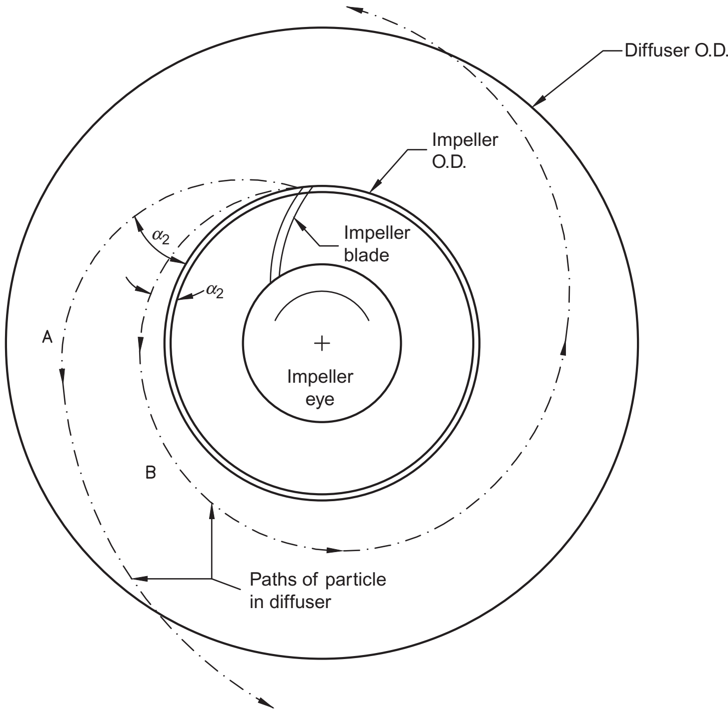
circumferential distance around the diffuser (Fig. 8.73). If this distance becomes longer for any reason, the flow is exposed to more wall friction which dissipates the kinetic energy. As flow is reduced, the angle is reduced which extends the length of the trajectory through the diffuser (Fig. 8.74). When the flow path is too long, insufficient pressure rise (head) is developed and surge occurs.

Occasionally, vane diffusers are used to force the flow to take a shorter, more efficient path. Fig. 8.75 shows the flow pattern in a vane diffuser. The vane diffuser can increase the aerodynamic efficiency of a stage by approximately 3% but this efficiency gain results in a narrower operating span on the head-capacity curve with respect to both surge and stonewall. The figure also shows how the path of a particle of gas is affected by off-design flows. At flows higher than design, impingement occurs on the trailing side of the diffuser vane creating shock losses which tend to bring on stonewall. Conversely, flow less than the design encourages surge, due to the shock losses from impingement on the leading edge of the vane.

Despite adverse effects on surge, the vane diffuser should be applied where efficiency is of utmost importance, particularly with small high-speed wheels.

Stationary guide vanes may be used to direct the flow to the eye of the impeller. Depending upon the head requirements of an individual stage, these vanes may direct the flow in the same direction as the rotation or tip speed of the impeller, an action known as prerotation or preswirl. The opposite action is known as counter-rotation or counter swirl. Guide vanes set at zero degrees are called radial guide vanes.

The effect guide vanes have on a compressor's curve is illustrated in Fig. 8.76. Note that prerotation reduces the head or unloads the impeller. Prerotation tends to reduce the surge flow. Counter-rotation increases the head and tends to increase the surge flow.



A – Normal condition, 900D flow angle. Relatively short flow path. Not much friction loss.

B – Tendency toward surge. Shallow flow angle. Long flow path. Large friction loss. Possibility that part of the flow field will re-enter the impeller.

Fig. 8.74 Flow trajectory in a vaneless diffuser. As flow is reduced, the angle is reduced which extends the length of the trajectory through the diffuser.
Courtesy of the Turbo-machinery Laboratory, from Proceedings of the 12th Turbo-machinery Symposium, Texas A&M University, College Station, TX, 1983.

Movable inlet guide vanes are occasionally used on single-stage compressors, or on the first stage of multistage compressors driven by electric motors at constant speed. The guide vane angle can be manually or automatically adjusted while the unit is on stream to accommodate operating requirements Because of the complexity of the adjusting mechanism, the variable feature can only be applied to the first wheel in almost all designs.

8.3.4.6 External symptoms and effects of surge

During the flow reversals of surge, loud clattering noises will be heard from the compressor. Surge can often be recognized by check valve hammering, piping vibration, noise, and wiggling of pressure gages. Mild cases are sometimes difficult to discern.

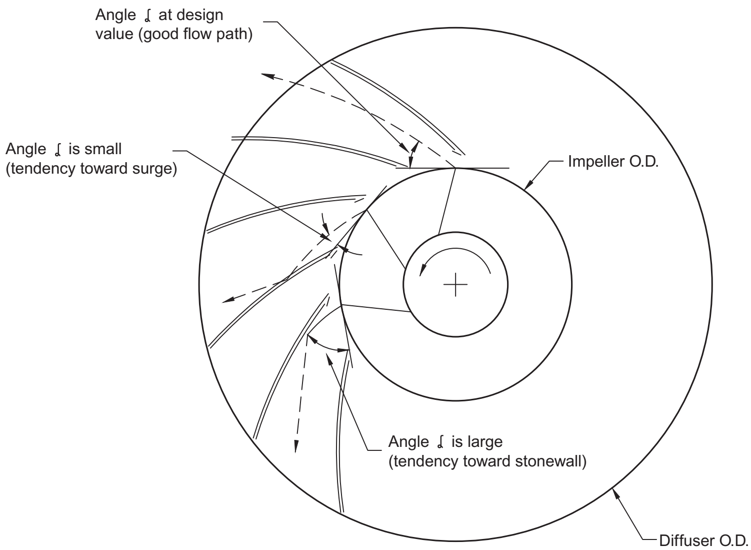


Fig. 8.75 Flow pattern in a vaned diffuser.

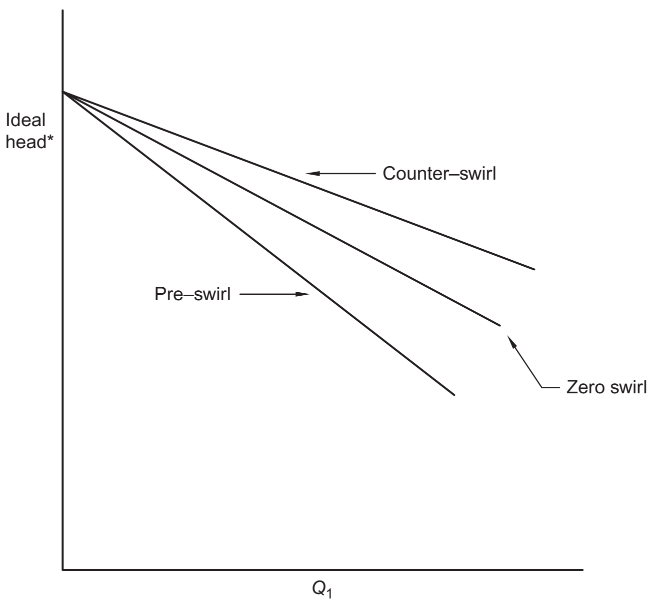


Fig. 8.76 Effect of guide vane setting (stationary or variable). (* For impeller with backward leaning blades.)

This occurs at frequent, constant intervals. If allowed to continue operating, the compressor will sustain severe internal damage.

Mechanical damage can be caused by either high discharge temperatures that result from recycling hot gas and/or large thrust reversals of the compressor rotor. High discharge temperatures can cause seal and bearing failures. Thrust reversals can overload thrust bearings resulting in catastrophic failures.

The effects of surge can range from a simple lack of performance to serious damage to the compressor and/or the system. Internal damage to labyrinths, diaphragms, thrust bearing, and rotor can be experienced. Surge often excites lateral shaft vibration. It can also produce torsional damages to items such as couplings and gears. Externally, devastating piping vibrations can occur causing structural damage, misalignment, and failure of fittings and instruments.

Some of the usual causes of surge, other than from machine design, are as follows:

- Restricted suction or discharge such as a plugged strainer
- Process changes in pressure or gas composition
- Mis-positioned rotor or internal plugging of flow passages
- Inadvertent speed change such as from a governor failure

8.3.4.7 Surge control

A centrifugal compressor can be brought out of a surge condition by reducing the head and/or increasing flow. Also, depending on the existing speed and the slope of the surge line, increasing the compressor speed may bring it out of surge. The primary means of avoiding surge is by increasing flow through the compressor. The easiest and most common way to do this is to open a bypass or recycle valve which allows discharge gas to be recirculated to the suction (refer to Fig. 8.77). A surge control system composed of various components which send and receive signals from flow and pressure measuring devices operates this valve by sensing that the compressor is nearing operating in surge.

8.3.5 Stonewall

Another factor affecting the theoretical head-capacity curve is “choke” or “stonewall.” Stonewall is a point at which a centrifugal compressor is operating at maximum flow and minimum head. When this point is reached for a given gas the impellers are physically unable to handle the gas and the flow through the compressor cannot be increased without further internal modifications. This point is also known as choked flow, since it is a point at which the highest stable flow occurs (refer to Fig. 8.78).

Stonewalling causes a noticeable drop in pressure and an increase in vibration due to turbulent flow within the compressor. Increased vibration could damage the compressor. Operation in this region causes excessive use of horsepower, occasionally to the point of overload and frequent flaring.

The terms surge and stonewall are sometimes incorrectly used interchangeability, probably due to the fact that serious performance deterioration is observed in either case.

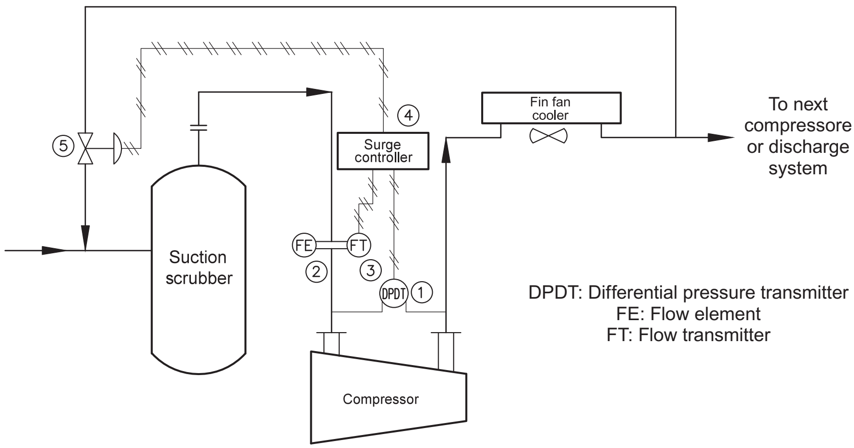


Fig. 8.77 Typical surge control system (items 1 through 5 are typical components of a surge control system).

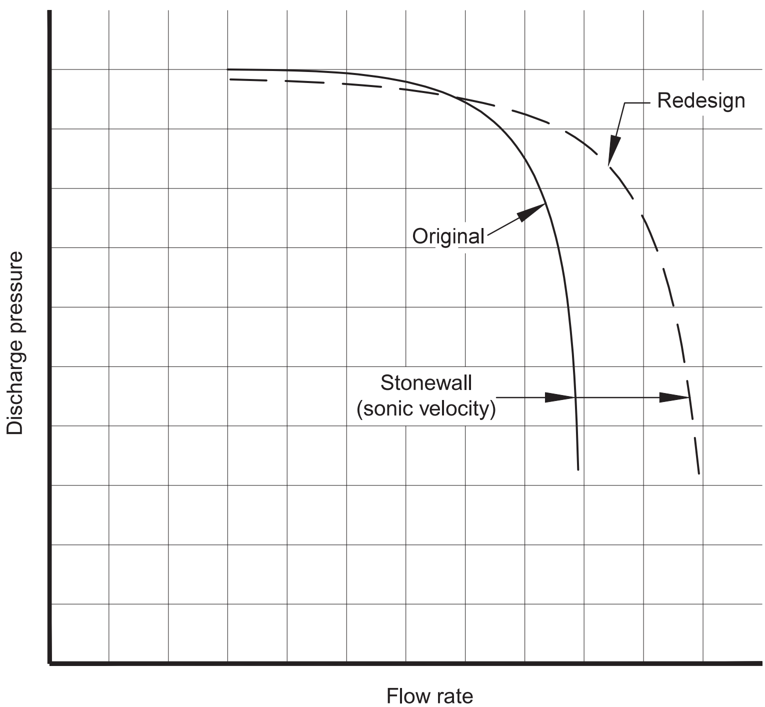


Fig. 8.78 Centrifugal compressor performance curves illustrating stonewall.

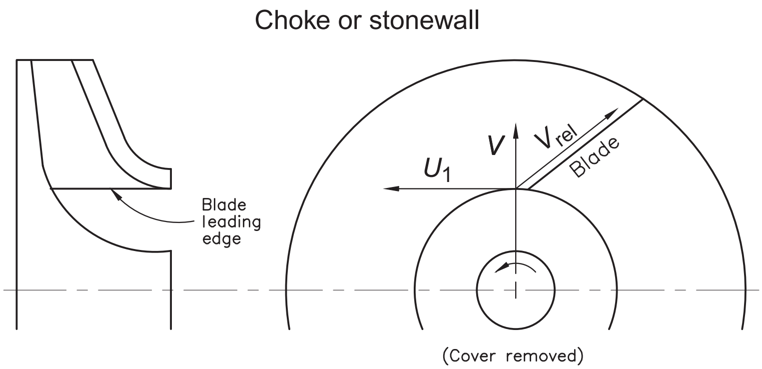


Fig. 8.79 Inlet gas velocity vector—design flow.
Courtesy of the Elliott Company.

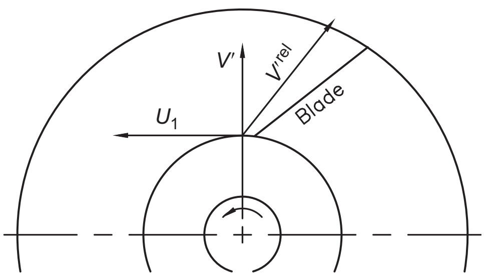


Fig. 8.80 Inlet gas velocity vector—negative incidence angle (onset of choke).
Courtesy of the Elliott Company.

In theory, a compressor stage is considered to be in stonewall when the Mach Number equals one. At this point the impeller is choked and no more flow can be passed. Industry practice normally limits the inlet Mach Number to less than 0.90 for any specified operating point.

The vector diagram shown in Fig. 8.79 shows an inlet gas velocity vector which lines up well with the impeller blade at design flow.

The ratio of the inlet gas velocity (relative to the impeller blade) to the speed of sound at inlet is referred to as the relative *inlet Mach Number*.

$$\text{Mach No.} = \frac{V_{rel}}{a_1} \tag{8.9}$$

where:

$$a_1 = \sqrt{g\ k\ ZRT_1}$$
$$= \text{speed of sound at inlet}$$

As flow continues to increase, the incidence angle of the relative gas velocity, with respect to the impeller blade, becomes negative as shown in Fig. 8.80. The negative incidence angle results in an effective reduction of the flow area and impingement of the gas on the trailing edge of the blade, contributing to flow separation and the onset of choke.

It is important to note the choke effect is much greater for high molecular weight gas, especially at low temperatures and lower "*k*" values. Thus maximum allowable compressor speed may be limited on high molecular weight applications, with a corresponding reduction in head per stage.

8.4 Centrifugal compressor selection criteria

8.4.1 Application range

Table 8.2 is a chart of capacity vs. pressure for both horizontally and vertically split centrifugal compressors. Normally, manufacturers do not design a compressor to match an application; they fit the application to one of a series of existing compressor casings or frame sizes. Therefore the designer must check the manufacturer's catalogs for data required to make selection estimates. Table 8.2 provides data for a series of compressor casings based on a comparison of data from industry.

In addition, the minimum discharge cubic feet per minute (DCFM) should be considered. Current impeller designs limit impeller inlet CFM to approximately 300–500 initial cubic feet per minute (ICFM). Thus process conditions resulting in a discharge volume of less than approximately 250 DCFM may be unacceptable.

8.4.2 Horsepower and efficiency estimates

The major benefit of a designer determining their own estimates, rather than turning everything over to a manufacturer, is the designer will develop a better understanding of the application. The designer is then in a better position to discuss it with the

Table 8.2 Preliminary selection values for multistage centrifugal compressors

Casing size (frame)	Flow range (ft^3/min)	Head coefficient (μ)[a]	Nominal impeller diameter (inches)	Typical speed (rpm)	Maximum[b] polytropic head/stage (ft)
1	500–7000	0.48	14–16	12,000	12,000
2	2000–18,000	0.49–0.50	18–22	9,000	12,000
3	8000–31,000	0.50–0.51	24–30	8,500	12,000
4	10,000–44,000	0.50–0.51	36	5,600	12,000
5	30,000–65,000	0.51–0.52	42–45	4,400	12,000
6	60,000–100,000	0.52–0.53	54	3,600	10,000
7	80,000–160,000	0.53–0.54	60	3,000	10,000

[a]Based on backward leaning blades.
[b]For 28–30 molecular weight at normal temperatures.

manufacturers, evaluate alternate selections, and even catch errors in a manufacturer's estimates.

Fig. 8.81 is a plot of polytropic efficiency vs. inlet volume flow. This chart may be used for estimating polytropic efficiencies. Manufacturers use a computer to calculate compressor performance on a stage-by-stage basis. Performance is based on each preceding stage, new impeller conditions, including compressibility (Z) and k values to determine the individual performance for each successive stage.

If specific stage data is unavailable, overall calculations using average compressibility and a k-value based on the average flange-to-flange temperature will provide reasonably accurate results. (Refer to Chapter 7 for compressibility equations.)

Estimate overall efficiency from Fig. 8.81, using average CFM from:

$$CFM_{avg} = \frac{\text{Inlet ACFM} + \text{Disch.ACFM}}{2} \tag{8.10}$$

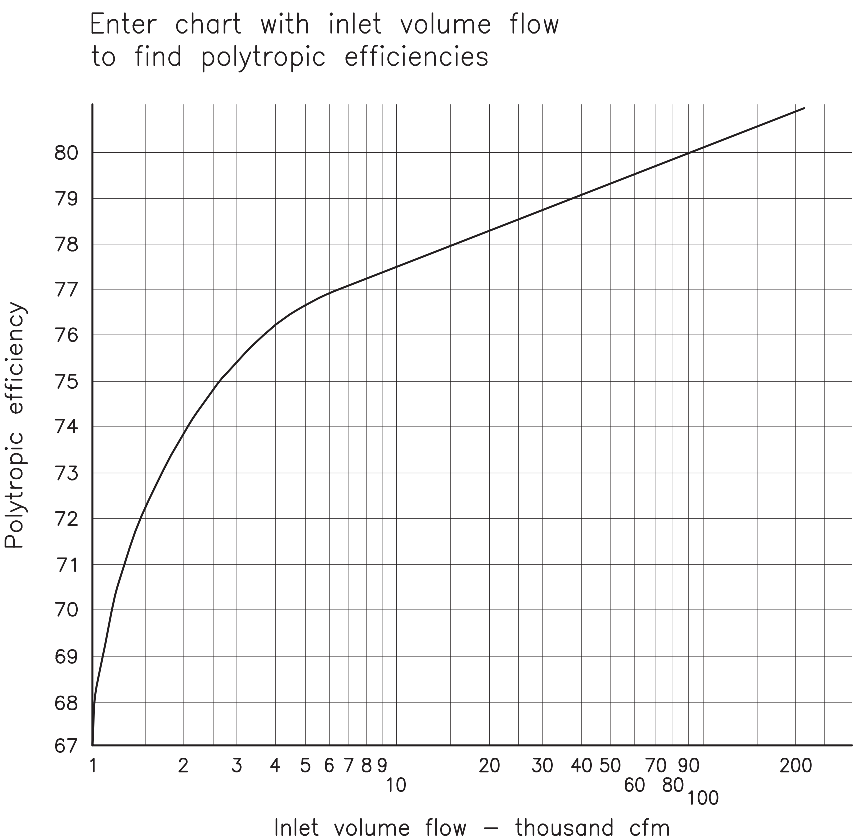


Fig. 8.81 Polytropic efficiency vs. inlet volume flow.
Courtesy of Dresser-Rand.

where discharge ACFM is determined using Eq. (7.16) and an efficiency of 75%.
Determine $n-1/n$ from:

$$\frac{n-1}{n}=\frac{k-1}{k\eta_p} \tag{8.11}$$

Recalculate head, discharge temperature, and gas horsepower (GHP) from:

$$H_p = z_{\text{avg}}RT_1\frac{r^{\frac{n-1}{n}}-1}{\frac{n-1}{n}} \tag{8.12}$$

where:
H_p = Polytropic head, foot-lbf/lbm (N-m/kg)

$$T_2 = T_1 r^{\frac{n-1}{n}} \tag{8.13}$$

Field units

$$\text{Gas power HP} = \frac{wH_p}{33{,}000\ \eta_p} \tag{8.14a}$$

SI units

$$\text{Gas power kW} = \frac{wH_p}{1000\ \eta_p} \tag{8.14b}$$

where:
w = weight flow in lbs/min (kg/sec)

Estimate shaft power using:

$$\text{Shaft power} = \text{Gas power} + \text{bearing loss} + \text{oil seal loss}$$

Where bearing loss is determined from Fig. 8.82, and oil seal loss is determined from Fig. 8.83. The casing size in the figures is selected by comparing the cfm_{avg} with the flow range in Table 8.2.

8.4.3 Head/stage

Even though special impeller designs are available for higher heads, a good estimate for a typical multistage compressor is approximately 10,000 ft/stage (3048 m/stage). This is based on an assumed impeller flow coefficient of 0.5 and a nominal impeller tip speed of 800 fps (244 mps).

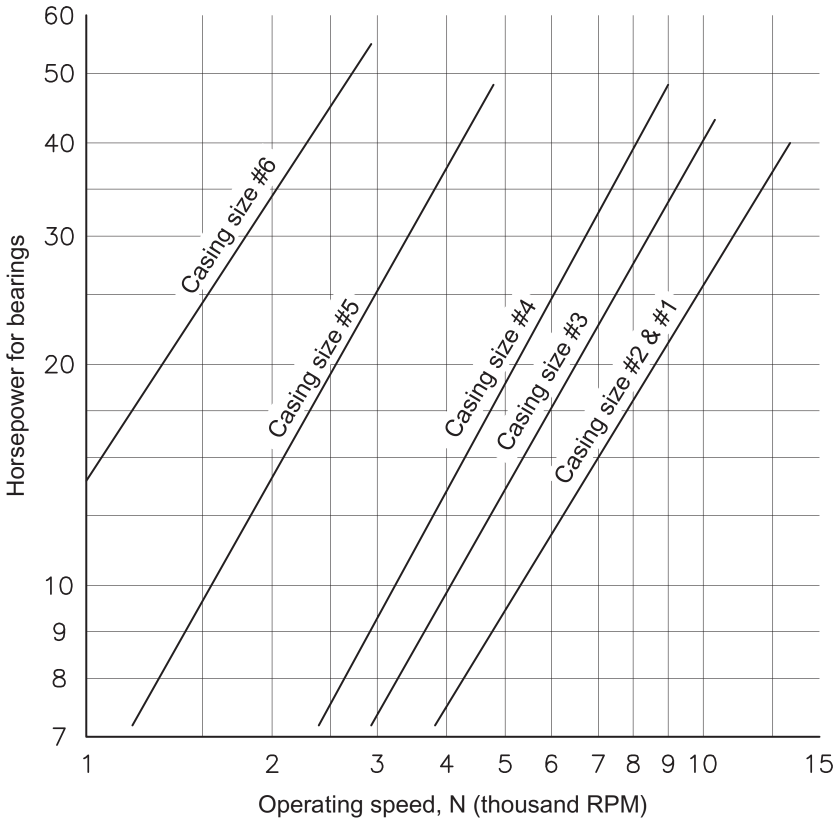


Fig. 8.82 Bearing losses vs. casing size and speed.
Courtesy of Dresser-Rand.

The actual head per stage varies between manufacturers and individual impeller designs, ranging from 9000 to 12,000 feet (2743–3658 m) for 28–30 molecular weight gas at normal temperatures.

Head per stage is limited by:

- Impeller stress levels.
- Inlet Mach number.

8.4.3.1 Impeller stress level

The following speed margins are defined by API:

- Rated (design) speed: 100%
- Maximum continuous speed: 105% of rated speed
- Trip speed: 110% of maximum continuous
- Overspeed: 115% of maximum continuous

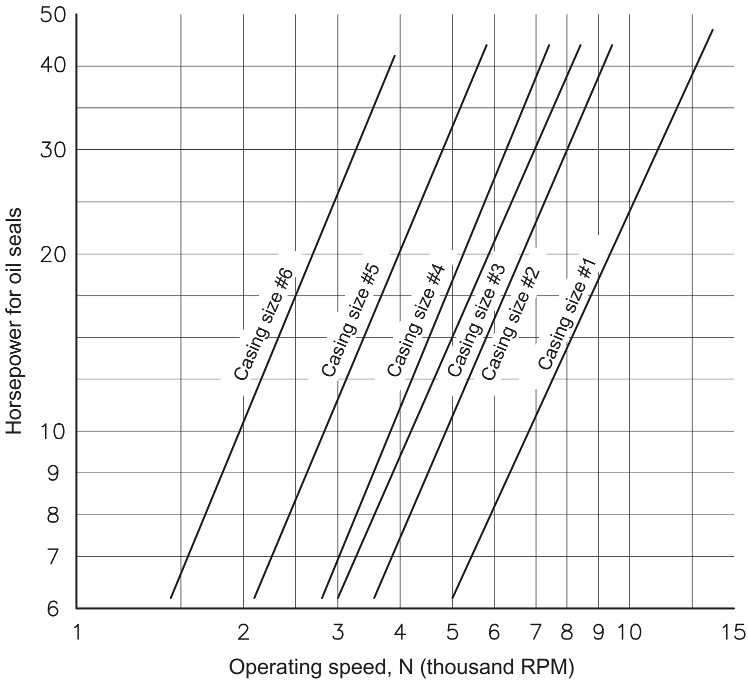


Fig. 8.83 Oil seal losses vs. casing size and speed.
Courtesy of Dresser-Rand.

Table 8.3 Impeller stress levels at various speeds

Speed	API specification	Speed %	Stress %	Stress as % of minimum specified yield strength[a]
Design	100%	100	100	69
Maximum continuous	105% of design	105	110	75
Trip	115% of design	115	130	90
Over speed	115% of max. cont.	121	145	100

[a]Based on assumption that stress at over speed testing approaches but does not equal minimum specified yield strength.

Table 8.3 identifies the impeller stresses at various rotational speeds. Reduced yield strengths required for corrosive gas will correspondingly reduce maximum head per stage through reduction in speed.

8.4.3.2 Inlet mach number

An increase in gas molecular weight, or a decrease in k, Z or inlet temperature will result in an increase in inlet Mach Number. For high molecular weight or low-temperature applications, Mach Number may limit head per stage for a given design.

8.4.4 Stages/casing

The maximum number of stages per casing should normally be limited to eight. It is usually limited by rotor critical speeds, although in a few cases temperature can be a limiting factor.

Most multistage centrifugal compressors operate between the first and second critical (flexible-shaft rotor). Fig. 8.84 shows the location of critical speeds in relation to the operating speed range. API specifies the required separation between critical speeds and the compressor operating range. As the bearing span is increased to accommodate additional impellers, the critical speed decreases, with the second critical approaching the operating range. While some manufacturer's catalogs indicate as many as 10 or more stages per casing, designs exceeding eight impellers per case should be carefully evaluated against operating experience from similar units.

For compound, or sidestream loads, additional stage spacing may be required to allow for intermediate exit and/or entry of the gas. In these applications, the number of impellers would be reduced accordingly.

8.4.5 Discharge temperature

If the calculated discharge temperature exceeds approximately 350°F (177°C), cooling should be considered to avoid problems with compressor materials, seal components, and clearances. The exact temperature limit is dependent on factors such as the gas compressed, compressor materials, allowable temperature of the seal oil, and the type of seals. Also, note that discharge temperature will increase as flow is reduced toward surge.

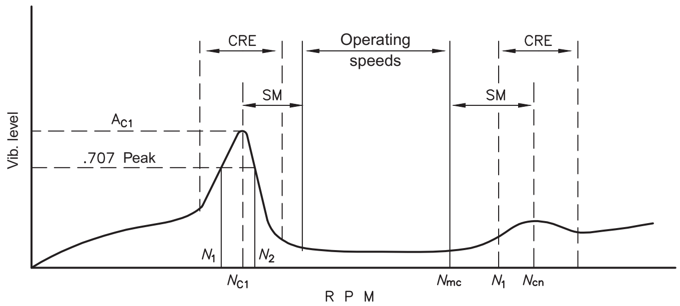
8.5 Centrifugal compressor performance maps

8.5.1 General considerations

Centrifugal compressors' performance is usually expressed in terms of a group of curves that graphically display the range of:

- Flows.
- Efficiencies.
- Heads.
- Speeds.

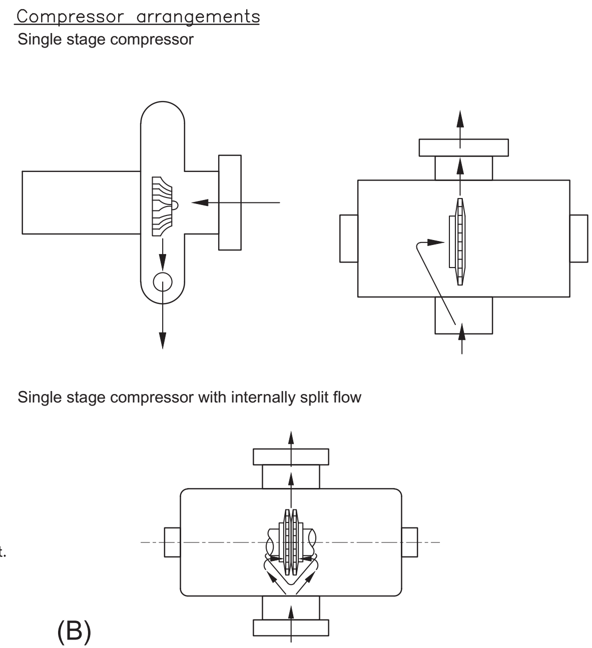
Fig. 8.85 shows the group of curves referred to as the operating envelope or performance map of a centrifugal compressor. Performance maps are a function of:



N_{c1}	=	Rotor 1st critical, center frequency. cycles per minute
N_{cn}	=	Critical speed, nth
N_t	=	Trip speed
N_{mc}	=	Maximum continuous speed. 105 percent
N_1	=	Initial (lesser) speed at .707 x peak amplitude (critical)
N_2	=	Final (greater) speed at .707 x peak amplitude (critical)
N_2-N_1	=	Peak width at the "half-power" point
AF	=	Amplification factor $= \frac{N_{c1}}{N_2-N_1}$
SM	=	Separation margin
CRE	=	Critical response envelope
A_{c1}	=	Amplitude $@N_{c1}$
A_{cn}	=	Amplitude $@N_{cn}$

Note : Curve shape is for illustration only and does not necessarily represent any actual rotor response plot.

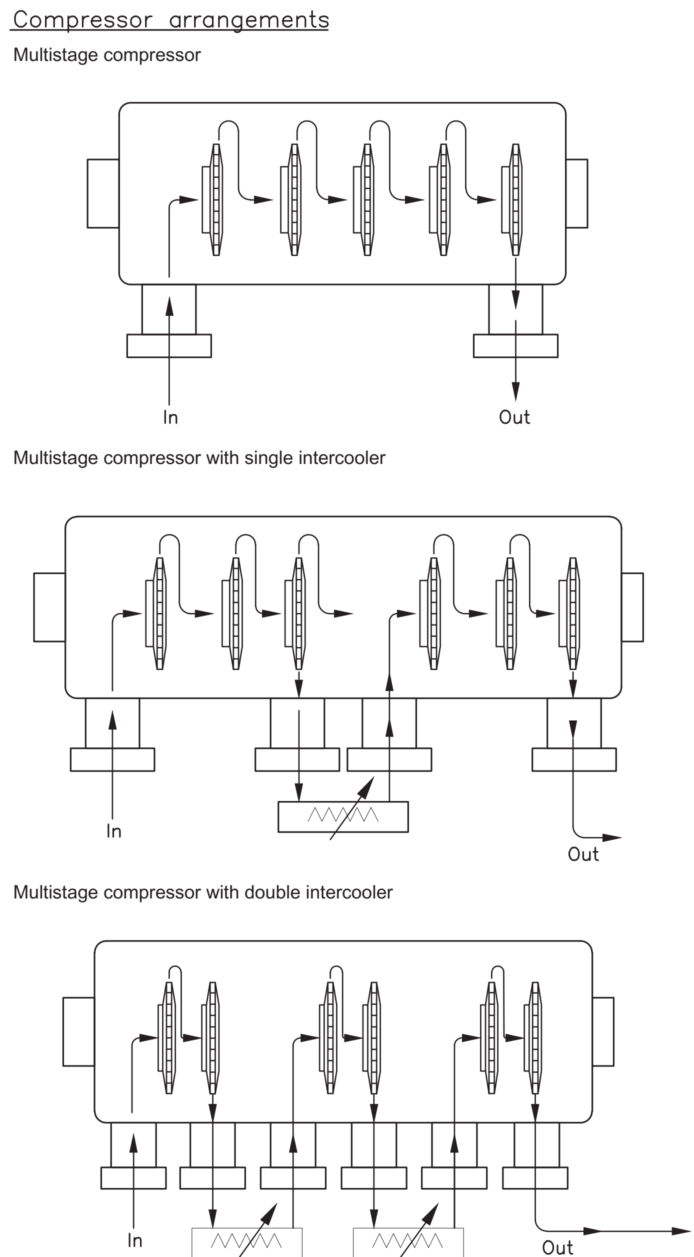
(A)



(B)

Fig. 8.84 (A) Rotor response plot. (B) Single-stage centrifugal compressor arrangements. *Top left*: Radial flow; *Top right*: Axial flow; *Bottom*: Single stage with internally split flow.

(continued)



(C)

Fig. 8.84—cont'd (C) Multistage centrifugal compressor arrangements. *Top*: Multistage compressor; *Center*: Multistage compressor with a single intercooler; *Bottom*: Multistage compressor with double intercoolers.
(A) Courtesy of American Petroleum Institute.

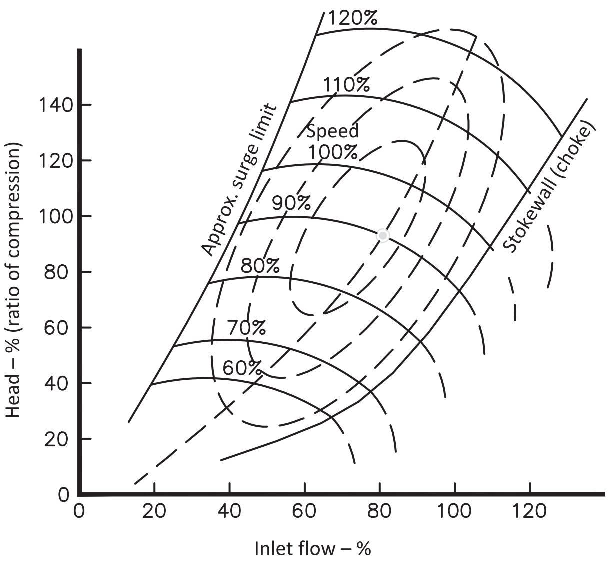


Fig. 8.85 Centrifugal compressor performance map.

- Case design.
- Rotating speed.
- Size of impellers.
- Number of stages.

A performance map cannot be used to predict the performance of a compressor other than the one for which it was developed. The three types of performance maps that are produced to estimate a compressor's performance are as follows:

- Head vs. capacity (actual volume flow)
- Dimensional
- Semidimensional

8.5.2 Head vs. capacity map

The head vs. capacity map is the most commonly used map because it is independent of change in the base conditions of:

- Pressure.
- Temperature.
- Gas Composition.

Fig. 8.86 illustrates a head vs. capacity map. Base conditions are usually listed in the top left corner. Isentropic (adiabatic) head and actual cubic feet per minute (ACFM) are the coordinates and are plotted for lines of constant:

- Speed
- Isentropic efficiency (%)
- Surge limit line (maximum head, minimum flow)

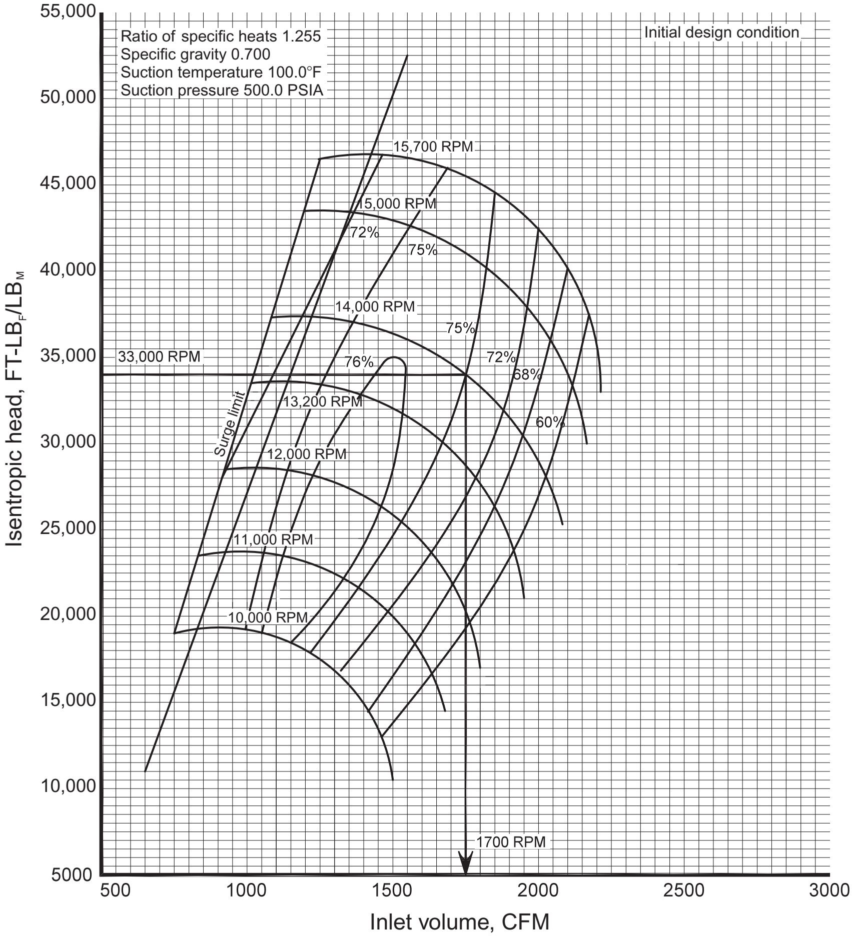


Fig. 8.86 Head vs. capacity performance map.

The line to the right of the surge limit line is referred to as the surge control line. The operating point should always be located to the right of the surge limit line. If any two values of a compressor's operation point are known, the other two values can be determined from the map.

Example 8.1. Head vs. capacity map

Given:

Head vs. Capacity performance map shown in Fig. 8.86.

Determine:

Flow through the compressor and the operating efficiency for the following conditions:

Compressor speed = 14,000 rpm
Adiabatic head = 33,000 ft-lb$_f$/lbm

Solution:

1. Locate 33,000 ft-lb/lb_m on the left side of the map, and draw a horizontal line across to the 14000 rpm speed line
2. Draw a vertical line down to the base of the map, and read the flow at this point. Read 1700 cfm
3. Read the intersection of the horizontal and vertical lines for compressor operating efficiency. Read 75%
4. If a point does not fall on one of the speed or efficiency lines plotted, it will be necessary to interpolate

8.5.3 Dimensional map

The dimensional performance map is shown in Fig. 8.87. This map allows one to predict

- The horsepower and speed requirements when specific flow and pressure conditions are known.
- The maximum possible flow and pressure when maximum horsepower is known.

Pressure (psia) and standard volume flow are the coordinates and are plotted for lines of constant:

- Speed (rpm)
- Horsepower

Also shown are the surge limit and surge control lines. The map can be based on:

- Constant suction pressure, in which case the vertical scale will reflect a range of discharge pressures
- Constant discharge pressure, in which case the vertical coordinates will show a range of suction pressures.

Example 8.2. Dimensional map

Given:

The dimensional map is shown in Fig. 8.87.

Determine:

T values for discharge pressure, horsepower, and standard volume flow have been divided by the map. Horsepower and speed required for the following conditions:

Flow = 75 Mscfd
Discharge Pressure = 1150 psia

Solution:

1. Draw a horizontal line from 1150 psia on the left side of the map to the intersection of a vertical line drawn from 75 Mscfd from the bottom of the map
2. Interpret the intersection point for horsepower and speed required. Read 4000 BMP and 14,000 rpm

If changes in the base conditions of specific gravity, temperature, or pressure occur, significant inaccuracy results.

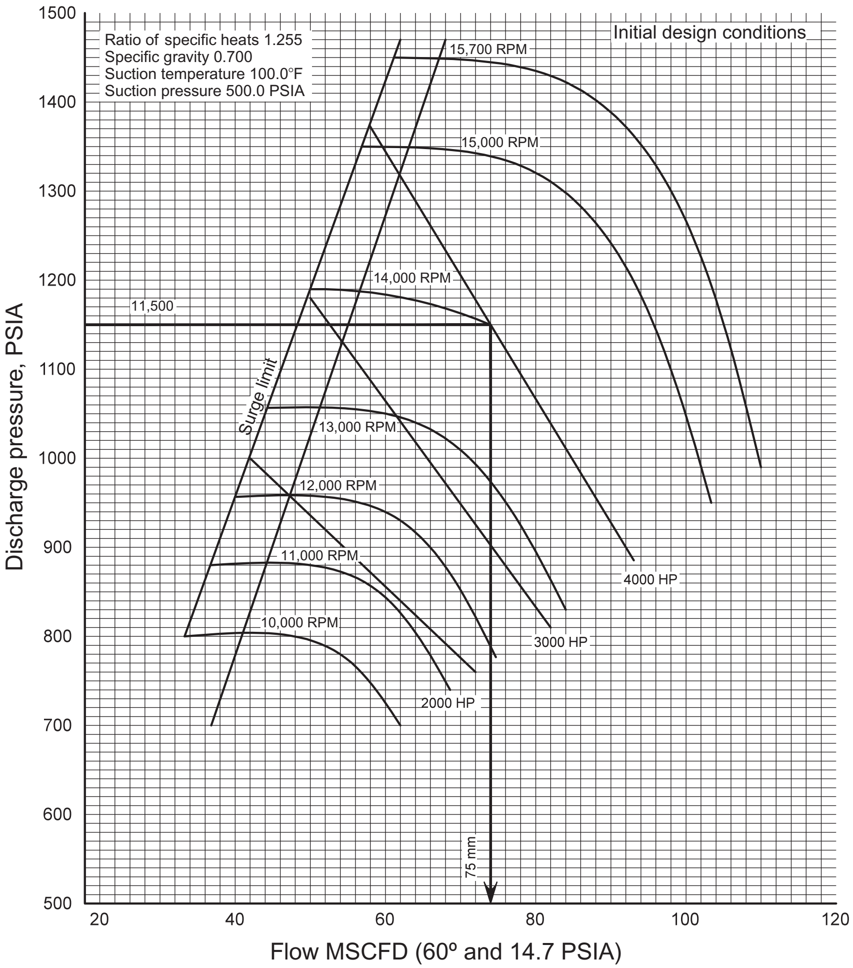


Fig. 8.87 Dimensional performance map.

Thus the dimensional map is useful only for conditions very close to the base conditions.

If base conditions change, the head vs. capacity map or the semidimensional map should be used.

8.5.4 Semidimensional map

The semidimensional performance map is shown in Fig. 8.88. The map is the same as the dimensional map, except that values for discharge pressure, horsepower, and standard volume flow have been divided by the map base pressure.

The semidimensional map can be used when the actual pressure is different from the pressure on which the dimensional map was based. The map is applied by multiplying the operating point values for pressure, flow, and horsepower by the actual base pressure.

Example 8.3. Semidimensional map

Given:

Semidimensional map given in Fig. 8.88.

Determine:

Maximum flow for the following conditions:

Horsepower available = 3500 BHP
Suction pressure = 700 psia
Discharge pressure = 1400 psia

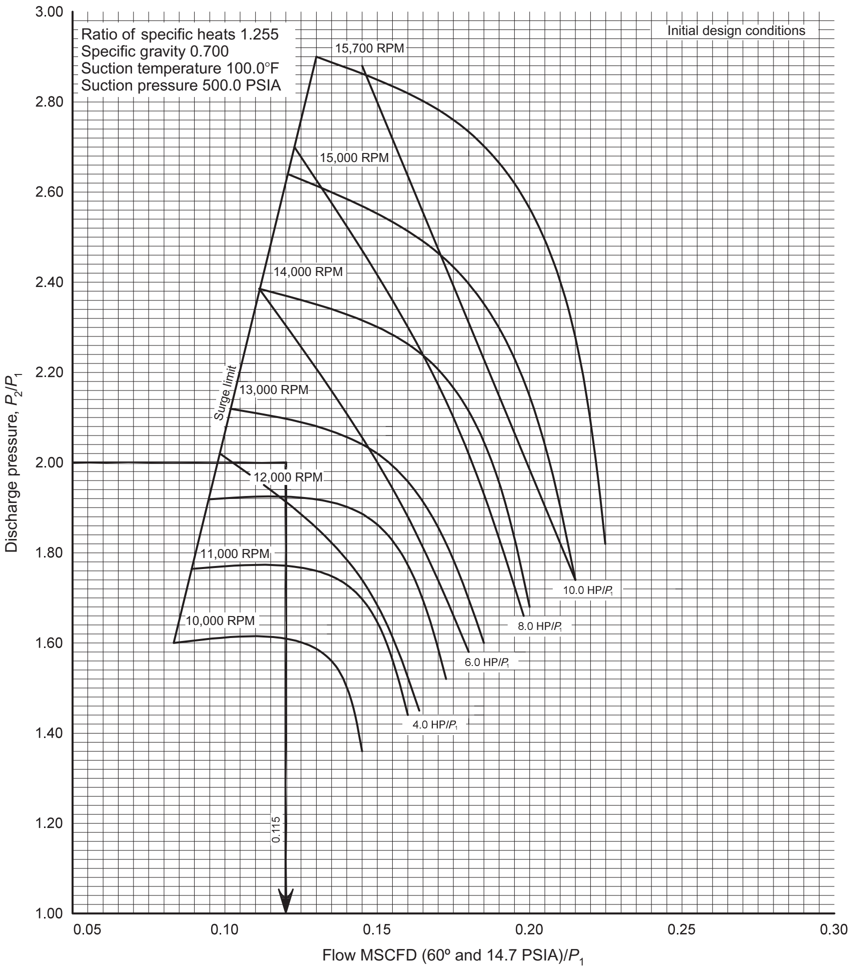


Fig. 8.88 Semidimensional map.

Solution:

1. Divide the horsepower and discharge pressure by the actual pressure base

$$\frac{P_2}{P_1} = \frac{1400}{700} = 2.0$$

2. Draw a horizontal line from 2.0 on the left of the map to the 5.0 HP/P_1 line (halfway between 4.0 and 6.0 HP/P)
3. Draw a vertical line from this point down to the base and read this value

$$0.115\,\text{Mscfd}/P_1$$

4. Determine the maximum flow

$$(700)\,(0.115) = 80.5\,\text{Mscfd}$$

If actual conditions are different from the base conditions, the semidimensional map may not be accurate. In this case, the head vs. capacity map should be used.

8.6 System considerations

8.6.1 Effect of system changes

A centrifugal compressor, like a centrifugal pump, operates at the intersection of its performance curve and the system resistance curve. For constant inlet conditions, the operating point of a variable-speed unit can be changed by either a change in speed or by altering the system curve. Constant-speed unit performance can only be modified by changing the system curve.

Example 8.4. Effect of reducing the inlet pressure

Given:

Fig. 8.89 shows a centrifugal compressor's operating point "A" at the intersection of the system resistance curve and centrifugal compressor's "solid line" performance curve.

Determine:

Determine the effects on the pump's performance if the inlet pressure is reduced.

Solution:

Calculations using the fan laws (assuming a constant inlet volume flow) would indicate revised operation at point "C."

However, since the compressor would actually seek a new operating point at the intersection of its revised performance curve and the system curve, the resulting operation would be at point "B."

If the effects of the system curve are large, estimates made using the fan laws will yield a significant error.

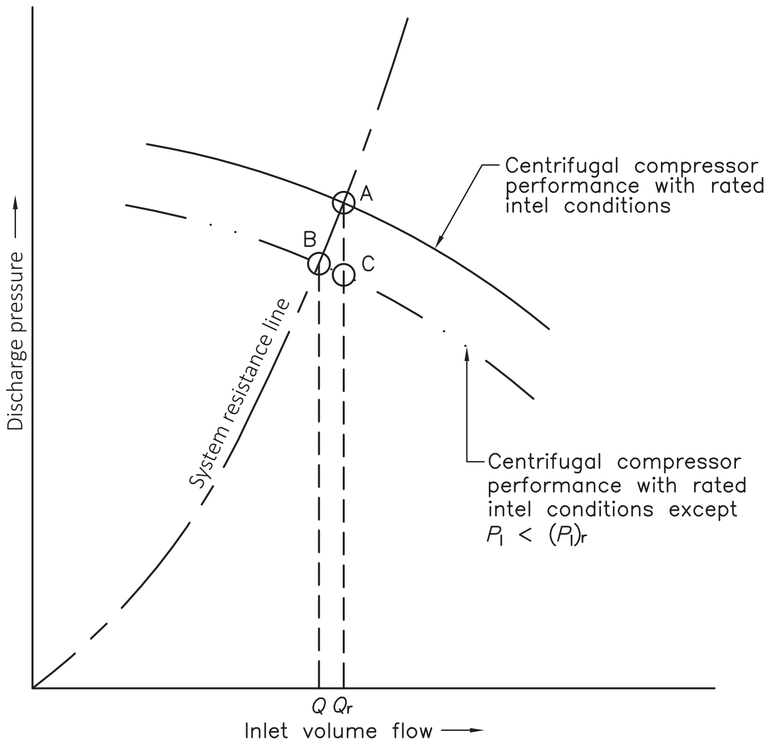


Fig. 8.89 Effect on performance due to a change in inlet pressure.
Courtesy of Compressors: Selection and Sizing, by Royce Brown, 1986, Gulf Publishing Company, Houston, TX.

8.6.2 Stable operating speed ranges

The compressor stability range is discussed in connection with performance curves and surge in Section 9.3. In addition to performance stability, a satisfactory margin must be maintained between the operating speed range and the critical speeds of both the compressor and driver.

Although API 617 defines these required margins, the following can be used as a general guideline:

- *Lateral critical*—should not fall in the range from 15% below any operating speed to 20% above the maximum continuous speed.
- *Torsional critical*—(complete train) no torsional critical should fall in the range from 10% below any operating speed to 10% above maximum continuous speed.

8.6.3 Power margins

The rated horsepower for centrifugal compressor drivers should be a minimum of 110% of the maximum horsepower required for any specified operating point. For motor drivers, it is necessary that the motor be carefully matched to the compressor, and items reviewed, such as:

- Motor speed-torque characteristics,
- Accelerating-torque requirements of the compressor, and
- Motor supply voltage during acceleration.

Steam turbines should have a maximum continuous speed 105% of rated compressor speed. Driver requirements are further detailed in API 617. API Standards 611 and 612 cover general-purpose and special purpose steam turbines.

8.6.4 Series operation

When two or more casings, or sections, are operated in series, the manufacturer will usually furnish two performance maps: one for each casing, and one showing overall casing performance. For determination of the surge volume, use the overall curve.

In most situations, it is desirable to have an individual antisurge recycle line around each casing or around each section of compression of compound casings. It is not practical for one antisurge control to accommodate two casings or sections at operating conditions significantly removed from the rated point. In addition, the overall operating stability range can be improved because the antisurge controls can be set for the stability range of each casing rather than the overall range for all casings.

8.6.5 Weather protection

Centrifugal compressors are normally suitable for unprotected outdoor installations. However, daily temperature fluctuations can affect equipment alignment. Cold temperatures, heavy rains, salt atmosphere, blowing dirt or sand can make maintenance difficult, and maintenance of equipment cleanliness impossible.

Most equipment specification packages include detailed requirements for weather protection of controls and instrumentation. However, conditions vary between locations. Therefore it is important to obtain specific site information from site personnel. Also, make sure the specifications accurately reflect what the field has found to be most trouble free.

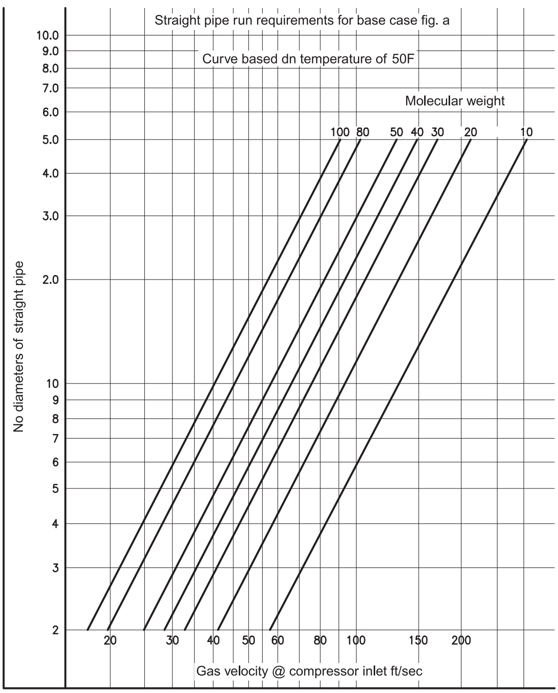
8.6.6 Inlet piping considerations

The inlet piping configuration is an important factor that must be carefully evaluated to ensure satisfactory compressor performance. Performance predictions are based on a smooth, undisturbed flow pattern into the eye of the first impeller. If the flow has any rotation or distortion as it enters the compressor, performance will be reduced.

Fig. 8.90 includes a set of curves that may be used as guidelines to establish a minimum length of straight pipe to be used ahead of the inlet. The base case is shown in Fig. 8.90 (2 of 2) and consists of an elbow turned in the plane of the rotor. Inlet piping methods are shown as multistage compressors and they may be used for axial-entry, single-stage compressors by obtaining the multiplier for the base case and taking the final result and multiplying by 1.25. The higher multiplier accounts for the more sensitive nature of the axial inlet. The pipe lengths are conservative, but should a vendor recommend a longer length, the vendor's recommendation should be followed. When the minimum inlet lengths of piping are difficult to achieve, vanned elbows and straightening vanes can be used (refer to Fig. 8.91).

The nozzle loads, or forces and moments, that the compressor can accommodate without misalignment are generally specified by the manufacturer.

API 617 specifies an arbitrary 1.85 times the limits defined by the NEMA SM-23 Standard. This results in limits which are not practical for all machine types. The



Compressor inlet correction factors
for various piping arrangements

		Factor	Figure
1.	One long radius elbow (plane parallel to rotor)	1.0	a
2.	One long radius elbow (plane normal to rotor)	1.50	b
3.	Two elbows at 90° to each other with second elbow plane parallel to rotor	1.75	c*
4.	Two elbows at 90° to each other with second elbow plane normal to rotor	2.0	d
5.	Butterfly valve before an elbow		
	a. valve axis normal to compressor inlet	1.5	e*
	b. valve axis parallel to compressor inlet	2.0	–
6.	Butterfly valve in straight run entering compressor inlet		
	a. valve axis normal to rotor	1.5	–*
	b. valve axis parallel to rotor	2.0	–
7.	Two elbows in same plane (parallel to rotor)	1.15	1*
8.	Two elbows in same plane (normal to rotor)	1.75	–
9.	Gate valve (wide open)	1.0	–*
10.	Swing check valve (balanced)	1.25	–*

*Factors also apply to single stage, axial inlet compressors.

Note :
1. Factors are applied to the base straight run requirements from the chart.
2. Factors for butterfly valves assume minimum throttling at design conditions. If heavy throttling is required, factors should be doubled.
3. For axial inlets, use 1.25 with appropriate figure.

Fig. 8.90—cont'd

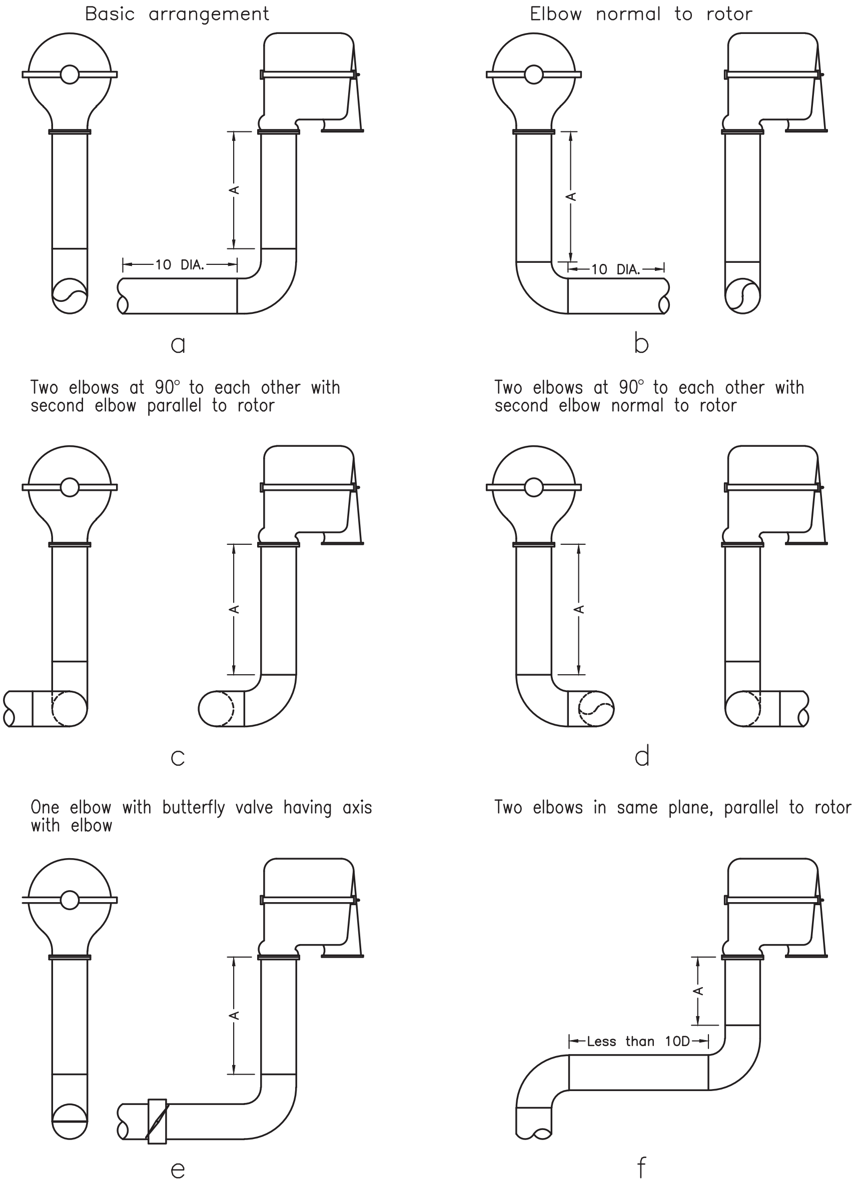


Fig. 8.90—cont'd Chart for minimum straight inlet piping ahead of a compressor inlet. Use this chart and the given factors in conjunction with methods of inlet piping to determine Dimension "A."
1 of 2: Courtesy of Elliott Company. 2 of 2: Courtesy of Elliott Company.

criteria relate allowable loadings only to flange size. For example, a lightly constructed unit with 8-inch, ASME 150 class flanges would be expected to withstand the same loadings as a heavy barrel casing with 8-inch, ASME 2500 class flanges.

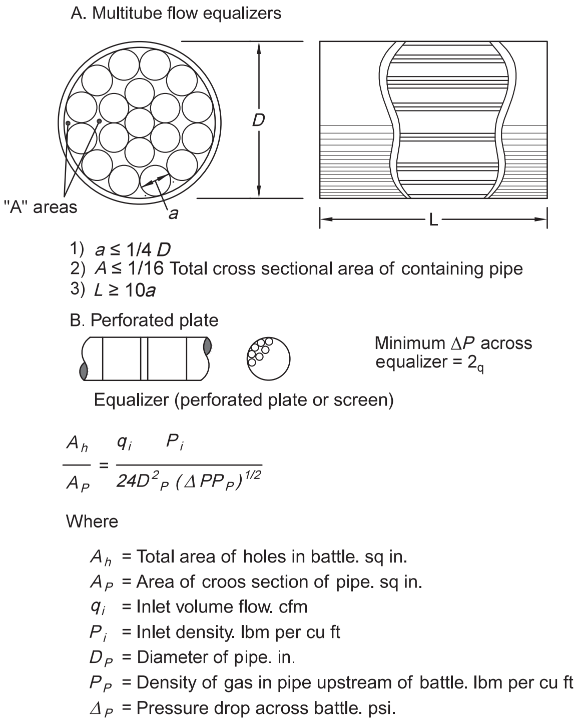


Fig. 8.91 Straightening vanes. Courtesy of Elliott Company.

The design and location of piping supports, and the accommodation of thermal expansion, are generally left to the piping designer. This should be checked in detail during construction to ensure correct installation of piping, and that the location and setting of supports is in accordance with design drawings and specifications.

The following additional items should be considered when reviewing the overall compressor piping design.

- High-velocity streams generate noise. Maximum velocity can be limited by the amount of noise that is allowed.
- No side connections (such as the balance piston return line) should be put in the straight piping run ahead of the compressor inlet.
- When a permanent strainer is used, specified compressor inlet pressure must include an allowance for strainer pressure drop.
- To avoid problems prior to start-up, the compressor manufacturer should be advised of the description and location of each strainer.
- Woven wire mesh should not be used in strainers for centrifugal compressors. Wire mesh has the tendency to plug very rapidly, requiring frequent removal, and in some cases, it has been ingested into the compressor causing serious internal damage.
- Inlet strainers should be located in the first pair of flanges away from the compressor's nozzle. Strainers should not be located right at the suction nozzle, since excessive flow distortion could result.

8.6.7 Lube and seal oil systems

The lubrication of centrifugal compressors is normally handled by a pressurized system, which also provides the seal oil and control oil in some cases. One system usually supplies all machines in a given train (such as the compressor, any gears, and the driver).

A basic pressurized lube system consists of a reservoir, pumps, coolers, filters, control valves, relief valves, instrumentation, and other auxiliaries specific to the application.

Seal oil may be provided from a combined lube and seal oil system, or from a separate seal oil system. Generally, combined systems are selected for sweet gas services. Separate seal oil systems are generally selected for compressors in services that contain hydrogen sulfide or other corrosive or toxic gases. In either type of system, the inner (called "sour") seal oil leakage is not returned to the reservoir. The outer (called "sweet") is returned to the reservoir. Under certain conditions, it is possible for sour gas to migrate into the outer seal oil stream that is returned to the reservoir. Having a separate system positively avoids contamination of the lubricating oil and subsequent corrosive attack of Babbitt-lined bearings and other components served by the lubricating oil system.

API 614, "*Lubrication, Shaft-Sealing, and Control Oil Systems for Special Purpose Applications*" cover the design, manufacture, and testing of the overall system, as well as individual components. Used as a reference, they provide guidelines based on user experience which can easily be scaled down or tailored to fit any requirement.

The system may be designed either as a console or baseplate-mounted package, with all components mounted on a single baseplate, or alternatively as a multiple-package arrangement, with system components separated into individually packaged units. In this case, the individual component packages are piped together in the field.

Oil return lines must slope toward the reservoir(s) to allow gravity draining. This is often overlooked when piping is being laid out. Also, one must be careful to avoid "head knockers" when laying out pipe.

Offshore installations may require a system mounted integrally with the compressor/driver baseplate, with off-mounted air coolers.

The console arrangement, because of its compact layout, may limit or restrict access to various components making maintenance difficult. The multiple-package arrangement allows greater flexibility in locating the individual packages for improved maintenance access. A major disadvantage of the multiple-package arrangement is that the complete system is seldom shop tested and therefore performance is not verified prior to arrival on site.

Careful attention at all phases from initial specification through installation and start-up will contribute significantly to trouble-free compressor train start-up and operation. Historical maintenance data from many compressor installations indicate approximately 20%–25% of centrifugal compressor unscheduled downtime results from instrument problems (many of these associated with operation and control of the lube and seal system).

When designing or modifying a system, obtain specific input from field personnel regarding site requirements, preferences, and operating experience. Field personnel may have already modified the basic system to correct problems experienced, found a particular type or brand of instrument that functions better under their site conditions, or standardized on components to reduce spare parts inventories, and so on.

The following highlights areas requiring special attention:

- For critical or nonspared equipment, include a main and an identical full-sized auxiliary oil pump (not to be confused with an emergency oil pump which is normally of much smaller capacity, sized only to handle lube and seal requirements during coast-down). A popular drive arrangement for turbine-driven compressors is a steam-turbine-driven main oil pump with an electric motor-driven auxiliary. This arrangement has the advantage that auto-start control of the electric-motor driven unit is relatively simple and reliable with rapid acceleration to full speed and rated pressure output. For installations where steam is not available, several alternative drive combinations are used, including motor, shaft driven, and in a few cases air or gas expanders. With motor-driven main and auxiliary pumps, each should be supplied by an independent power source.
- Consider adequate oil flow to bearings and seals during coast-down following a trip of the auxiliary pump. The two approaches used most often involve either an emergency oil pump or overhead rundown tanks. Overhead rundown tanks are typically located to provide an initial pressure (head) equal to the low oil pressure trip pressure. API requires capacity to be sufficient to supply oil for a minimum of three minutes. In the majority of cases this is adequate. A second method is an emergency oil pump. This pump would probably be direct current (DC) motor driven, with power supplied by a battery-backed UPS system.
- Manufacturers often insist that the response time of a motor-driven auxiliary pump is sufficient to avoid pressure decay tripping the main unit, and therefore accumulators are not required. However, several tests have shown this not to be the case. The option should always be held open so that accumulator requirements are based on the system demonstrating acceptable stability during the prescribed testing.
- The system rundown tanks and the accumulators are sometimes confused. The rundown tanks provide lubrication and cooling to bearings and seals during coast-down. The accumulator is designed to maintain system pressure within specified limits during transient conditions or upsets, thus avoiding machinery trips.
- When oil seals are used, the manufacturer is normally asked to guarantee a maximum value for this inner seal oil leakage. The guaranteed value is often found to be considerably lower than actual leakage on test or following start-up. Since size of the degassing tank is based on this leakage rate, the tank often ends up being undersized. API specifies that the degassing tank be sized for a minimum of three times the guaranteed inner seal oil leakage. Actual machine be shutdown. Leakage, however, has in some instances exceeded quoted values by more than 10 times. The manufacturer's sizing criteria should be verified based on review of leakage rate tests for similar seals.
- For centrifugal lube oil pumps, the pump head should be compared to the maximum allowable filter pressure drop (of dirty filters) to ensure that sufficient oil flow is provided to the machinery as the filters become dirty.
- Shaft-driven main lube oil pumps are not recommended, since any maintenance or repair of this pump requires the machine to be shutdown.

8.7 Materials of construction

8.7.1 General considerations

Selection of casing material is influenced by the service involved. Steel casings are required by API 617 for air or nonflammable gas at pressure over 400 psig (27.6 bar) or calculated discharge temperature over 500°F (260°C) (anywhere in the operating range), and for flammable or toxic gas. Stainless steel and high nickel alloys are generally used for low-temperature refrigeration units. A materials guideline which covers recommended materials for compressor components is included in Appendix of API 617.

Even though manufacturers have a background of experience in applying materials and manufacturing processes to special applications, never assume the manufacturer completely understands your process.

The designer should include a complete process gas analysis, with emphasis on corrosive agents, and water vapor, together with any anticipated variation in composition, off-design or alternative operating conditions, or possible process upsets. Specifications should encourage the manufacturer to offer alternatives or comment based on their experience.

When defining the operating environment, also consider the possibility of contaminant buildup during compressor shutdown which might contribute to subsequent component failure. For example, the addition of water or cleaning chemicals during a unit shutdown may add one of the components that lead to a sulfide stress cracking failure.

API imposes specific design limitations for corrosive gas applications. However, actual operating experience may dictate addition or modification to these requirements. API also contains an Appendix of material specifications for major compressor parts.

The following discussion will help one recognize applications where the potential for problems may exist. Detailed descriptions of the failure mechanisms mentioned are beyond the scope of this text.

8.7.2 Sulfide stress cracking

A prevalent problem is sulfide cracking of highly stressed components, especially impellers. It requires the presence of hydrogen sulfide, water in the liquid state, acid pH, and tensile stress.

The use of inhibitors has been investigated, although in most cases the practical solution of operation in this environment has been a change of material. Studies indicate that for materials with yield strengths between 100,000 and 110,000 psi (690–759 MPa), stress levels required for sulfide cracking are near the yield strength. In contrast, materials with yield strengths of 140,000 psi (966 MPa) exhibited susceptibility at stresses as low as 30,000 psi (207 MPa).

Additional studies have resulted in establishing the generally accepted API 617 guidelines, which limit material yield strength to 90,000 psi (620 MPa) or less, and a hardness not exceeding Rockwell C22.

8.7.3 Stress corrosion cracking

Materials operating where the combination of tensile stress, a corrosive medium present, and a concentration of oxygen is acceptable to stress corrosion cracking. The effects of stress and corrosion combine to produce spontaneous metal failure. Because all conditions required for stress corrosion cracking are less likely to exist in a normal environment, corrosion cracking is not as common. Also, materials modified for sulfide cracking produce a material less susceptible to stress corrosion.

8.7.4 Hydrogen embrittlement

Compressors handling hydrogen (hydrogen at partial pressures greater than 100 psig (6.9 bar), or concentrations greater than 90-molar-percent at any pressure) are susceptible to hydrogen embrittlement. This embrittlement occurs when a metal is stressed in a hydrogen-rich atmosphere.

Metals highly prone to embrittlement include high-strength steels and high-strength nickel base alloys. Those having only a slight tendency includes titanium, copper, austenitic stainless steels, and aluminum alloys, with most materials commonly used on centrifugal compressors falling in between. The most practical solution has been found in selection of material properties compatible with the process involved.

API 617 limits impellers to 120,000 psi (8274 bar) yield strength and a hardness less than Rockwell C34. Fig. 8.92 shows that this stress level is for overspeed RPM and is therefore conservative at running speed.

8.7.5 Low temperature

Standard compressor casing materials are generally good for temperatures of −20 to −50°F (−28 to −46°C). Below these temperatures, standard materials become brittle, and materials with improved low-temperature properties must be used.

Nickel-based steel alloys are generally used, with suitable alloys available for both fabricated and cast casings, for temperatures to approximately −150°F (−101°C). Special nickel alloys and austenitic stainless steels may be used for temperatures to −320°F (−195°C).

Speed	Speed %	Stress %	Stress as % of minimum specified yield strength*
Design	100	100	69
Maximum continuos	105	110	75
Trip	115	130	90
Overspeed	121	145	100

* Percentages based on assumption that stress at overspeed testing approaches but does not equal the minimum specified yield strength. Actual conditions will reduce percentages by some factor.

Fig. 8.92 Impeller Stresses at various speeds of rotation. Courtesy of Elliott Company.

Also review other component materials for compatibility with the operating temperature range. The materials Appendix of API 617 is an appropriate guide for material selection since temperature limits specified indicate limits commonly applied by compressor manufacturers. An unusual example of the application of low-temperature material requirements is an air compressor located in a cold climate region. Although this compressor might be located in an enclosed (even heated) building, it could be exposed to inlet air temperature well below −50°F (−46°C). Suction throttling would further reduce inlet temperatures.

Where reduced maximum yield strength and hardness are specified, apply the same requirements to any welding and repair procedures.

8.7.6 Impellers

Centrifugal compressor impellers are most commonly made from alloy steel forgings of AISI 4140 or 4340. Materials such as AISI 410 stainless steel and precipitation hardened stainless steels (including Armco 17-4 pH or 15-5 ph) may be used in situations where corrosion resistance is required. Austenitic stainless steels, monel, and aluminum, although somewhat limited in their application, are used in some special cases, Fig. 8.93 identifies the chemical analysis of various impeller materials. Fig. 8.94 provides a listing of mechanical properties.

8.7.7 Nonmetallic seals

Elastomeric seal requirements in centrifugal compressors are generally handled by O-rings. Since compressor applications seldom involve pure gases or fluids, selection of the proper O-ring material can become quite difficult. One should carefully evaluate the operating environment, considering factors such as temperature, pressure, and fluid composition (with special emphasis on corrosiveness of the gas). Operating experience in the same or similar service is of prime importance. Fig. 8.95 provides "application charts" for typical O-ring materials.

8.7.8 Coating considerations

Coatings are not widely used to improve corrosion or erosion resistance of compressor internals. Problems include:

- Surface preparation prior to coating.
- Maintenance of critical tolerances.
- Balancing coated components.
- Protection of coating during handling.
- Modification of established manufacturing procedures.

Selection of compatible materials or material properties is generally the most practical approach.

Iron base alloys

	Carbon	Manganese	Silicon	Chromium	Molybdenum	Nickel	Copper	Columbium
AISI 4140	0.38–0.43	0.75–1.00	0.20–0.35	0.80–1.10	0.15–0.25	—	—	—
AISI 4340	0.38–0.43	0.60–0.80	0.20–0.35	0.70–0.90	0.20–0.30	1.65–2.00	—	—
AISI 410	0.15 max.	1.00 max.	1.00 max.	11.50–13.50	0.50 max.	0.50 max.	—	—
AISI 304	0.08 max.	2.00 max.	1.00 max.	18.00–20.00	—	8.00–12.00	—	—
AISI 304L	0.03 max.	2.00 max.	1.00 max.	18.00–20.00	—	8.00–12.00	—	—
AISI 316	0.08 max.	2.00 max.	1.00 max.	16.00–18.00	2.00–3.00	10.00–14.00	—	—
AISI 316L	0.03 max.	2.00 max.	1.00 max.	16.00–18.00	2.00–3.00	10.00–14.00	—	—
Armco 17–4PH	0.07 max.	1.00 max.	1.00 max.	15.50–17.50	—	3.00–5.00	3.00–5.00	0.15–0.45

Nickel base alloys

	Carbon	Manganese	Chromium	Molybdenum	Nickel	Copper	Columbium & Tantalum	Aluminum	Titanium	Iron
Monel K–500	0.25 max.	1.50 max.	—	—	63.00–70.00	Balance	—	2.00–4.00	0.25–1.00	2.00 max.
Inconel 625	0.10 max.	0.50 max.	20.00–23.00	8.00–10.00	Balance	—	3.15–4.15	0.40 max.	0.40 max.	5.00 max.

Aluminum base alloys

	Copper	Silicon	Iron	Manganese	Magnesium	Zinc	Cromium	Titanium
2025	3.90–5.00	0.50–1.20	1.00 max.	0.40–1.20	0.05 max.	0.25 max.	0.10 max.	0.15 max.
C355	1.00–1.50	4.50–5.50	0.20 max.	0.10 max.	0.40–0.60	0.10 max.	—	0.20 max.

Fig. 8.93 Chemical analysis of impeller materials.
Courtesy of Elliott Company.

Material	Ultimate tensile strength psi, min.	0.2% Yield strength, psi	% Elongation, min.	% Reduction in area, min.	BHN
Iron base alloys					
AISI 4140	100,000	80,000–90,000	16	50	212–235
AISI 4140	115,000	95,000–120,000	16	50	269–321
AISI 4340	120,000	105,000–120,000	17	43	248–302
AISI 4340	125,000	115,000–135,000	16	50	269–321
AISI 410	100,000	80,000–90,000	14	40	212–235
AISI 304	75,000	30,000 min.	40	50	205 max.
AISI 304L	75,000	30,000 min.	40	50	187 max.
AISI 316	75,000	30,000 min.	40	50	217 max.
AISI 316L	65,000	25,000 min.	40	50	207 max.
Armco 17–4PH	130,000	105,000 min.	16	50	277–341
Nickel base alloys					
Monel K–500	130,000	90,000 min.	20	–	250 min.
Inconel 625	120,000	60,000 min.	30	–	—
Aluminum base alloys					
2025T6	52,000	32,000 min.	10	–	100 min.
C355T62	44,000	33,000 min.	2.5	–	—

Fig. 8.94 Mechanical properties of impeller materials.
Courtesy of Elliott Company.

8.8 Instrumentation and control selection

8.8.1 Standard instrumentation

Fig. 8.96 shows instrumentation commonly used on centrifugal compressors. API 614 and 617 data sheets include additional instrumentation options. These data sheets provide a good checklist for defining the requirements of a specific application. Whenever alarms and shutdowns are used, it is important to make sure they are installed with provisions to allow testing.

Temperature ranges of elastomeric "O" rings in air

Temperature range °F	Preferred elastomeric material
−150° to −40°	Low temperature silicone (special)
−80° to +450°	Standard silicone
−40° to +175°	Chloroprene
−40° to +212°	Nitrile
−40° to +400°	Fluoroelastomer
+68° to +550°	Perfluoroelastomer
+450° to +500°	High temperature silicone (special)

"O" ring application chart

Media groups	Temperature °F max.	Elastomeric material
Hydrogen & methane	212 300 400	Nitrile Fluoroelastomer Fluoroelastomer
Chlorine (dry), ethylene & propylene	200 300 400	Fluoroelastomer Fluoroelastomer Fluoroelastomer
Air & nitrogen	212 300 400 450	Nitrile Fluoroelastomer Fluoroelastomer Silicone
Isobutane, butane & propane	212 400	Nitrile Fluoroelastomer
Water, flue gas, ethylene, feedgas & recycle	212 400 450	Nitrile Fluoroelastomer Silicone
Refinery feedgas, benzene, toluene, xylene, coke oven gas, and many synthetic functional fluids	400	Fluoroelastomer
Petroleum base lubricants	250 350	Nitrile Fluoroelastomer
Wet refinery feedgas	400 450	Fluoroelastomer Silicone
Ammonia (gas)	175 212 450	Chloroprene Nitrile Silicone
Oxygen	200 300 400	Fluoroelastomer Fluoroelastomer Silicone

Fig. 8.95 O-ring application charts.
Courtesy of Elliott Company.

	Indicator	Alarm	Shutdown
Lube and seal system			
Lube oil pump discharge pressure		X	
Oil header pressure (each level)	X		
Low lube-oil header pressure		X	X
Standby oil pump running	X		
Seal-oil pump(s) discharge pressure	X	X	
Seal-oil differential pressure		X	
Standby seal-oil pump running	X		
Low seal-oil level		X	
Low seal-oil pressure		X	X
Run-down tank level		X	X
Compressor			
Compressor flow rate	X		
Compressor suction pressure low and high (each section)	X		
Compressor discharge pressure low and high (each section)	X		
High compressor discharge temperature		X	WS[a]
Journal bearing temperature	WS[a]	WS[a]	WS[a]
Thrust bearing temperature	WS[a]	WS[a]	
High liquid K.O. levels	X	X	X
Surge event		X	
Shaft Vibration	X	X	X
Axial Position	X	X	X

a *WS*, when specified.

Fig. 8.96 Instrumentation typically used on centrifugal compressors.

8.8.2 Compressor control

The control system regulates the compressor output to satisfy the process requirements and keeps the compressor from operating in surge. Performance requirements are usually established during the process-design phase, based on a cooperative effort between the process engineer and machinery engineer. Although control parameters for an existing process may already be set, (making selection of the compressor control system relatively straightforward), a process update or modification, a change in type of compressor or driver, or a need for improved efficiency, may dictate a change.

An understanding of the effect of varying gas conditions on compressor performance is necessary to properly evaluate control alternatives. Fig. 8.97 shows the performance curve for a centrifugal compressor operating at constant speed with varying inlet conditions.

8.8.3 Control system selection

8.8.3.1 Variable speed vs. constant speed

The two most common control methods are variable-speed and constant-speed suction throttling. Adjustable inlet guide vanes are sometimes used, primarily on single-stage units. Turbine-driven centrifugal compressors typically use variable speed, with either pressure or flow as the controlled variable. Variable-speed motors and hydraulic or electric variable-speed couplings are seldom applied to centrifugal compressors due to their added cost, and because they significantly lower the efficiency of the unit.

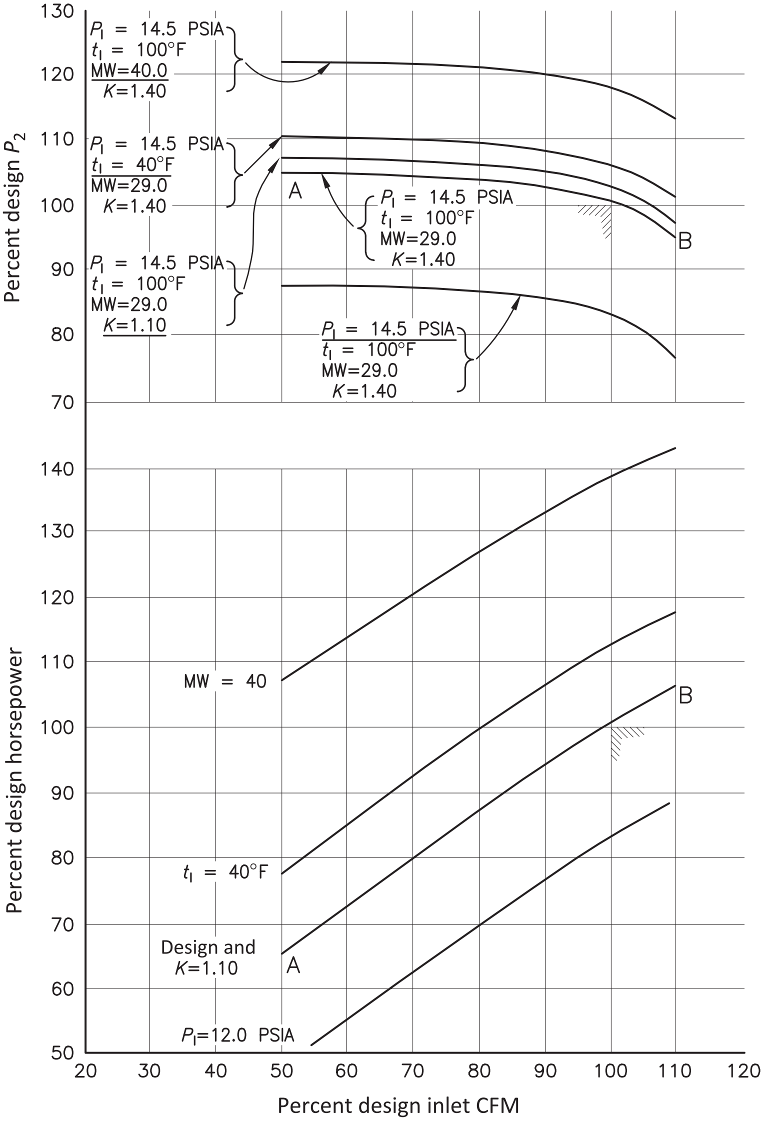


Fig. 8.97 Effects of changing gas conditions at constant speed.
Courtesy of Elliott Company.

A review of centrifugal compressor characteristics highlights the differences between these two methods:

- *Variable speed***:** Capacity varies directly with speed and the head varies proportional to the square of speed. Therefore, as speed is reduced, capacity and head are reduced to meet the process requirements, with a corresponding reduction of horsepower and a minimum loss in efficiency.

- *Constant speed*: Operation essentially produces a constant head. Throttling reduces the inlet and outlet pressures but adds losses by introducing added resistance to the system.

Fig. 8.98 shows typical constant-speed performance curves indicating the effect of suction throttling. Fig. 8.99 shows typical variable-speed performance curves. A comparison gives an indication of the difference in power requirements between the two methods.

For a capacity requirement of 80%, suction throttling requires approximately 86% horsepower. For the same 80% capacity, control by variable speed requires approximately 81% horsepower.

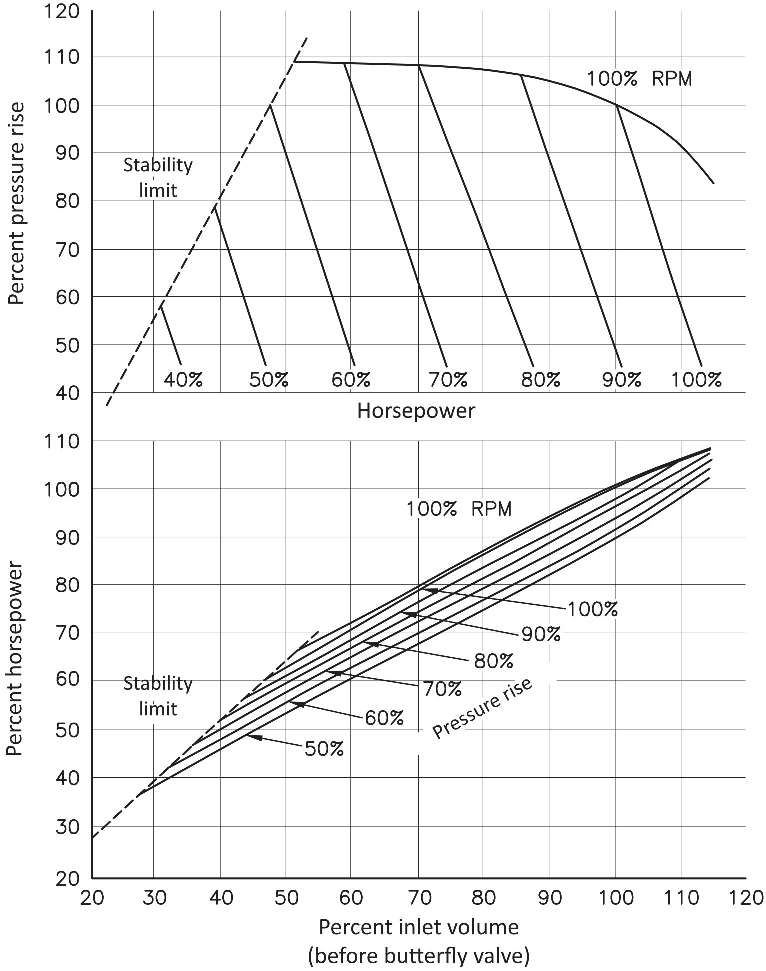


Fig. 8.98 Constant-speed performance curves.
Courtesy of the Elliott Company.

8.8.3.2 Parallel operation

Parallel operation of two or more compressors adds additional complexity to the control system. Slight variations in compressor performance characteristics, piping configuration, and instrument settings can cause one unit to take all the load, thus forcing the others into recycle, or alternatively causing endless "hunting" between units. For example, if one unit starts to recycle slightly ahead of the other and suction temperature is increased due to the recycle, its capacity to produce head will be reduced, thereby locking this unit into recycle. Alternatively, if suction temperature is reduced by recycle, head output is increased forcing the other unit into recycle starting a back-and-forth swing between units.

Simulation studies are often necessary because of the complexity involved in matching parallel compressors. The designer should direct their efforts toward developing the least complex control logic that will meet process and operating requirements. One common approach is to base load one unit, allowing the second unit to take process swings.

8.8.4 Surge control

In the case of air compressors, surge control is often accomplished by a discharge blowdown valve, regulated to maintain the required minimum flow to the compressor. This is based on a minimum flow setting and is applicable only for units operating at constant inlet conditions. In most applications, however, it is necessary to recycle flow back to the suction, through a bypass cooler, in order to maintain stable operation.

8.9 Retrofitting and rerating considerations

8.9.1 General considerations

Often it is desirable to modify process conditions to improve facility efficiency or to increase production. This often requires rerating an existing compressor. Before the designer spends a considerable amount of time and effort in redesigning the process, it is advisable to make a preliminary feasibility estimate to determine the rerate capabilities of the existing compressor. The feasibility estimate identifies various limitations and helps avoid completing a total process redesign only to find out that a compressor cannot meet these new requirements.

The major areas which require evaluation include capacity, pressure, speed, and power. The designer should consider consulting the original equipment manufacturer (OEM) and/or a company specialist before making significant changes to any critical (unspared) centrifugal compressor.

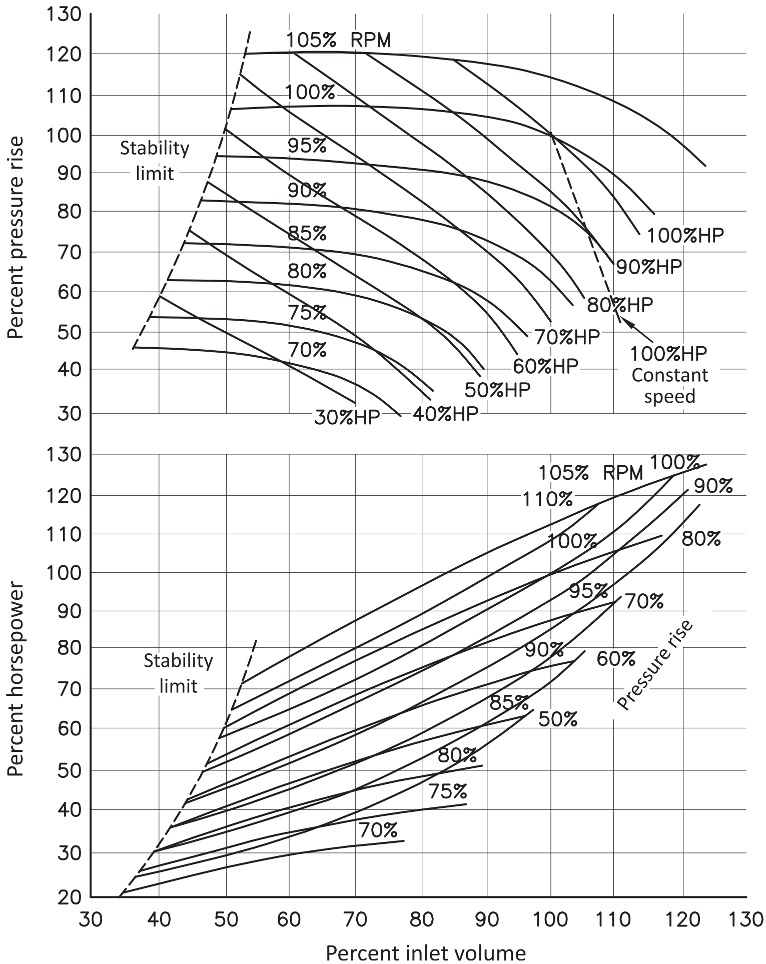


Fig. 8.99 Variable-speed performance curves.
Courtesy of the Elliott Company.

8.9.2 Capacity

Impellers and internal stationary components can be relocated and new ones added, the casing nozzle sizes are fixed. The maximum capacity that can be handled with reasonable pressure drop is dependent on the nozzle size and related to inlet gas velocity.

Inlet velocity is dependent on gas conditions, allowable noise levels, and inlet piping configurations. An acceptable rule of thumb is a maximum of 140 ft/sec (42.67 m/sec) for air or lighter gases and approximately 100 ft/sec (30.48 m/sec) for heavier hydrocarbons.

The actual inlet gas velocity can be calculated from:

$$V = 3.06 \frac{Q}{D^2} \tag{8.15}$$

where:
Q = ACFM in ft^3/minute at inlet pressure, temperature, Z, MW
D = Inside diameter of the nozzle, inches

If side load or compound inlets are involved, inlet gas velocity should be checked for all inlet connections.

8.9.3 Pressure

Next, check the pressure rating of the existing unit. During manufacturer, the casing was hydro-tested to 1-1/2 times the maximum operating pressure (nameplate rating). If the pressures involved in the rerate exceed the nameplate rating, it will be necessary to re-hydro-test the casing for the new. The following items should be noted:

- It may be necessary to check with the manufacturer to confirm that the casing design pressure is adequate for rerating and re-hydro-testing.
- Compressor operating characteristics, relief valve settings, or settle out pressures may set the maximum operating pressure.
- If set by compressor characteristics, use pressure rise to surge at maximum continuous speed.
- Sidestream or compound compressors may have been hydro-tested by sections with a different pressure for each. Check each section for compatibility with new conditions.

Check the compressor to determine its capability of producing the head required.

Use Eq. (8.2) to calculate the head for the rerated condition based on the desired pressure ratio. An attempt may be made to reuse some or all of the existing impellers, based on an overall polytropic efficiency of 70% for the initial estimate.

Initially estimate the speed from the fan (affinity) law:

$$N_2 = N_1 \left[\frac{H_2}{H_1}\right]^{\frac{1}{2}} \tag{8.16}$$

where:
N_1 = Original speed
N_2 = Rerated speed.
H_1 = Head for rerated pressure
H_2 = Head for original pressure

This same procedure will work for applications involving side loads or intercooling between sections. The head for each section is determined based on the conditions for the section, and the total head is the sum of the individual section heads.

8.9.4 Power

Since motor drivers are seldom oversized, anything more than a minor power increase may require a new motor. This requires close evaluation of proposed process changes to see if necessary improvements can be achieved while still staying within the driver's capabilities.

In contrast, turbines and gears can usually be modified to provide increased power. Although turbine data sheets will sometimes provide information regarding maximum steam flow or update capabilities, discussions with the manufacturer may be required.

From Eq. (8.14), one can see that gas horsepower (GHP) is directly proportional to weigh flow (w) and head (H), or:

$$\mathrm{GHP}_2 = \mathrm{GHP}_1 \left[\frac{w_2 H_2}{w_1 H_1}\right] \tag{8.17}$$

For example, if weight flow is increased by 10% and head is increased by 10%, the power requirement is increased by:

$$1.10 \times 1.10 = 1.21 \text{ or } 21\%$$

Furthermore, a driver power margin of 10% is recommended. Therefore the total recommended requirement is increased by:

$$1.21 + 10\%(1.21) = 1.33 \text{ or } 33\%$$

8.9.5 Speed

Lastly, review the speed based on impeller stress and compressor critical speeds.

Impeller stress levels are related to the impeller tip speed as discussed in Section 9.2. While maximum allowable tip speeds vary with manufacturer, impeller design, and material, a good rule of thumb for impellers with backward leaning blades is 900 ft/sec (274 m/sec) maximum tip velocity.

Determine impeller tip speed by:

$$u = \frac{DN}{229} \tag{8.18}$$

Or, using the 900 ft/sec (274 m/sec), maximum speed is:

$$N_{\max} = \frac{229(900)}{D} \tag{8.19}$$

Maintain the following critical speed separation margins:

- Any critical speed at least 20% below any operating speed
- Any critical speed at least 20% above maximum continuous speed

Revamping of the rotor may have some effect on critical speeds; however, ignore this effect for the initial feasibility estimate.

8.10 Axial compressors

8.10.1 General considerations

An axial compressor is a dynamic compressor similar in operation to that of a centrifugal compressor. They are high-speed, large volume compressors but smaller and usually more efficient than centrifugal compressors. The initial capital expense is usually higher than that of a centrifugal but is justified when one considers the lower operating expense (energy cost savings) when determining the total lifecycle cost. The pressure ratio per stage is less than that of the centrifugal. Therefore it takes approximately twice as many stages to achieve the same pressure ratio as would be required by a centrifugal. Typical pressure ratios for air service are 2.5–7 with the highest of about 14.

Axial compressors are found in only a few high capacity industrial gas services, generally services greater than 6000 Hp and over 75,000 cfm. They might be found in upstream petrochemical processes that include high capacity air compressor services. Most combustion gas turbines have an axial compressor in front of their combustor. A major application of the axial compressor is the aircraft engine.

Axial compressors are usually not applied in flammable or hazardous gas services because they have too many paths to atmosphere.

Table 8.4 lists typical application ranges of axial compressors.

Table 8.4 Typical application ranges of axial compressors

Ranges		
ICFM	Typical	75,000–250,000
	Low	30,000
	High	1,000,000
Discharge pressure	Typical	15–150 psig
	High	550 psig (special design for LNG plant)
Discharge temperature	Typical	400–650°F
	High	720°F
No. of stages per casing	Typical	6–15
	High	20 (special to 22)
Adiabatic head per stage, ft	Typical	4000–5000
	High	6000
Speed. RPM	2800–12,000	
Bhp per casing	Typical	6000–50,000
	High	over 100,000

8.10.2 Performance

Axial compressors are characterized by a purely axial flow path. The flow moves axially through a set of rotating blades, through the stationary stator blades for that stage, and so on to the next stage of rotating blades. Axial compressors have free-standing blades that are sealed at the blade tip. The compressor casings are designed for high capacity but not high pressure. They are physically smaller and lighter in weight than a centrifugal (radial) compressor of equivalent duty.

Axial compressors have a very narrow stable operating range at constant speed—about 12%. Some designs utilize one or more centrifugal impellers on the high-pressure end of the rotor which greatly improves the stable operating range. They can be fitted with variable stator vanes on the 1st through 5th (and higher) stages to widen the performance map. Machines so fitted often operate at constant speed.

Axial compressors are generally used for air service—high volume and relatively low pressure.

Advantages over a centrifugal compressor include:

- More efficient.
- Relatively quieter operation.
- Physically smaller and lighter than a centrifugal for the same duty.
- Speed is somewhat higher than that of a centrifugal for the same duty.
- Two casings can be put in tandem arrangement, but it is seldom done.
- Some designs have provisions for intermediate nozzles for intercooling or sidestreams, but this feature is not often employed.

8.10.3 Performance curve

An axial compressor's performance curve can be changed by altering a variable inlet stator vane. The variable stator vane gives the axial compressor, with its inherently steep pressure volume curve, the ability to be matched to a changing load without changing speed. Fig. 8.100 is a pressure-volume map for an axial compressor with a partial stator vane control. (Note: Only a portion of the stator blading is movable.)

Even though the axial compressor is a dynamic compressor, the shape of the axial compressor's performance curve is "steep" compared to that of a centrifugal compressor's "flat" curve. The characteristic horsepower curve for an axial compressor also contrasts the shape of a centrifugal horsepower. As shown in Fig. 8.101, the centrifugal compressor's required horsepower increases as flow (volume) increases while that of the axial compressor decreases. This characteristic needs to be considered when starting up an axial compressor. The axial compressor is unloaded by opening a discharge bypass or otherwise removing the downstream load restriction during start-up. This means that the lowest gas load is away from surge, compared to the centrifugal, which may be unloaded by discharge throttling. This will bring the machine up to speed range.

The steep pressure-volume curve allows the axial compressor to operate well in parallel with other axial compressors. Pressures do not have to match exactly to allow load sharing, as the steepness of the curve allows for adjustment without the possibility of going into surge or experiencing extreme load swings as normally happens when attempts are made to operate centrifugal compressors in parallel.

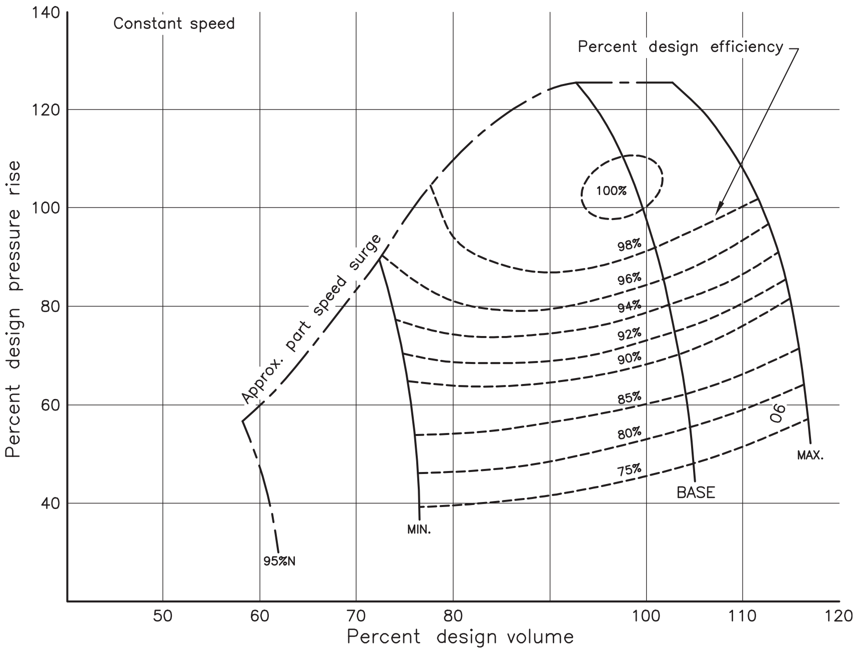


Fig. 8.100 Pressure-volume chart for an axial flow compressor with partial stator vane control. Courtesy of GE Energy.

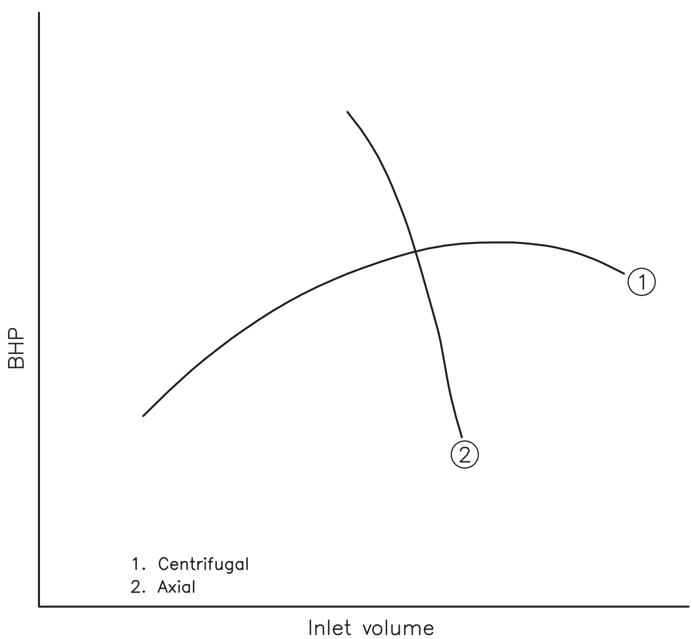


Fig. 8.101 Typical brake horsepower vs. inlet volume curves for a centrifugal and an axial flow compressor.

Axial compressors match up well with a centrifugal compressor in a tandem-driven, series-connected arrangement when higher process pressures are needed. Interstage matching is easy as the steep curve will provide sufficient changes in discharge volume to easily accommodate the requirements of the centrifugal compressor.

8.10.4 Mechanical components

8.10.4.1 Casings

Casings can be fabricated from steel or cast from cast iron or cast steel (refer to Fig. 8.102). Some designs incorporate an outer shell containing an inner shell, which acts as the stator vane carrier. In some designs, the stators are directly carried on the casing, which are a one-part construction. With this design, the casing is made up of three distinct parts (inlet, center, and discharge), bolted at two vertical joints. In the three-piece bolted construction, a combination of fabrications and castings can be used.

Some designs use a rectangular inlet section to provide more axial clearance (refer to Fig. 8.103). The reason to use an entire separate casing is that it forms a separate pressure casing that can be hydro-tested. The disadvantage is that there is more material involved which makes it more expensive.

Compressors that use the integral stator section or single-case construction are less expensive and have the advantage of keeping the stator carriers round due to the end-bolting being bolted to other components. The disadvantage is that there are more joints to seal and maintain.

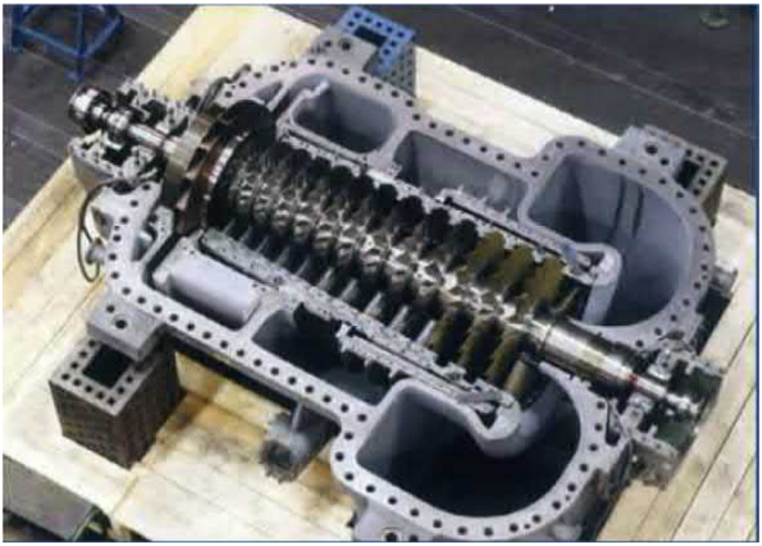


Fig. 8.102 Cutaway of axial compressor showing casing and rotor.

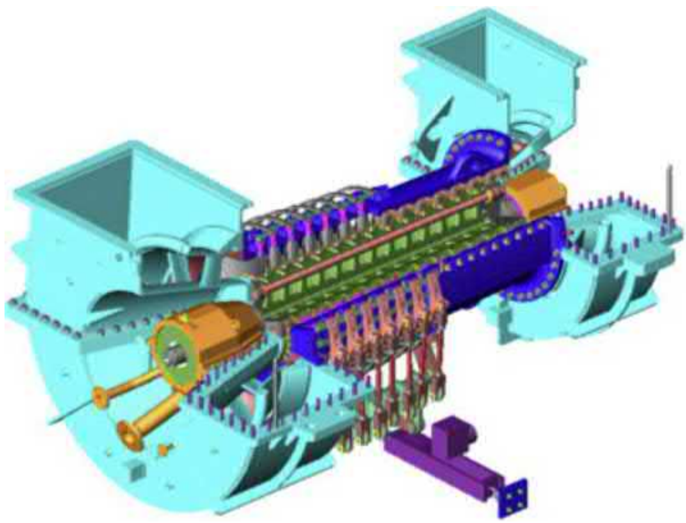


Fig. 8.103 Axial compressor casing with large rectangular inlet and outlet connections.

8.10.4.2 Casing connections

The casing inlet and outlet are flanged for connection to the piping system, with the exception of some rectangular inlet connections. Axial compressors are relatively small machines with disproportionately large piping which results in large forces and moments. Therefore it is imperative the manufacturer be contacted to determine allowable forces and moments.

8.10.4.3 Stators

Stators may be carried in a separate inner casing or may be carried by the outer center section of the main casing. When movable stator vanes are used, the vanes pass through the wall of the carrier. The outer side of the casings exposes the shank ends, which are used as shafts to connect to the linkage that will control the movement. The single case unit has some sealing problems that do not exist with the double-case construction. The single-case construction uses a lagging over the linkage that can act as a collector. For nonmovable construction, conical shank and bolt arrangements are used. Other bolting arrangements are manufactured to give the same flexibility as the conical shanks.

8.10.4.4 Bearings

The bearings used in axial compressors are the same journal and thrust type used in the centrifugal compressor.

8.10.4.5 Blades and diffuser

Inducer blades (adjustable inlet guide vanes) guide the fluid as it enters the wheel (refer to Fig. 8.104). They enable the characteristics of the compressor to be adapted to the variations in the characteristics of the system it supplies. In other words, the volume of fluid can be maintained constant by adjusting the position of these blades.

The purpose of the diffuser is to convert the dynamic pressure of the fluid into static pressure by reducing the velocity. The vanes in the diffuser are sometimes adjustable, in which case they play the same part as the adjustable inlet guide vanes.

8.10.4.6 Rotor

Rotor construction varies from vendor to vendor. The blades are attached to the outer surface of the rotor as shown in Fig. 8.105.

8.11 Foundations

8.11.1 General considerations

In addition to knowing the dimensions and weights of the machinery to be supported, engineers designing the foundation must know the magnitude, direction, and frequency of the dynamic forces that the machinery will exert on the foundation.

The importance of foundations to a compressor installation cannot be overemphasized. Foundations attenuate vibratory forces generated by the machinery and reduce transmission of these forces to the surrounding plant and equipment. Foundations also keep the machinery in alignment.

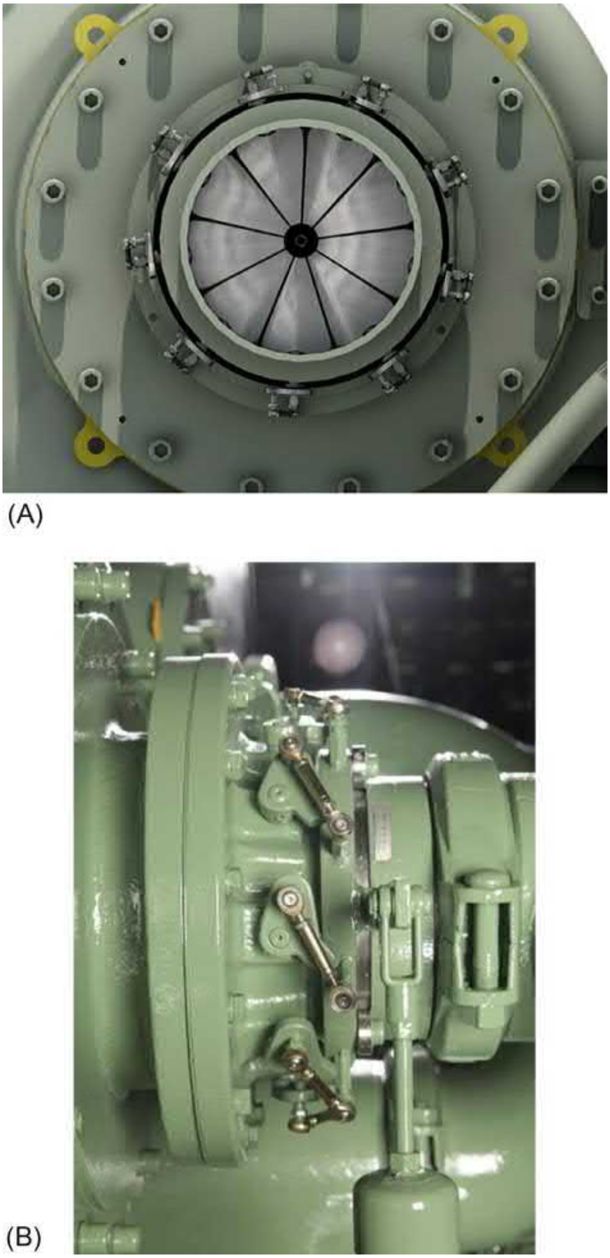
This section provides a basis for establishing the dynamic forces to be used by civil engineers in foundation design calculations. Soil mechanics, natural frequency calculations, bearing pressure, herein.

To perform these essential functions throughout the life of the installation, the foundation must be sized to support the weight of the machinery while imposing a tolerable bearing pressure on the soil or structure. It must be properly designed so that the system, consisting of the foundation, soil, machinery, and piping, is not at or near a resonant condition. It is particularly important on offshore structures, which may be susceptible to resonance from the machinery vibration.

The purchaser of the machinery is normally responsible for the design of the foundation. The vendor or manufacturer of the machinery will seldom take this responsibility because his expertise is not in this field. It would not be in his interest to accept the risks associated with the design. Additionally, the vendor does not have specific knowledge about the soil conditions at the site.

8.11.1.1 Foundation mounting

Centrifugal compressors are installed on either soleplates or fabricated steel baseplates. The baseplate may be on the non-self-supporting or self-supporting type, depending on site requirements. These intermediate supports provide a permanent



(A)

(B)

Fig. 8.104 Inducer blades (adjustable guide vanes).

mounting point for the machine feet, which can then be shimmed for final location and alignment. In many cases, the baseplate is extended to support both the driver and driven equipment, and in cases such as offshore installations, it can also contain the lube and seal system. The baseplate simplifies installation.

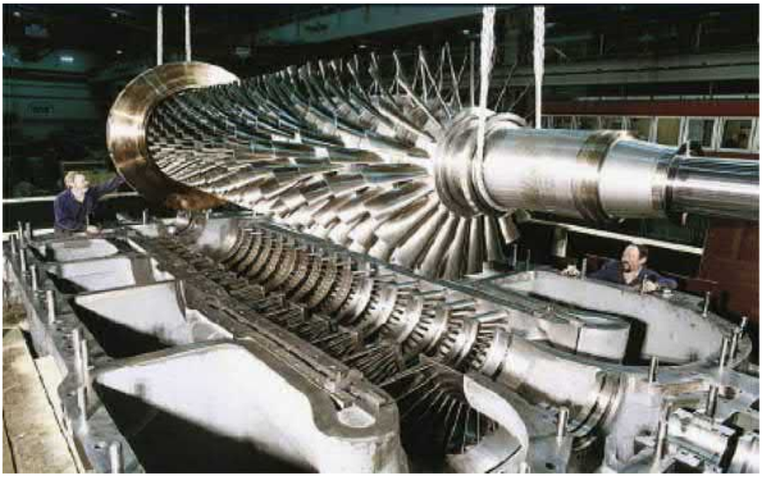


Fig. 8.105 Axial compressor rotor assembly.

8.11.1.2 Design basis for rotating compressors

Dynamic (centrifugal and axial) and rotary compressors generally exert much smaller dynamic forces than reciprocating compressors. Nevertheless, these forces should be accounted for to avoid a potentially serious vibration problem during operation of the compressor. A fault in the design of a concrete foundation is extremely difficult to correct after the concrete has been poured. There is no easy way to add mass, alter the stiffness, or adjust damping to change the natural frequency of a concrete foundation in an effort to move the system away from a condition of resonance. In a few cases, it has been necessary to break out an existing foundation and pour a redesigned foundation to solve a serious vibration problem. Obviously, such instances are exceeding expensive and time consuming.

While guidelines have been developed over the years for the allowable vibration of the foundation itself, criteria for defining the forces to be used in foundation design have been lacking. A misunderstanding between the foundation designer and the compressor manufacturer regarding the unbalanced forces to be allowed for in the design has contributed to many foundation vibration problems. These problems have commonly been caused by not designing for the actual dynamic forces, but rather for some lower value, due to communication problems between the foundation designer and the machine manufacturer.

Depending on how the question about unbalanced force is asked, the manufacturer might respond with the rotor's residual unbalance from the dynamic balancing machine. This balancing machine tolerance is an extremely small number which might be only 1/20th of the actual force at rated speed. At other times, arbitrary values are assumed for foundations design, yet they may not be representative of actual machine operation.

8.11.2 Dynamic forces

The dynamic force generated by the rotor(s) of rotary and dynamic compressors is related to the running speed and the vibration of the rotor. Because of the complexity of the subject, it is impossible to accurately predict the behavior of a rotor system with one or two simple equations.

Fortunately, standards have been developed for allowable limits of vibration for new machinery. One of the most widely used standards is the API for dynamic and rotary machines:

$$A_v = 2,\ \text{or} \left[\frac{12000}{N}\right]^{\frac{1}{2}}, \text{whichever is less} \tag{8.20}$$

where:
A_V = Peak-to-peak amplitude (displacement) of vibration in mild (0.001 inches)
N = Rated speed in RPM

Note: This equation is valid for speeds down to about 3000 RPM. Below 3000 RPM the limit is 0.002 inches (2 mils).

The following equation may be used for calculating the force used in foundation design. This equation is based on a vibration three times the amplitude calculated from Eq. (8.20). A safety factor of three is recommended because that is about the maximum vibration level where one would ever allow a compressor to continue to operate.

$$F = 4.3 \times 10^{-8} N^2 W_R A_v \tag{8.21}$$

where:
F = Dynamic force, lbs
N = RPM
W_R = weight of rotor, lbs

The force calculated is actually a rotating vector, and it should be assumed that it is acting perpendicularly at the center of the rotor. It should also be assumed that there will be a 50% reaction at each bearing from the unbalanced rotating force. The reactions at the machine's hold-down bolts can then be resolved.

Fig. 8.106 shows the resolution of these forces to bearing reactions. The latter reactions are transmitted to the foundation via soleplate or baseplate and anchor bolts. Note Eq. (8.21) can also be applied to the rotors of turbine drivers and gearboxes.

Occasionally the foundation designer may want to add a factor above the dynamic force determined by Eq. (8.21), although Eq. (8.21) is quite conservative. Five times the API vibration limit has been used as a design criterion in some cases, where there were special concerns about the design. This would provide a safety factor of 1.67 beyond Eq. (8.21). To make the calculation, substitute 7.1 for 4.3 in Eq. (8.21).

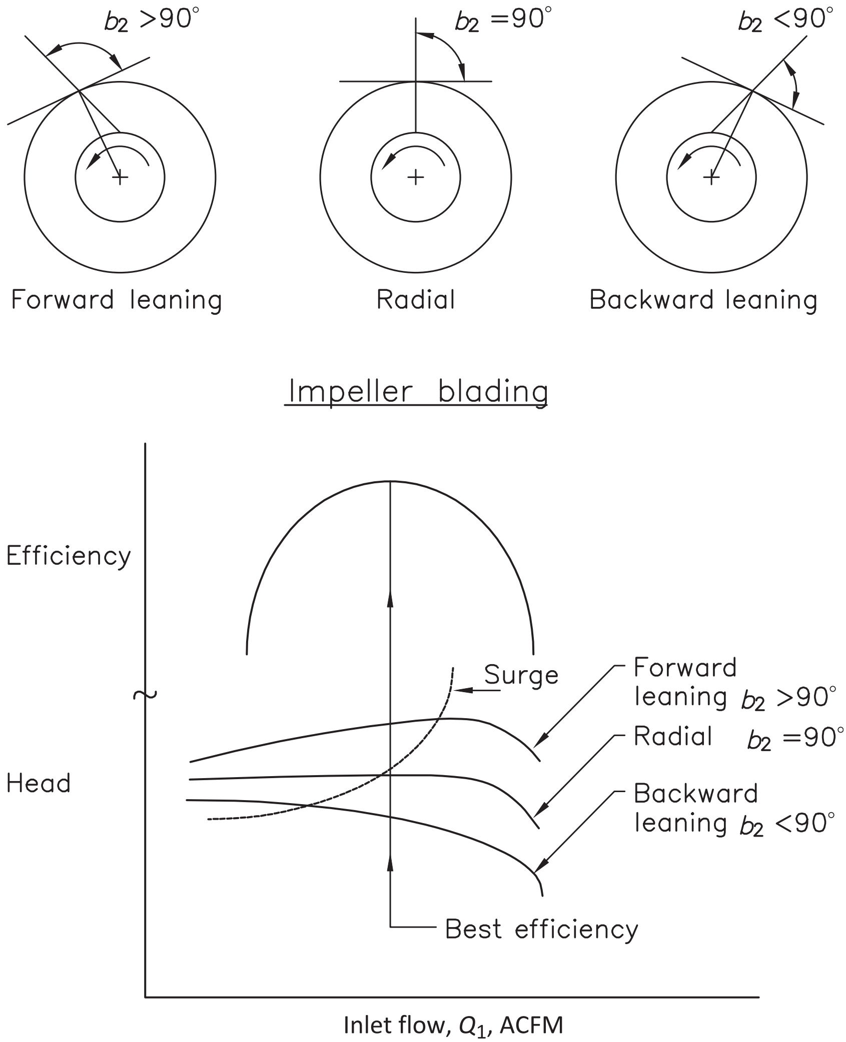


Fig. 8.106 Unbalanced forces from compressor and turbine rotors.

8.11.3 Other considerations

The question sometimes arises about whether the foundation would arrive if a large chunk of metal, such as a piece of an impeller or turbine blade(s), were thrown off the rotor while running at full speed. A second question might be whether the foundation should be designed to accommodate such occurrence. Foundations usually will survive such incidents, although some repairs to anchor bolts, hold-down bolts, or bearing pedestals may be necessary. Generally, such occurrences are not taken into account in the design. The forces involved are extremely high and it is impossible to predict their magnitude. It is suggested that bolting and structures be checked for adequacy at 10 times rated torque. This value is often used on turbine generator foundations, because a short can cause an instantaneous torque increase to that level. Similarly, a compressor rotor might cause such a torque increase in the event if there is a severe rub. It is recommended that the natural frequency of the foundation system be at least 30% above or below the frequency of any compressor or driver rating speed. As a rule of thumb, the weight of the foundation should be no less than three times the weight of the rotating machinery it supports.

8.12 Centrifugal compressor performance

Example problems 8.5 and 8.6 are based on the performance maps and data included in Figs. 8.107–8.110 for a Solar "Centaur T4500" tandem compressor unit.

Overal tandem	SD-15413	(2 Sheets)
LP unit	SD-15414	(1 Sheets)
HP unit	SD-15415	(1 Sheets)

Tandem gas compressor program P435

Comp	Prefix	DTA	Stages						
C160	107	7.50	2 DT	2 DT	1 DT	1 DT	1 DE		
C160	107	7.50	2 CT	1 CT	1 CT	1 CT	22 BT	2 BT	1 BE

P1, PSIA	255.00	509.49	
P2, PSIA	519.49	1264.99	
H-ISEN, PT-LBF/LBM	34592.	47644.	
Capacity, CFM	801.8	422.3	
STD flow, MMSCFD	20.00	20.00	
Power, HP	947.	1370.	= 2317 EF
Speed, RPM	20868.	20868.	
Pressure ratio	2.037	2.483	
T_1, DEG F	80.0	120.0	
T_2, DEG F	199.1	286.1	
Efficiency, ISEN	0.739	0.703	
Surge margin	0.261	0.232	
Specific gravity	0.6160	0.6160	
K_1	1.284	1.271	
K_2	1.249	1.228	
PCP	673.0	673.0	
PCT	366.0	366.0	
Z_1	0.9649	0.9453	
Z_2	0.9670	0.9605	
K – 50 DEG F	1.294	1.294	
K – 300 DEG F	1.225	1.225	

Fig. 8.107 Summary of overall tandem performance.

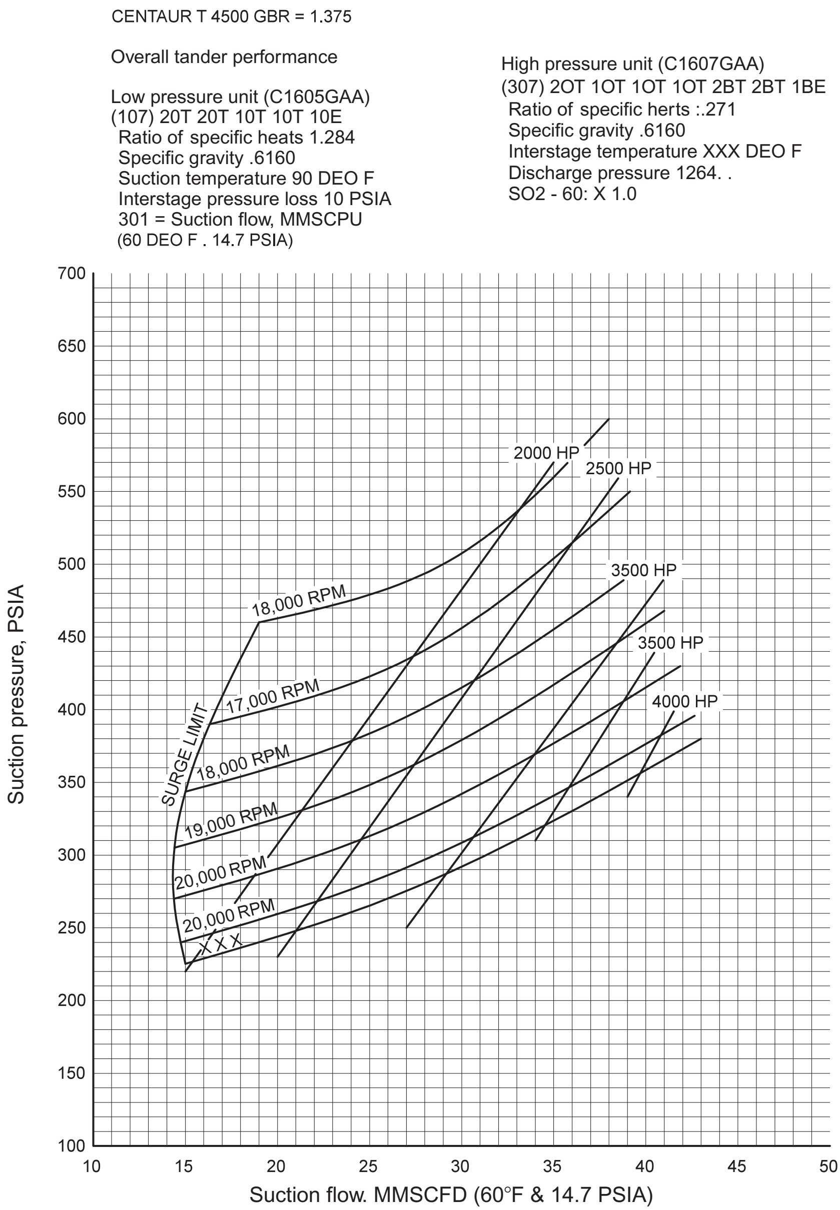


Fig. 8.108 Overall tandem performance map.

The compressor installation includes a gas turbine engine driving two centrifugal compressors. There is a speed-increasing gearbox between the gas turbine and the first compressor body. The second compressor body is directly connected to the first body. Both compressors rotate at the same speed.

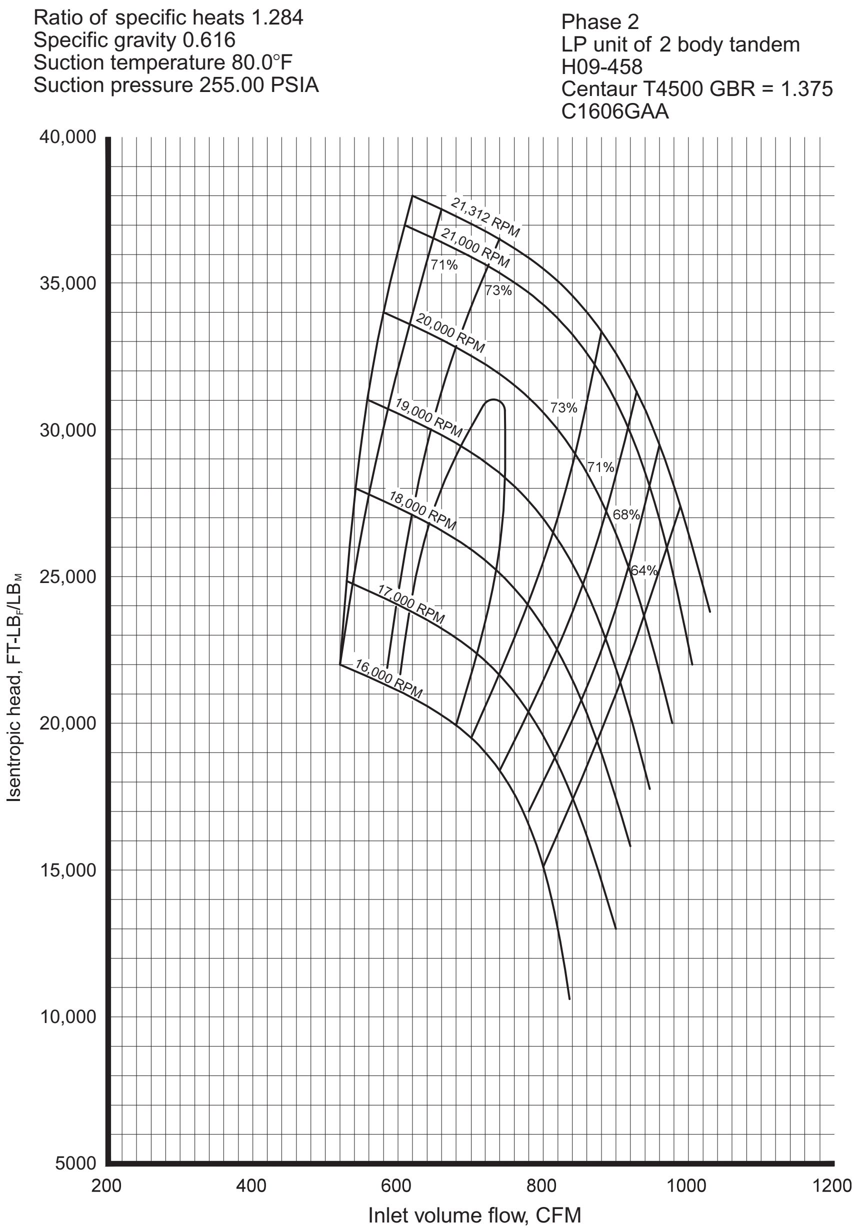


Fig. 8.109 Low-pressure performance map.

The gas enters the LP Unit (first stage) at 80°F (26.67°C). It is compressed, cooled to 120°F (48.88°C), and then passes through a scrubber before entering the HP Unit (second stage). After leaving the second stage the gas is cooled again before exiting the compressor package.

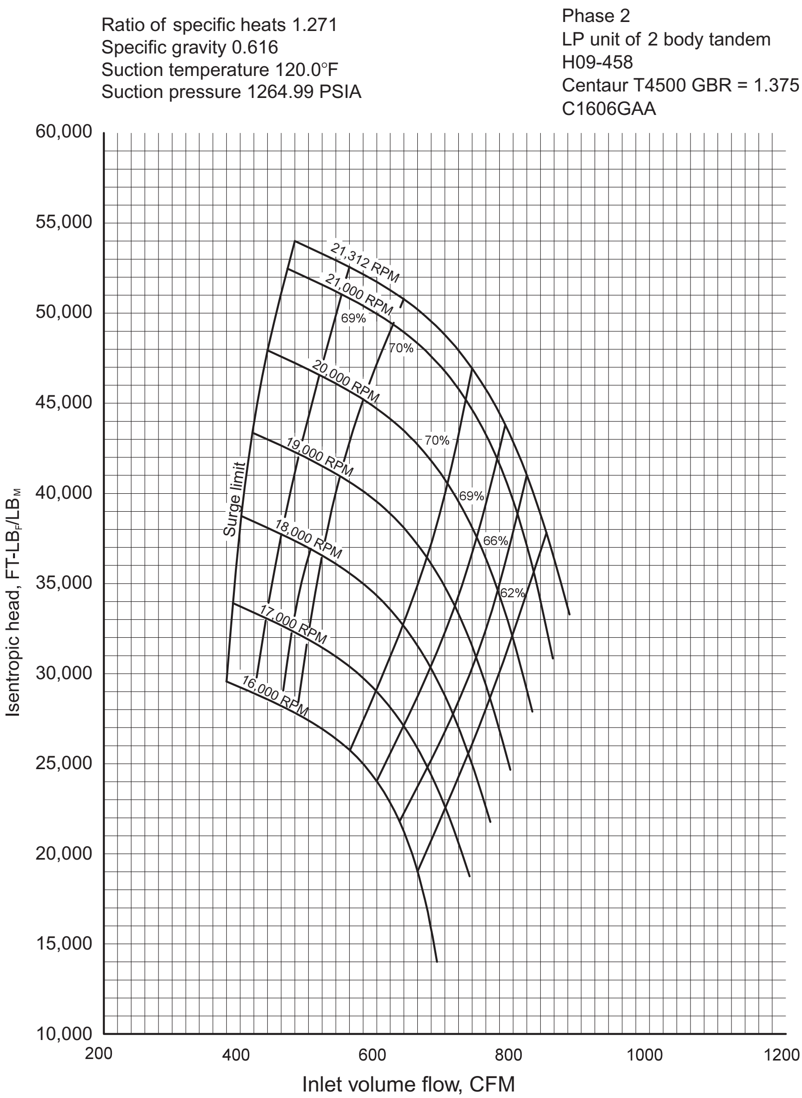


Fig. 8.110 High-pressure unit performance map.

For purposes of this example, it is assumed that no fluids enter or leave the compressor between the LP and HP units. The flow rate and gas composition are the same for both stages.

Gas properties to be used in this problem should be as shown on the data sheet in front of the performance curves. When both suction and discharge properties are shown, the average of the two values should be used for our calculations.

The values calculated by hand will not agree exactly with the curves. Calculation methods, gas properties, rounding, and interpolation will all contribute to differences between the calculated values and the curve data. Values should generally agree within 5%.

Example 8.5. Centrifugal Compressor Performance

Given:

Solar compressor performance maps (Figs. 8.107–8.110).

Determine:

1. Using the overall tandem performance map (Fig. 8.108) for discharge pressure of 1265 psia what are the BHP and RPM required for a flow of 30 MMSCFD and a suction pressure of 400 psia?
2. Calculate the BHP using the "long" equation presented in the "Compressors" section. Assume interstage pressure loss is 10 psi and the following reasonable estimates of gas properties from the data given: $Z=0.96$, $k=1.27$
3. Using the curve for the LP unit, calculate the isentropic compression efficiency for this stage knowing the speed from Question 1. What is the isentropic head? What is the discharge pressure? What is the BHP required for this stage?
4. If the turbine driving the compressor can develop 2000 HP at 16,000 RPM and 2500 HP at 19,000 RPM, what will happen if there is 30 MMSCFD available to compress?

Solution:

1. From the Fig. 8.108, at Q = 30 MMSCFD and P_s =400 psia, read 2500 hp @ 18,600 rpm
2. *Calculate HP*:

Assume R is a constant and calculate pressure for stages

$$R=\left(\frac{P_d}{P_s}\right)^{\frac{1}{n}}=\left(\frac{1265}{400}\right)^{\frac{1}{2}}=1.778$$

Thus *first-stage discharge* = (400) (1.778) = 711 psia

And

$$\text{Second}-\text{stage suction}=711-\Delta P_{\text{stage}}$$
$$=711-10=701$$

General equation for BHP:

$$\text{BHP}=0.0857\left[Z_{av}{}^{\frac{1}{k}}\right]\left[Z_s{}^{\frac{k-1}{k}}\right]\left[\frac{(Q_g)(T_s)}{E_mE_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_p}{P_s}\right)^{\frac{k-1}{kn}}-1\right]$$

Assume an isentropic efficiency of 74% for the first stage and 70% for the second stage, a mechanical efficiency of 95%, and a polytrophic efficiency of 0.8

From the gas properties Z = 0.96 and k = 1.27

First stage:

$$\text{BHP}=0.857\left[\frac{0.96+0.96}{2}\right]^{\frac{1}{1.27}}[0.96]^{\frac{1.27-1}{1.27}}\left[\frac{(30)(80+460)}{(0.74)(0.95)}\right]$$

$$\text{BHP}=\left[\frac{(1.27)(0.8)}{1.27-1}\right]\left[\left(\frac{711}{400}\right)^{\frac{1.27-1}{1.27\times0.8}}-1\right]$$

$$\text{BHP}=1178\ HP$$

Second stage:

$$\mathrm{BHP} = 0.857\left[\frac{0.96+0.96}{2}\right]^{\frac{1}{1.27}}[0.96]^{\frac{1.27-1}{1.27}}\left[\frac{(30)(120+460)}{(0.70)(0.95)}\right]$$
$$\mathrm{BHP} = \left[\frac{(1.27)(0.8)}{1.27-1}\right]\left[\left(\frac{1265}{701}\right)^{\frac{1.27-1}{(1.27)(0.8)}} - 1\right]$$
$$\mathrm{BHP} = 1376$$

Total HP = 1st stage + 2nd stage = 1178 + 1376 = 2554

3. Convert 30 MMSCFD to ACFM

$$\mathrm{ACFM} = 19.6\frac{Q_g Z_s T_s}{P_s}$$
$$\mathrm{ACFM} = 19.6\frac{(30)(0.96)(540)}{300}$$
$$\mathrm{ACFM} = 762$$

From Step 1 we know the *speed is 18,600 rpm*. Entering the low-pressure unit (Fig. 8.109) at 762 ACFM and 18,600 rpm, the isentropic efficiency is read at approximately 73.5%

The isentropic head is approximately 26,000 ft Ib/lb

Calculate the discharge pressure from the formula

$$H_{\mathrm{isen}} = 53.3[Z_{av}]\left[\frac{T_s}{S}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k_n}} - 1\right]$$

Rearranging this equation and solving for P_d

$$P_d = \left[\frac{P_s^{\frac{k-1}{k\eta}} S H_{is}(k-1)}{53.3 Z_{av} T_s k_\eta} + P_s^{\frac{k-1}{k\eta}}\right]^{\frac{k\eta}{k-1}}$$

$$P_d = \left[\frac{(400)^{\frac{1.27-1}{(1.27)(0.8)}}(0.62)(26,000)(1.27-1)}{5.33(0.96)(540)(1.27)(0.8)} + 400^{\frac{1.27-1}{(1.27)(0.8)}}\right]^{\frac{(1.27)(0.8)}{1.27-1}}$$
$$P_d = 688 \text{ psia}$$

Thus the BHP can be determined:

$$BHP = 0.857[0.96]^{\frac{1}{1.27}}[0.96]^{\frac{1.27-1}{1.27}}\left[\frac{(30)(540)}{(0.735)(0.95)}\right]$$

$$BHP = \left[\frac{(1.27)(0.8)}{1.27-1}\right]\left[\left(\frac{688}{400}\right)^{\frac{1.27-1}{(1.27)(0.8)}}-1\right]$$

$$BHP = 1113$$

4. The compressor speed will decrease. If the incoming gas flow rate is constant the suction pressure will increase since at a lower speed the compressor cannot compress as much gas. Assuming a straight line relationship, the compressor will slow to less than 18,000 rpm and the suction pressure will increase to 430 psia

Example 8.6. Centrifugal compressor performance

Given:

Solar compressor performance maps (Figs. 8.107–8.110)

Determine:

1. Use the map labeled "Overall Tandem Performance (Fig. 8.108)".
   Assume that the maximum compressor horsepower is limited to 2500 HP due to site ambient conditions. For a suction pressure of 300 psia and maximum horsepower, what is the flow rate through the compressor in MMSCFD?
2. Calculate the Inlet Volume Flow (ACFM) to LP Unit (first stage).
   Using the map for the LP Unit (Fig. 8.109), determine the isentropic head, knowing the speed from Question 1.
   What is the isentropic compressor efficiency at this same condition?
3. Using the equation(s) in the lecture notes, calculate the discharge pressure, discharge temperature, and the BHP for the LP Unit. Assume a mechanical efficiency of 98% and a polytrophic efficiency of 80%.
4. As a check of the calculations performed in steps 2 and 3, use the map for the HP Unit (second stage) (Fig. 8.110) and determine the same performance variables as for the LP Unit. Assume a 10-psi pressure drop from the discharge of the LP Unit to the suction of the HP Unit.
5. Compare the calculated discharge pressure of the HP Unit to the design point of 1265 psia. Compare the sum of the horsepower for the LP and HP units to the 2500-HP design point.

Solution:

1. From Fig. 8.108, the flow rate is 24.2 MMSCFD @ 20,200 rpm
2. Convert 24.2 MMSCFD to ACFM

$$ACFM = 19.6\frac{Q_g Z_s T_s}{P_s}$$

$$ACFM = 19.6\frac{(24.2)(0.9649)(540)}{300}$$

$$ACFM = 824$$

From Step 1 we know the rpm is 20,200 rpm. Entering the LP unit map (Fig. 8.109) at 824 ACFM and 20,200 rpm, the isentropic efficiency is approximately 73.5%

The isentropic head is approximately 31,000 ft-Ib/lb.

3. Calculate the discharge pressure from the formula:

$$H_{\text{isen}} = 53.3[Z_{av}]\left[\frac{T_s}{S}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k_n}} - 1\right]$$

Rearrange this equation and solve for P_d

$$P_d = \left[\frac{P_s^{\frac{k-1}{k}} SH_{is}(k-1)}{53.3Z_{av}T_sk} + P_s^{\frac{k-1}{k}}\right]^{\frac{k}{k-1}}$$

Determine Z_{AV}:

$$Z_{\text{AV}} = \frac{0.9649 + 0.9670}{2} = 0.9660$$

Solving for P_d:

$$P_d = \left[\frac{(300)^{\frac{1.27-1}{(1.267)}}(0.616)(31,000)(1.267-1)}{5.33(0.966)(540)(1.267)} + 300^{\frac{1.27-1}{(1.267)}}\right]^{\frac{(1.267)}{1.267-1}}$$

$$P_d = 517 \text{ psia}$$

Calculate the brake horsepower from the formula:

$$\text{BHP} = 0.0857\left[Z_{av}^{\frac{1}{k}}\right]\left[(Z_s)^{\frac{k-1}{k}}\right]\left[\frac{(Q_g)(T_s)}{E_mE_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_p}{P_s}\right)^{\frac{k-1}{kn}} - 1\right]$$

Substituting:

$$\text{BHP} = 0.857[0.966]^{\frac{1}{1.267}}[0.9649]^{\frac{1.267-1}{1.267}}\left[\frac{(24.2)(540)}{(0.735)(0.98)}\right]$$

$$\text{BHP} = \left[\frac{(1.267)}{1.267-1}\right]\left[\left(\frac{571}{300}\right)^{\frac{1.27-1}{(1.267)}} - 1\right]$$

$$\text{BHP} = 1035$$

Calculate the discharge temperature from the formula:

$$T_d = T_s \left[\left(\frac{P_d}{P_s} \right)^{\frac{k-1}{kn}} \right]$$

Assume $n = 0.8$

$$T_d = 540 \left[\left(\frac{571}{300} \right)^{\frac{1.267-1}{(1.267)(0.8)}} \right]$$

$$T_d = 690^{\circ}\mathrm{R} = 180^{\circ}\mathrm{F}$$

4. Convert 24.2 MMSCFD to ACFM at the HP unit inlet:

$$\mathrm{ACFM} = 19.6 \frac{Q_g Z_s T_s}{P_s}$$

$$P_s = 571 - 10 = 561 \text{ psi}$$

$$\mathrm{ACFM} = 19.6 \frac{(24.2)(0.9453)(580)}{561}$$

$$\mathrm{ACFM} = 464$$

From Step 1 we know the rpm is 20,200 rpm. Entering the HP unit map (Fig. 8.110) at 464 ACFM and 20,200 rpm, the isentropic efficiency is approximately 69.8%.

The isentropic head is approximately 41,000 ft-Ib/lb.

Calculate the discharge pressure from the formula:

$$P_d = P_d = \left[\frac{P_s^{\frac{k-1}{k}} SH_{is}(k-1)}{53.3 Z_{av} T_s kn} + P_s^{\frac{k-1}{k}} \right]^{\frac{k}{k-1}}$$

Determine Z_{AV}:

$$Z_{\mathrm{av}} = \frac{0.9453 + 0.9605}{2} = 0.9529$$

Determine k_{AV}:

$$K = \frac{1.271 + 1.228}{2} = 1.250$$

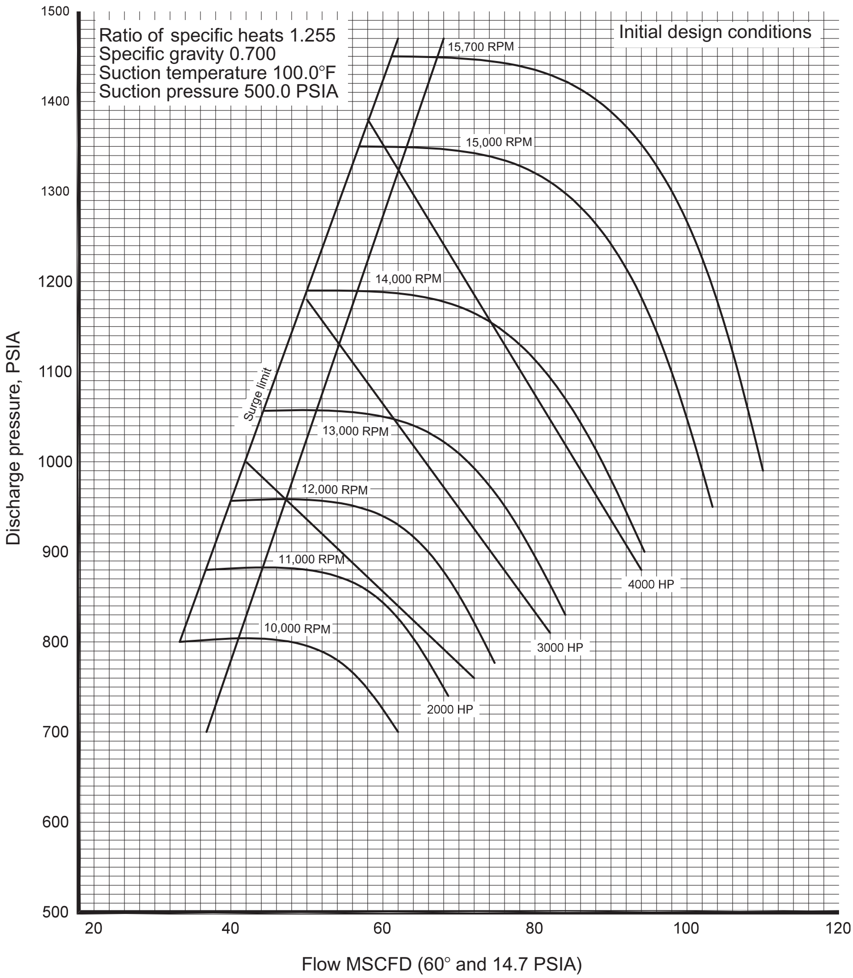


Fig. 8.111 Dimensional performance map.

Substituting and solving for P_d:

$$P_d = \left[\frac{(651)^{\frac{1.25-1}{(1.25)}}(0.616)(41,000)(1.25-1)}{5.33(0.9529)(580)(1.25)} + 561^{\frac{1.25-1}{(1.25)}}\right]^{\frac{(1.25)}{1.25-1}}$$

$$P_d = 1238 \text{ psia}$$

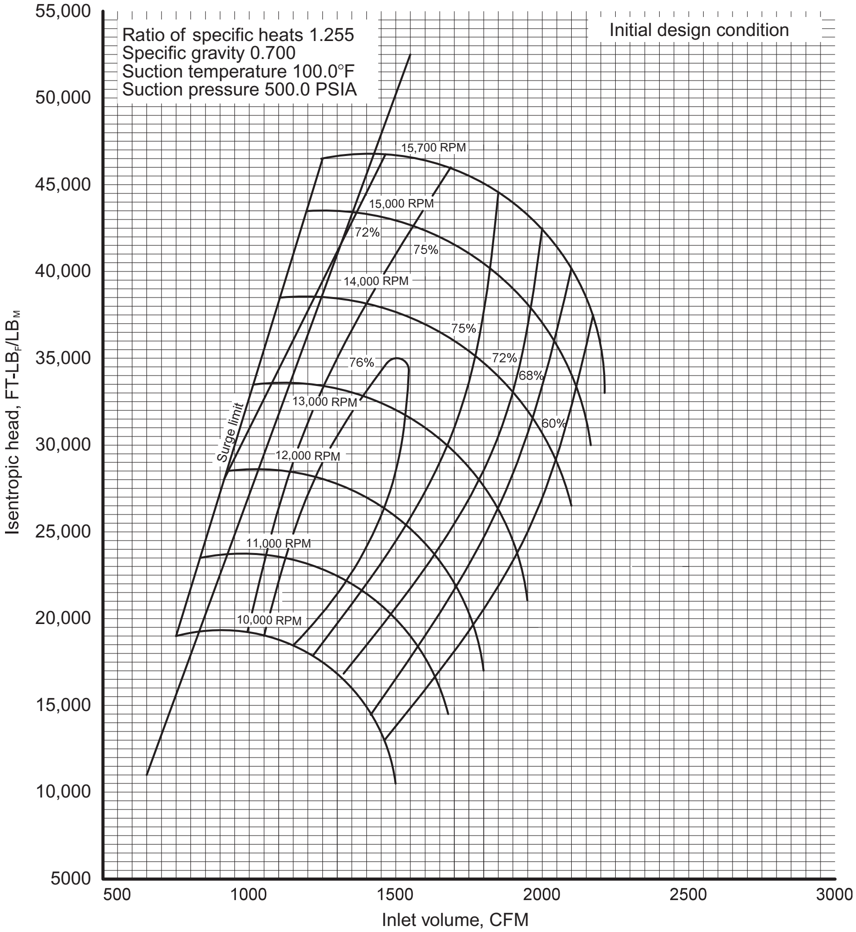


Fig. 8.112 Head vs. capacity performance map.

Calculate the horsepower:

$$\mathrm{BHP} = 0.0857\left[Z_{av}^{\frac{1}{k}}\right]\left[(Z_s)^{\frac{k-1}{k}}\right]\left[\frac{(Q_g)(T_s)}{E_m E_a}\right]\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_p}{P_s}\right)^{\frac{k-1}{kn}}-1\right]$$

$$\mathrm{BHP} = 0.857[0.9529]^{\frac{1}{1.25}}[0.9529]^{\frac{1.25-1}{1.25}}\left[\frac{(24.2)(580)}{(0.698)(0.98)}\right]$$

$$\mathrm{BHP} = \left[\frac{(1.25)}{1.25-1}\right]\left[\left(\frac{1238}{561}\right)^{\frac{1.25-1}{(1.25)}}-1\right]$$

$$\mathrm{BHP} = 1435$$

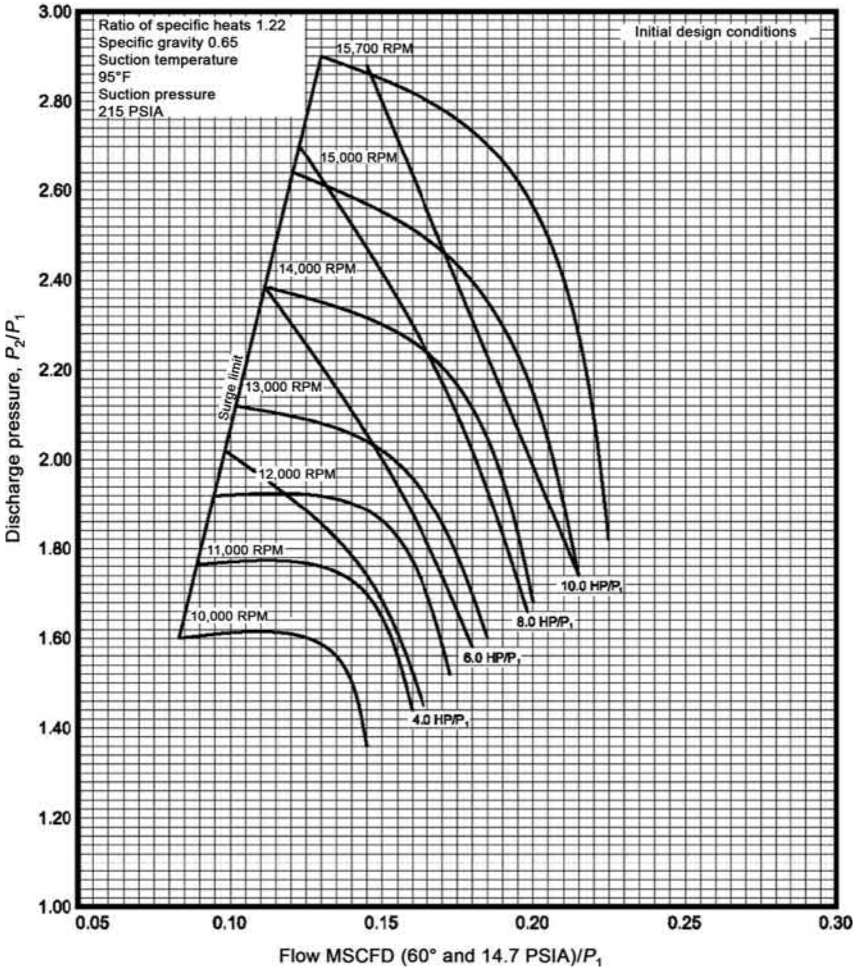


Fig. 8.113 Semidimensional performance map.

Calculate the discharge temperature:

$$T_d = 540\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k\eta}}\right]$$

$$T_d = (580)\left[\left(\frac{1238}{561}\right)^{\frac{(1.25-1)}{(1.25)(0.8)}}\right] = 707^{\circ}\mathrm{R} = 247^{\circ}\mathrm{F}$$

5. Comparison or results:

Calculated discharge pressure = 1238 psia

$$(100)\left(\frac{1265-1238}{1265}\right)=2.1\%$$

Total horsepower = 1035 + 1435 = 2470

$$100\left(\frac{2500-2470}{2500}\right)=1.2\%$$

8.13 Exercises

1. The vertically split centrifugal compressor___________________________
   a. Is bolted together along the inlet and outlet nozzles
   b. Is split along the rotor assembly
   c. Must have the rotor assembly removed for inspection and cleaning
   d. Has very poor high pressure capability
   e. c and d

2. List the common applications for centrifugal compressors in production operations.
   a. ____
   b. ____
   c. ____

3. Match the following terms:

_____ impeller	a. Composed of the shaft and impellers
_____ rotor	b. turn and direct gas flow through the compressor
_____ guide vanes	c. The basis for compressor performance (i.e., affects the flow of gas)
_____ stage	d. Consists of the impeller, inlet guide vanes, and diaphragm

4. Match the following terms:

______ pressure ratio	a. The point at which the compressor is at minimum head and maximum flow
______ head	b. Energy added to the gas
______ isentropic efficiency	c. A condition of unstable flow
______ ACFM	d. Comparison of inlet and outlet pressures

Continued

______ surge.	e. Flow through the compressor varies directly with the speed
______ stonewall	f. The amount of volume throughput per unit of time at inlet conditions
______ affinity	g. The ratio of the head produced and head required after internal losses

5. Leakage of gas from the compressor case and leakage of seal oil into the compressor case is prevented by_______________________________
   a. Maintaining the buffer gas at a pressure above the compressor suction pressure
   b. Maintaining seal oil pressure above buffer gas pressure
   c. Maintaining a stable suction pressure above the seal oil pressure
   d. s and b
   e. b and c
6. The most commonly used performance map is the _________________
   a. Dimensional
   b. Semidimensional
   c. Thermodynamic
   d. Enthalpy vs. Polytrophic
   e. Head vs. Capacity
7. The basic cause of surge in a centrifugal compressor is due to _____________________________
   a. Poor suction design
   b. Impeller boundary layer separation
   c. Envelope collapse
   d. Bypass valve stuck in the open position
   e. Contaminated seal oil
8. The most common method to alleviate a surge condition is to ______________________________
   a. Decrease the rpm
   b. Shut down and restart
   c. Open the bypass valve
   d. Operate the compressor to the left of the surge limit line
   e. C and D only
9. Continued operation of a compressor in surge will__________________
   a. Not affect the discharge pressure
   b. Result in mechanical failure
   c. Produce low discharge temperatures
   d. Does not affect the compressor overall efficiency
   e. B and C only

Questions 10–13

A centrifugal compressor (98% efficiency) is used on an offshore platform to compress 30 Mscf/D of methane at 1200 psig. The suction pressure is 800 psig, the discharge temperature is 165°F, the suction temperature is 85°F, and the ambient temperature is 60°F. Given the following gas properties,

$Z_{\text{AVG}} = 0.96$
$Y = 0.63$

10. What is the isentropic (adiabatic) efficiency of the compressor?
- **a.** 45%
- **b.** 52%
- **c.** 67%
- **d.** 77%
- **e.** 89%

11. What is the head required?
- **a.** 18,158ft-lb$_r$/lb$_m$
- **b.** 27,158ft-lb$_r$/lb$_m$
- **c.** 38,612fMb$_r$/lb$_m$
- **d.** 57,691 ft-lb$_r$/lb$_m$

12. What is the mass flow through the compressor?
- **a.** 172.6 lb$_m$/min
- **b.** 493.8 lb$_m$/min
- **c.** 791.4 lb$_m$/min
- **d.** 1003.4 lb$_m$/min
- **e.** 1206.5 lb$_m$/min

13. What is the horsepower required?
- **a.** 398 BHP
- **b.** 841 BHP
- **c.** 999 BHP
- **d.** 1293 BHP
- **e.** 1705 BHP

14. Using Fig. 8.111, what will be the required horsepower and speed for the centrifugal compressor with a discharge pressure of 1000 psia and a flow of 50 Mscf/D?

	BHP	**rpm**
a.	200	13,000
b.	550	14,350
c.	1200	16,750
d.	1850	14,500
e.	2250	12,350

15. Using Fig. 8.112, what will be the speed and head required for the compressor if the inlet volume flow is 2000 ft^3/min and the efficiency is 70 percent?

	Head	rpm
a.	16,000 ft-lbr/lbm	13,000
b.	32,000 ft-lbr/lbm	14,800
c.	29,000 ft-lbr/lbm	12,500
d.	40,000 ft-lbr/lbm	15,500
e.	22,000 ft-lbr/lbm	13,500

Questions 16 and 17

A gas turbine/centrifugal compressor package is to be installed at a producing location and will operate at 200 psig (suction pressure) when production is 32 Mscf/D. The compressor manufacturer proposes a unit which operates according to the performance map in Fig. 8.113. The compressor has an adiabatic efficiency of 81 percent and a mechanical efficiency of 98 percent. The compressor is driven by a 2000-horsepower gas turbine. The following conditions were provided by the reservoir engineer:

Suction gas temperature = 95°F
Gas specific gravity = 0.65
Average Compressibility Factor = 0.988
k-value = 1.22
Atmospheric pressure = 14.7 psia

16. What is the expected discharge pressure that the compressor will deliver for these conditions?
  1. 356 psig
  2. 491 psig
  3. 552 psig
  4. 703 psig
  5. 1067 psig

17. After accepting the proposed unit from the vendor, you are informed by the reservoir engineer that the separator pressure will be 75 psig instead of 200 psig. Is the horsepower provided adequate for the flow rate and the expected discharge pressure? What will be the new flow capability of the compressor?
  a. Yes, 35 Mscf/D
  b. Yes, 19.2 Mscf/D
  c. Yes, 12.51 Mscf/D
  d. No, 11.13 Mscf/D
  e. No, 15.54 Mscf/D

18. A centrifugal compressor will be used on an offshore platform to compress natural gas. The suction pressure is 275 psig, the discharge pressure is 1000 psig, and the suction temperature is 86°F. Given the following gas properties, $Z_{AVG} = 0.93$, $S = 0.65$, $k = 1.27$, np = 0.8, determine the following:
  a. Compression ratio
  b. Discharge temperature

c. Isentropic Head

d. Brake Horsepower using the "quick look" BHP formula.

19. The compressor in problem 23 has been designed to deliver the isentropic head as calculated. Upon start-up, the actual suction conditions are as follows:

$S = 0.6$
$T_1 = 104°F$
$k = 1.3$
$Z_{AVG} = 0.94$

Calculate the actual discharge pressure which will be achieved. Assume a constant speed driver.

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Reciprocating compressors

9

Nomenclature

λ	acoustic wavelength, m (ft)
a	manufacturer's recommended valve losses
A	the total product of the actual lift in service and the valve opening for all inlet valves per cylinder, mm^2 (in^2)
a_{m2x}	vertical acceleration of m_2, mm/s^2 (in/s^2)
a_{m2y}	horizontal acceleration of m_2, mm/s^2 (in/s^2)
a_{m3x}	horizontal acceleration of m_3, mm/s^2 (in/s^2)
a_{mlx}	vertical acceleration of m_1, mm/sec^2 (in/s^2)
a_{mly}	horizontal acceleration of m_1, mm/s^2 (in/s^2)
a_p	cross-sectional area of piston, mm^2 (in^2)
a_r	cross-sectional area of rod, mm^2 (in^2)
BHP	brake hp., kW (hp)
C_{CE}	crank-end clearance, mm^3 (in^3)
C_{HE}	head-end clearance, mm^3 (in^3)
C	cylinder clearance, percent of piston displacement
C_p	specific heat at constant pressure
C_v	specific heat at constant volume
d	distance from main bearing to mass center of crank, mm (in)
d_c	diameter of cylinder, mm (in)
d_i	pipe internal diameter, mm (in)
d_r	diameter of rod, mm (in)
E	mechanical efficiency of units, percent
E_a	adiabatic efficiency, percent
E_v	volumetric efficiency, percent
f	0.6 to 0.8 for centrifugal compressors
f	1.0 for reciprocating compressors
F	an allowance for interstage pressure drops
f_P	pulsation frequency, cycles/s
k	ratio of gas specific heats, $\frac{C_P}{C_v}$
L	connecting rod length, mm (in)
m_1	rotating mass of the crank acting at d, kg (lb)
m_2	rotating mass of the connecting rod acting at crank radius r, kg (lb)
m_3	mass of piston, piston rod, and crosshead, kg (lb)
MW	molecular weight of gas
N	1,2,3, ….cycles/rev
n	number of stages
N_P	1 for single-acting cylinders, 2 for double-acting cylinders
P	average line pressure, kPa (psig)
P_{cf}	maximum allowable unfiltered peak-to-peak pulsation level expressed as a percentage of average absolute line pressure at the compressor cylinder flange.
PD	piston displacement m^3/h, ACFM
P_d	stage discharge pressure, kPa (psia)

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00009-0

PP	allowable pulsation level (peak to peak), percent of average pressure
Ps	stage suction pressure, kPa (psia)
P_u	pressure in unloaded end, kPa (psia)
q_a	gas flow rate at suction conditions of temperature and pressure, std. m^3/h (ACFM)
Q_g	gas flow rate std. m^3/h (MMSCFD)
q_g	gas flow rate std. m^3/h (SCFM)
R	compression ratio $\left(\frac{P_d}{P_s}\right)$ of a compressor stage
r	crank radius, mm (in)
R_c	compressor speed rps, rpm
$\mathbf{RL_c}$	rod load in compression, kg (lb)
$\mathbf{RL_t}$	rod load in tension, kg (lb)
s	stroke length, mm (in)
$\mathbf{SV_{CE}}$	swept volume, crank end, mm^3 (in^3)
$\mathbf{SV_{HE}}$	swept volume, head end, mm^3 (in^3)
$\mathbf{SV_{HV+CE}}$	swept volume, head end and crank end, mm^3 (in^3)
T	gas temperature, K (°R)
T_d	stage discharge temperature, K (°R)
T_s	stage suction temperature, K (°R)
V_g	average gas velocity, m/s (ft/min)
Z_{av}	average of suction and discharge compressibility
Z_d	compressibility factor at discharge conditions
Z_s	compressibility factor at suction conditions

9.1 Engineering principles

9.1.1 General considerations

Chapter 8 provided general information and fundamental compression equations applicable to all types of compressors. This chapter presents additional equations, charts, and recommendations specifically related to reciprocating compressors. Reciprocating piston compressors can be classified into several types. The machinery presented in this chapter can be applied to the more common crosshead compressors having power ratings from 150 to several thousand horsepower, with speeds ranging from 250 to 1500 RPM.

Fig. 9.1 shows a cross-section of a typical single-cylinder (one crank-throw) reciprocating compressor and includes the basic nomenclature that will be used throughout this chapter.

9.1.2 Compression cycle

Fig. 9.2 shows an ideal indicator diagram, also known as a pressure-volume ($P-V$) diagram, followed by a series of cylinder illustrations depicting piston movement and valve position. The figure shows in diagram form one complete crankshaft revolution which includes a complete compression cycle. The indicator diagram is used to represent a reciprocating compressor's performance. The diagram represents the pressure

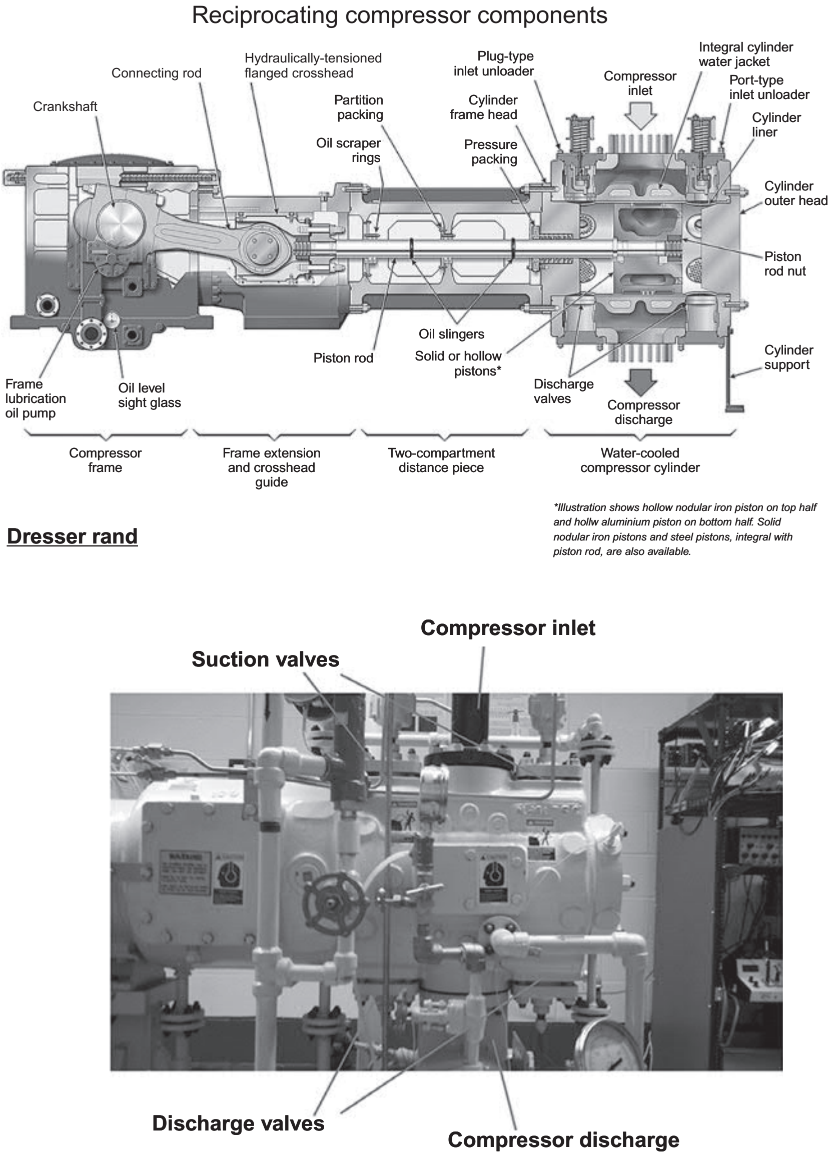


Fig. 9.1 Typical reciprocating compressor components.
Courtesy of Dresser-Rand.

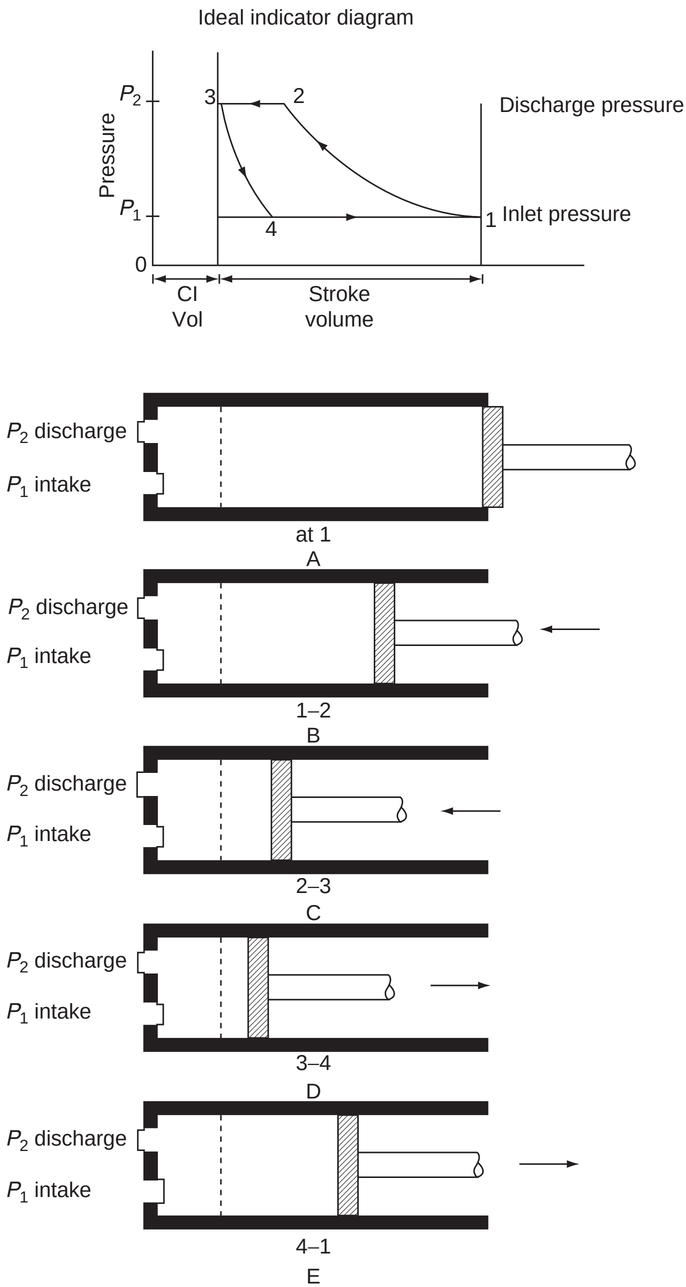


Fig. 9.2 Steps in the cycle of a reciprocating compressor.
Courtesy of Compressors: Sizing and Selection, by Royce Brown, Gulf Publishing Company, Houston, TX, 2005.

in the cylinder of a compressor in relation to cylinder volume. As the piston moves back and forth in the cylinder, the volume of the cylinder changes.

Referring to Fig. 9.2:

Position A: The location where the piston is at the lower end of the stroke (bottom dead center) and is at *Path 1* on the indicator diagram. At this point the cylinder has filled with gas at intake pressure P_1. Note that the valves are both closed.

Position B: At B the piston has started to move to the left. This is the compression portion of the cycle and is shown by *Path 1–2*.

When the piston reaches *Point 2* on the indicator diagram, the exhaust valve starts to open. The discharge portion of the cycle is shown at *Position C*. This is shown on the indicator diagram *Path 2–3*. Note that the discharge valve is open during this period while the intake valve is closed. The gas is discharged at the discharge line pressure P_2.

Position C: When the piston reaches *Point 3* it has traveled to the upper end of its stroke (top dead center). Physically, at this point in the stroke, there is a space between the piston face and the head. The space results in a trapped volume and is called *clearance volume*.

Position D: Next in the cycle, the piston reverses direction and starts the expansion portion of the cycle, as illustrated by Position D in the figure. *Path 3–4* shows this position of the cycle. Here the gas trapped in the clearance volume is reexpanded to the intake pressure. Note that the discharge valve has closed and the intake valve is still closed.

Position E: At *Point 4* the expansion is complete and the intake valve opens. The intake portion of the cycle is shown at Position E. This is indicated by *Path 4–1* on the indicator diagram. The cylinder fills with gas at intake line pressure P_1.

When the piston reaches Point 1 the cycle is complete and starts to repeat.

9.1.3 Volumetric efficiency

The volumetric efficiency (E_V) is defined as the ratio of the compressor's actual capacity to its theoretical capacity. As shown in Fig. 9.3, the distance from *Points A to C* represents the full length of the piston's stroke while the actual intake of the gas to the compressor is indicated by the distance from *Points B to C*. As shown in Fig. 9.4, the distance from points B to C to the distance from A to C indicates volumetric efficiency.

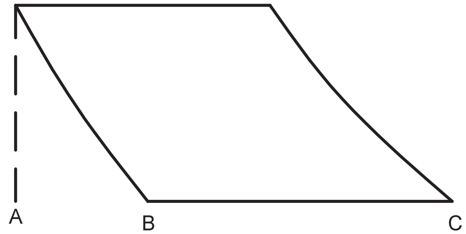


Fig. 9.3 Indicator diagram illustrating Piston Stroke (A to C) and actual intake of the compressor (B to C).

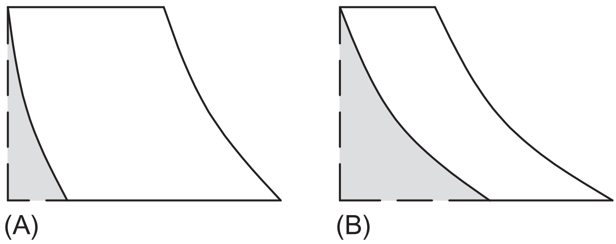


Fig. 9.4 Indicator diagrams illustrating volumetric efficiency. Diagram (A) shows a higher volumetric efficiency than diagram (B).

Volumetric efficiency is an important variable in reciprocating compressor calculations as it affects the diameter, stroke, and speed for a given compressor capacity. As volumetric efficiency decreases:

- Actual capacity decreases.
- Total horsepower required decreases.
- Mechanical efficiency remains essentially the same.
- Horsepower required per MMSCFD remains essentially the same.

The compressor cylinder's actual inlet flow (ICFM) is the product of the volumetric efficiency and the cylinder's displacement over time, expressed in cubic feet per minute (CFM).

Volumetric efficiency is related to the clearance volume in the cylinder (the volume to the left of the dotted line in Fig. 9.2). On the indicator diagram, it is the volume between Points 0 and 3. The total volume displaced by a full stroke of the piston in the single-acting cylinder shown in Fig. 9.2 is the volume between Points 1 and 3. Clearance is usually expressed as a percentage of the displaced volume as follows:

$$C = \frac{V_c}{V_{cyl}}(100) \tag{9.1}$$

where:

C = percent clearance.
V_c = clearance volume, cubic inches
V_{cyl} = displacement volume, cubic inches

The theoretical volumetric efficiency, in percent, of a cylinder is:

$$E_{v(theo.)} = 100 - C\left(r^{\frac{1}{k}} - 1\right) \tag{9.2}$$

where:

R_t = pressure ratio
$= P_2/P_1$ or P_d/P_s
$k = C_p/C_v$

9.1.3.1 Corrections to volumetric efficiency for gas characteristics

Corrections must be made to the volumetric efficiency to account for:

- Valve losses.
- Nonideal reexpansion of gas in the clearance volume.
- Internal leakage.
- Compressibility.
- Other effects.

Each manufacturer has its own set of empirical corrections. Therefore several corrections for volumetric efficiency are used in the industry. Several of these corrections were compared for natural gas (SG = 0.72) and for propane. They are all within about a 5% for pressure ratios between 2 and 4. For hydrogen-rich gases with molecular weights below 10, the corrections varied as much as 17% at a pressure ratio of four. However, the maximum pressure ratio for hydrogen-rich gases is typically about 3 to keep the discharge temperature below 300°F (149°C). At a pressure ratio of 3, the deviation is <10%.

As shown in Fig. 9.5, the volumetric efficiency is the ratio of the volume of gas actually drawn into the cylinder, corrected to suction pressure and temperature, to the volume of piston displacement. Reasons why the volume of gas drawn into the cylinder does not equal the volume of the piston displacement include:

- High clearance and high ratios.
- Heating of the gas during admission to the cylinder.
- Reexpansion of the gas trapped in the clearance space from the previous stroke.

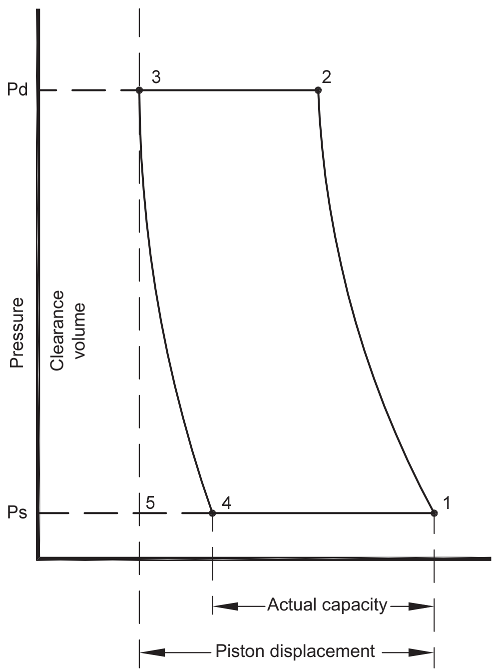


Fig. 9.5 Volumetric efficiency of a reciprocating compressor.

Since the effect of the earlier items cannot be measured, manufacturers disagree on the appropriate equation to use to determine volumetric efficiency. Reexpansion has by far the greatest effect. Volumetric efficiency obtained by any approach will be approximately the same over the range of compression ratios normally used in gas operations.

The following equation yields results that are approximately the average of the other formulas when used for more ordinary lubricated cylinder compressor applications.

$$E_v = 100 - a - R - C\left[\left(R^{\frac{1}{k}}\right)\left(\frac{Z_s}{Z_d}\right) - 1\right] \tag{9.3}$$

where:

a = manufacturer's recommended valve losses (use 4 if unknown), percent
E_v = volumetric efficiency, percent
R = compression ratio (P_d/P_s) of the compressor stage
C = cylinder clearance, percent of piston displacement
Z_s = compressibility factory at suction conditions
Z_d = compressibility factor at discharge conditions
k = ratio of specific heats, $\frac{c_p}{c_v}$
q_a = gas flow rate at suction conditions of temperature and pressure m^3/h (ACFM)
PD = piston displacement, m^3/h (ACFM)

9.1.3.2 Mechanical corrections

Eq. (9.3) applies to lubricated cylinders. Nonlubricated cylinders have a lower volumetric efficiency due to greater piston-to-liner clearances and other factors. For nonlubricated applications, change the "97" in Eq. (9.3) to "94."

Speed also affects volumetric efficiency; Eq. (9.3) applies to slower speed machines up to 600 RPM. For 1000 RPM machines, volumetric efficiency is about 3% less, and the volumetric efficiency determined from Eq. (9.3) should be multiplied by 0.97.

9.1.3.3 Applications and limitations

Eq. (9.3) is not highly accurate. It should only be used for estimating when pressure ratios range from *about 2 to 5*. In addition, the equation assumes the cylinder design provides ample valve flow area, the valve dynamics are satisfactory, and pressure pulsations are moderate. If any one of these conditions is abnormal, the volumetric efficiency prediction is questionable.

Although volumetric efficiency is an important sizing consideration, it does not affect power consumption. Volumetric efficiency is an important factor if the designer is troubleshooting a compressor capacity problem or considering a rerate. Manufacturers may be consulted to determine highly accurate values of volumetric efficiency based on propriety information and programs. Fig. 9.6 provides a "quick look" or approximation for volumetric efficiency for various clearance "C" and compression ratios.

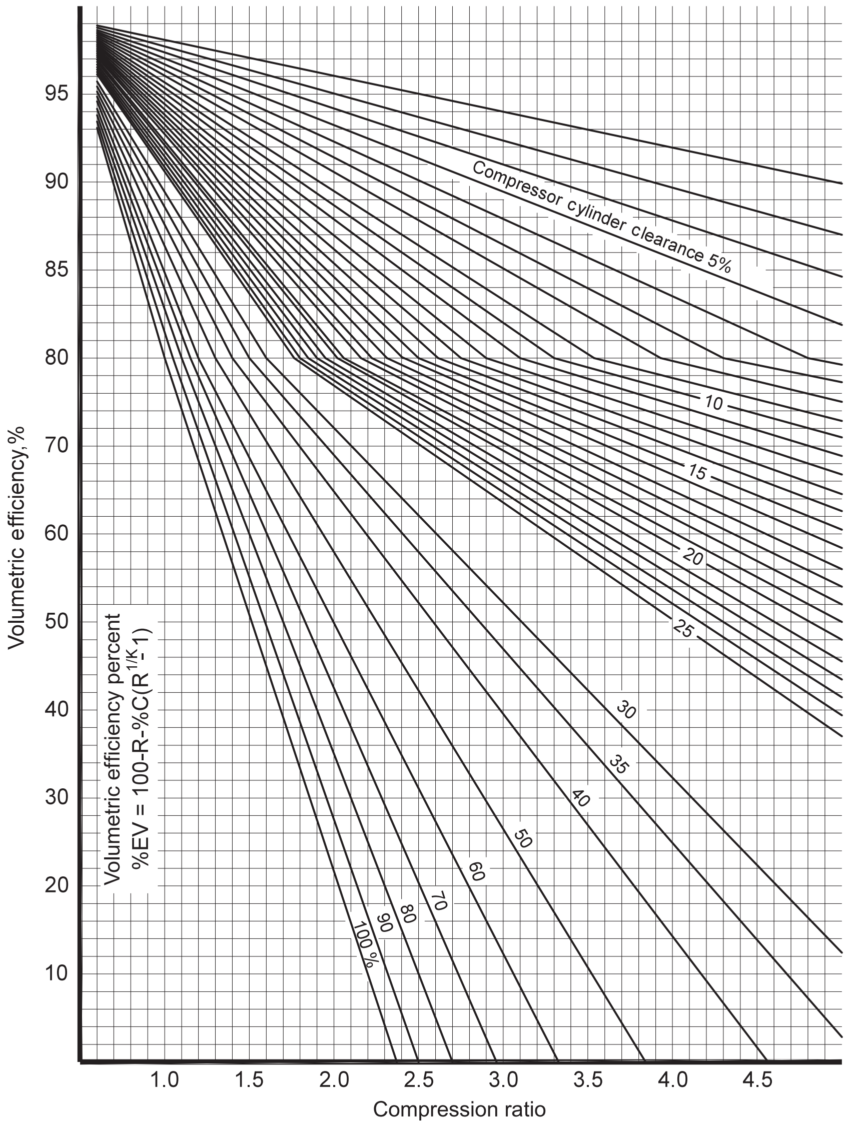


Fig. 9.6 "Quick look" approximations for compressor volumetric efficiency.

9.1.3.4 Actual inlet flow

Actual inlet flow (ICFM) to the cylinders is determined by the equation:

$$\mathrm{ICRM} = Q = V_d E_v \tag{9.4}$$

where:

V_d = displacement rate, CFM
E_V = volumetric efficiency, expressed as a decimal value, not a percent.

9.1.4 Compression ratios

With special materials and cooling and lubrication systems, some reciprocating compressors can have discharge pressures of about 6000 psi (41,368 kPa). This is not common; a more usual discharge limit is around 2000 psi (13,790 kPa). Many reciprocating compressors are designed for maximum discharge pressures of about 1480 psi (10,204 kPa) (the pressure rating of an ASME 600 series fitting at 100°F (37.8°C).

In understanding compressors, it is more important to consider the compressor ratio than the discharge pressure. The overall compressor ratio, R_t, is the ratio of the absolute discharge pressure to the absolute suction pressure and is given by:

$$R_t = \frac{P_d}{P_s} \tag{9.5}$$

where:

R_t = overall compressor ratio
P_d = stage discharge pressure, kPa (psia)
P_s = stage suction pressure, kPa (psia)

The compression ratio per stage, R, is based on absolute pressures and the number of stages required. It is defined as follows:

$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \tag{9.6}$$

where:

n = number of stage

Normal compression ratio limits per stage vary from slightly over 1.0 for booster service to roughly 5.0 for special high ratio machines. Staging compression is efficient because it saves power. Fig. 9.7 shows a simplified pressure–volume curve to illustrate the impact of staging on horsepower.

By installing a second stage, one may cool the gas discharged from the first stage from point A to point D. The actual volume of gas to be compressed is then lower than it would have been if the gas had continued to be compressed in a single-stage compressor without interstage cooling. The area ABCD represents the energy saved by performing the compression in two stages. This saving must be balanced by the cost for increased piping and equipment, energy lost due to pressure drop in interstage piping, and increased maintenance costs. To minimize power usage, one should initially select an equal compression ratio in each stage.

The compression ratio per stage is limited for any compressor by the need to limit the discharge temperature. As the ratio per stage increases, the discharge temperature increases.

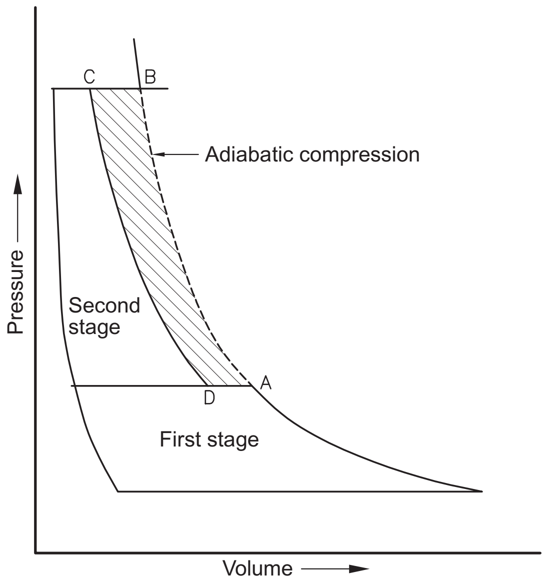


Fig. 9.7 Pressure-volume curve showing the horsepower reduction by multi-staging.

9.1.5 Discharge temperature

The discharge temperature for any single stage of compression can be calculated from:

$$T_d = T_s \left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} \tag{9.7}$$

where:

T_d = stage discharge temperature, K(°R)
T_s = stage suction temperature, K (°R)
P_d = stage discharge pressure kPa (psia)
P_s = stage suction pressure, kPa (psia)
k = ratio of specific heats, $\frac{c_p}{c_v}$
c_p = specific heat at constant pressure
c_v = specific heat at constant volume

It is desirable to limit discharge temperatures to below 250°F to 275°F (121°C to 135°C) to assure adequate packing life and to avoid lube oil degradation. At temperatures above 300°F (149°C), eventual lube oil degradation is likely, and if oxygen is present, ignition is possible. Under no circumstances should the discharge temperature be allowed to exceed 350°F (177°C).

The discharge temperature can be lowered by

- Cooling the suction gas.
- Reducing the compression ratios per stage by adding more stages of compression.

Reciprocating compression follows lines of constant entropy (isentropic). Therefore there is no polytropic factor in the temperature and power equations. This is not the case for centrifugal compressors.

Fig. 9.8 provides a "quick look" approximation for determining the ratio of specific heats (k). Fig. 9.9 provides a "quick look" approximation for determining the discharge temperature for various values of k and R.

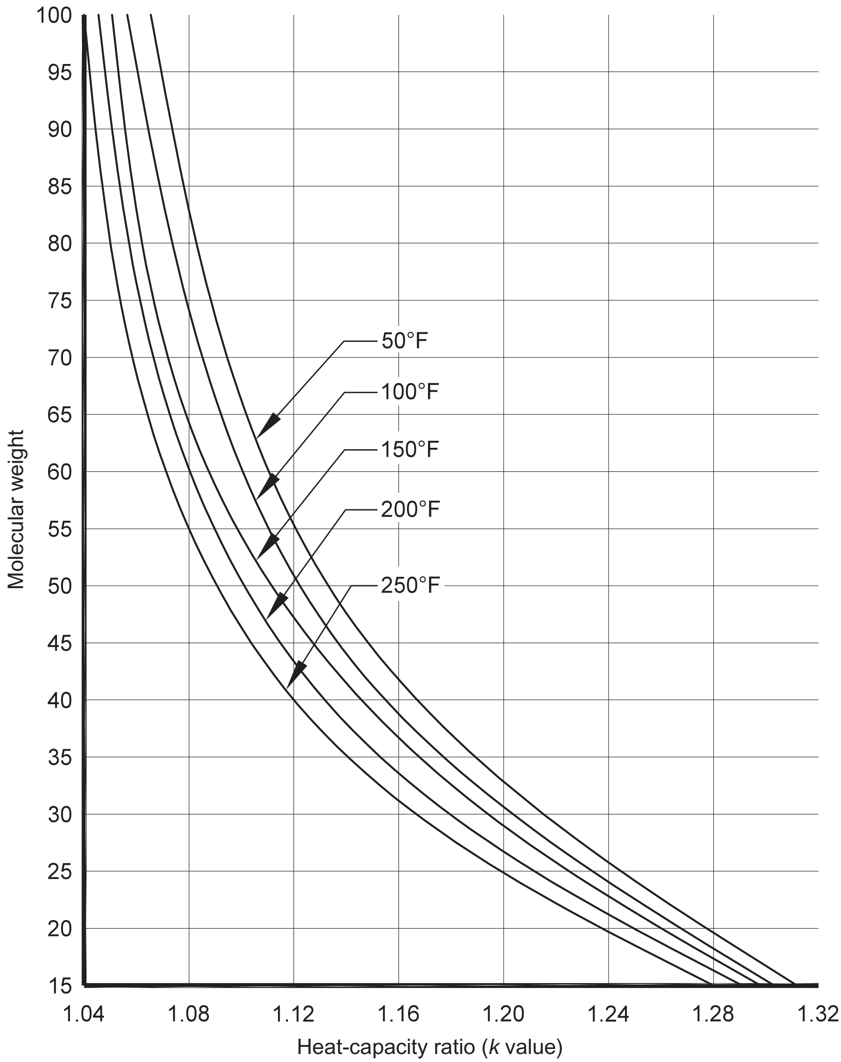


Fig. 9.8 "Quick look" approximation for determining the heat capacity or ratio of specific heats (k-value).

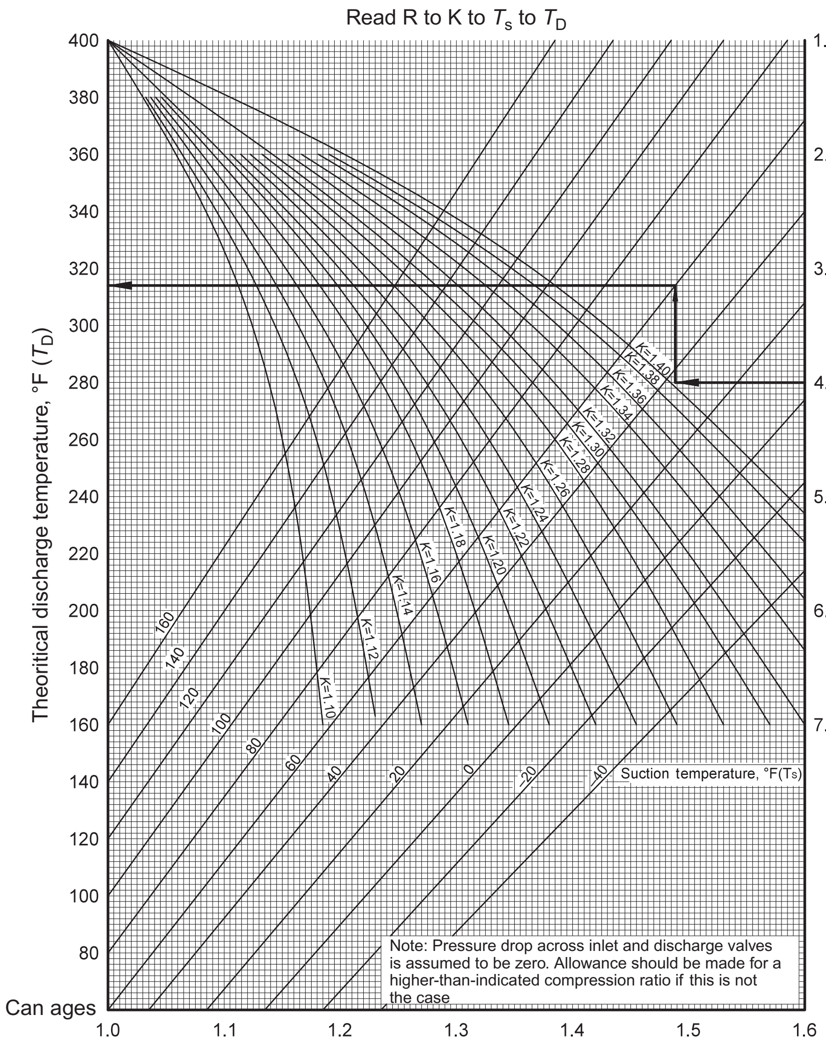


Fig. 9.9 "Quick look" approximation for determining the gas discharge temperature.

9.1.6 Capacity

Using a known piston displacement and efficiency, the gas throughput can be calculated from:

$$q_a = E_v \times PD \tag{9.8}$$

where:

q_a = gas flow rate at suction conditioning of temperature and pressure, m^3/h (ACFM(Actual ft^3/min))

E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

Displacement calculations are normally expressed in actual cubic feet per minute. However, process designers tend to express gas flow rates related to standard conditions of temperature and pressure. Common units are standard cubic meters per hour and standard cubic feet per minute (SCFM) or millions of standard cubic feet per day (MMSCFD). Since gas is very compressible, the flow rate expressed as volume per unit time at actual conditions of pressure and temperature varies considerably from the flow rate in volume per unit time if the gas were at standard conditions.

In this text, standard conditions are referenced to 14.7 psia (101.35 kPa) and 60°F, or 520°R (15.56°C, or 288.9K). Several variations of standard conditions exist in the literature for specific equations, tables, and charts. The differences, however, are minor and can often be neglected for the types of engineering calculations needed by the project engineer.

Eq. (9.9) converts the flow rate, q_a, from ACFM to SCFM. It is easily derived using the ideal gas law.

Field unit

$$q_g = 35.4\frac{q_a P_s}{T_s Z_s} \tag{9.9a}$$

Metric unit

$$q_g = 2.85\frac{q_a P_s}{T_s Z_s} \tag{9.9b}$$

where:

q_g = gas flow rate at standard conditions of temperature and pressure stdm^3/h (SCFM)
P_s = stage suction pressure, kPa (psia)
T_s = stage suction temperature, K (^{O}R)
Z_s = compressibility factor at suction condition

Field unit

$$Q_g = 0.051\frac{q_a P_s}{T_s Z_s} \tag{9.10a}$$

Metric unit

$$Q_g = 2.85\frac{q_a P_s}{T_s Z_s} \tag{9.10b}$$

where;

Q_g = gas flow rate, stdm^3/h (MMSCFD)

9.1.7 Piston displacement

Piston displacement is the actual volume of the cylinder swept by the piston per minute; it is a function of rpm, cylinder internal diameter, and stroke length. Cylinders can be either single acting or double acting. A single-acting cylinder can compress gas on the head end of the piston (the end of the cylinder away from the compressor frame) or on the crank end of the piston (the end of the cylinder nearest the crank). A double-acting cylinder compresses gas on the head end when it is moving toward the head and on the crank end when it is moving toward the crank.

Piston displacement can be calculated as follows:

Single-acting cylinder (head-end displacement)

Field unit

$$PD = \frac{\left(d_c^2\right)(s)(R_c)}{2200} \tag{9.11a}$$

Metric unit

$$PD = \frac{d_c^2 s R_c}{353{,}650} \tag{9.11b}$$

Single-acting cylinder (crank-end displacement)

Field unit

$$PD = \frac{\left(d_c^2 - d_r^2\right)(s)(R_c)}{2200} \tag{9.12a}$$

Metric unit

$$PD = \frac{\left(d_c^2 - d_r^2\right) s R_c}{353{,}650} \tag{9.12b}$$

where:

PD = piston displacement, m^3/h (ACFM)
s = stroke length, (in)
R_c = compressor speed, rps (revolutions per minute)
d_c = diameter of cylinder, mm (in)
d_r = diameter of rod, mm (in)

Double-acting cylinder (sum of head-end and crank-end displacement)

Field unit

$$PD = \frac{\left[2(d_c)^2 - (d_r)^2\right](s)(R_c)}{2200} \tag{9.13a}$$

Metric unit

$$PD = \frac{\left[2(d_c)^2 - (d_r)^2\right] s R_c}{353{,}650} \tag{9.13b}$$

9.1.8 Compressor clearance

Clearance is the volume remaining in a cylinder when the piston is at the end of its stroke. This is the sum of the volume between the head of the cylinder and the piston, the volume between the crank end and the cylinder, and the volume under the valve seats. The total clearance is expressed in percent of the total piston displacement and is normally between 4% and 20%.

As the piston starts its suction stroke, the gas that remains in the cylinder in the clearance areas expands until the pressure in the cylinder is equal to the pressure in the line outside the cylinder. The greater the clearance, the longer it takes for the suction valves to open and the lower the actual volume of new gas that enters the cylinder.

Adding clearance reduces throughput at constant suction pressure and may be necessary under certain conditions to prevent driver overload. Adding clearance also increases suction pressure at constant throughput. This may help minimize interstage pressure changes if throughput changes over life.

Clearance is normally expressed as a percentage of displacement and can be calculated as follows:

Single-acting cylinder (head-end clearance)

$$\text{Percent C} = \frac{C_{\text{HE}}}{\left(d_c^2\right)(0.785)(s)} \times 100 \tag{9.14}$$

Single-acting cylinder (crank-end clearance)

$$\text{Percent C} = \frac{C_{\text{CE}}}{\left(d_c^2 - d_r^2\right)(0.785)(s)} \times 100 \tag{9.15}$$

Double-acting cylinder (head-end + crank-end)

$$\text{Percent C} = \frac{C_{\text{HE}} + C_{\text{CE}}}{\left(2d_c^2 - d_r^2\right)(0.785)(s)} \tag{9.16}$$

where;

Percent C = cylinder clearance, percent of piston displacement
d_c = diameter of cylinder, mm (in)
d_r = diameter of rod, mm (in)
C_{HE} = head-end clearance, mm ^{3} (in^3)
C_{CE} = crank-end clearance, mm^3 (in^3)
s = stroke length, mm (in)

The throughput may be changed by changing the piston displacement. The most common type of displacement effect is by speed control. Higher speed equates to more actual volume displacement per minute.

Displacement may also be controlled by removing or deactivating suction valves. This has the effect of reducing the displacement by converting a double-acting cylinder into a single-acting cylinder.

Changing the cylinder liner, installing a sleeve, or boring the cylinder will directly affect the diameter of the cylinder either upward or downward. Liners must be specified when initial equipment orders are placed.

Finally, decreasing the rod diameter causes the crank-end displacement to increase. Before making such a change it is important to check that the smaller diameter rod has sufficient strength to handle its rod loading.

Always check with the manufacturer for specific details about changing the displacement.

9.1.9 Power

In specifying a compressor, it is necessary to choose the basic type, the number of stages of compression, and the power required. For these choices to be made, the volume of gas, suction and discharge pressure, suction temperature, and gas specific gravity must be known.

The detailed calculation of power and number of stages depends on the choice of type of compressor, and the type of compressor depends in part on power and number of stages. A first approximation of the number of stages can be made by assuming a maximum compression ratio per stage of between 3.0 and 4.0 and choosing the number of stages such that the ratio per stage (Eq. 9.5) is any number <4.

A first approximation for power can then be made by using Fig. 9.10 or the following approximate equation:

Field unit

$$\mathrm{BHP} = 22\, RnFQ_g \tag{9.17a}$$

Metric unit

$$\mathrm{BHP} = 0.0139\, RnFQ_g \tag{9.17b}$$

where:

BHP = approximate brake horsepower, kW (hp)
R = compression ratio (P_d/P_s) of a compressor stage
n = number of stage
F = an allowance for interstage pressure drop
= 1.00 for single-stage compression.
= 1.08 for two-stage compression.
= 1.10 for three-stage compression
Q_g = gas flow rate, stdm^3/h (MMSCFD)

A more accurate determination of power can be determined by applying Eq. (9.18) to each stage of compression.

Field unit

$$\mathrm{BHP} = 0.0857[Z_{av}]\left[\frac{(Q_g)(T_s)}{E_a E_m}\right]\left[\frac{k}{(k-1)}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right] \tag{9.18a}$$

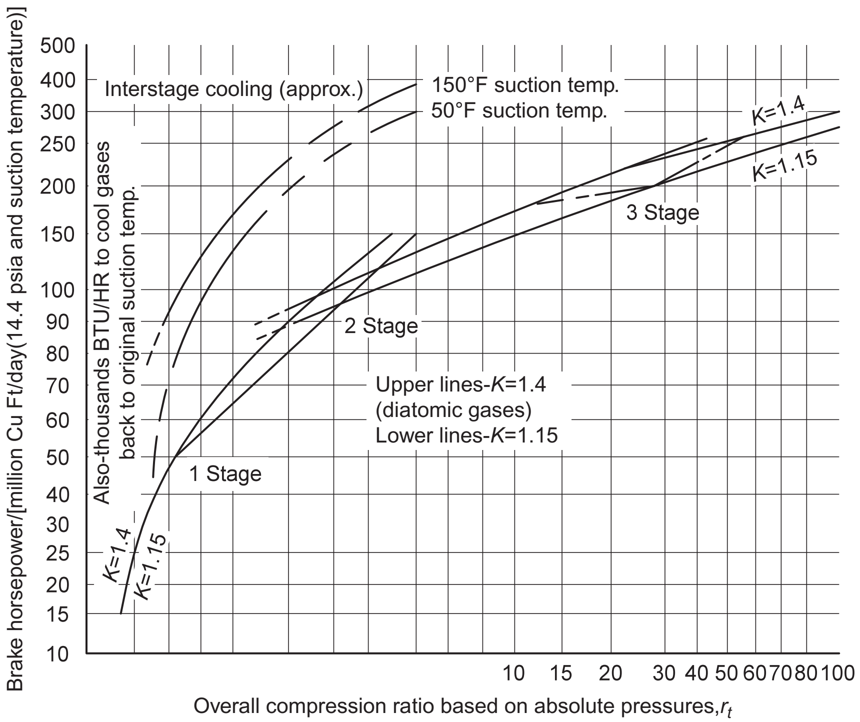


Fig. 9.10 "Quick look" approximation for determining brake horsepower.

Metric unit

$$\text{BHP} = 9.74 \times 10^{-5} [Z_{av}] \left[\frac{Q_g T_s}{E_a E_m}\right] \left[\frac{k}{(k-1)}\right] \left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right] \tag{9.18b}$$

where:

BHP = brake power, kW (hp)
Q_g = gas flow rate, stdm^3/h (MMSCFD)
T_s = stage suction temperature, K (°R)
Z_{av} = average of suction and discharge compressibility factor, $\frac{(Z_s - Z_d)}{2}$
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
E_m = mechanical efficiency of units,

- high-speed reciprocating units use 0.93 to 0.95.
- low-speed reciprocating units use 0.95 to 0.98

E_s = adiabatic efficiency, 0.85 to 0.90
k = ratio of gas specific heats, $\frac{C_p}{C_v}$
P_s = stage suction pressure, kPa (psia)
P_d = stage discharge pressure, kPa (psia)

The mechanical efficiency, E_m, of the unit is not to be confused with the volumetric efficiency, Ev, discussed before. Mechanical efficiency takes into account mechanical friction in the compressor in relating the energy input to the compressor as BHP to the energy required to compress the gas. The total power for the compressor is the sum of the power required for each stage and an allowance for interstage pressure losses. It is generally assumed that there is a 3% loss of pressure in going through the cooler, scrubbers, piping, and so on, between the actual discharge of the cylinder and the actual suction of the next cylinder. For example, if the discharge pressure of the first stage is 100 psia (689.5 kPa), the pressure loss is assumed to be 3 psi (20.7 kPa) and the suction pressure of the next stage is assumed to be 97 psia (668.8 kPa). That is, second-stage suction pressure is not equal to the first-stage discharge pressure. Interstage pressure losses may be as high as 5% in some applications.

9.1.10 Number of stages

The number of stages is governed by the following factors:

- Allowable discharge temperature.
- Rod loading.
- Existence of a fixed sidestream pressure level (where flow is added to or withdrawn from main flow of compressor)
- Allowable working pressure of available cylinders.

9.1.10.1 Discharge temperature

Discharge temperature is the most important factor affecting the number of stages. Compressors are generally limited to 300°F (149°C) for most gases in upstream and downstream facilities. API 618 further limits the discharge temperature of hydrogen-rich gases to 275°F (135°C). These limits restrict the stage pressure ratios. It is often necessary to increase the number of stages so that intercoolers can be added to keep the discharge temperature within limits, while achieving the required overall pressure ratio. Knowing the discharge temperature limit, Eq. (9.7) can be rewritten to find the allowable pressure ratio as follows:

$$r = \left(\frac{t_d + 460}{t_s + 460}\right)^{\frac{k}{k-1}} \tag{9.19}$$

Fig. 9.9 may also be used to find R corresponding to a given discharge temperature limit.

Adding intercoolers to a centrifugal compressor tends to save horsepower. However, with reciprocating compressors there will seldom be any benefit in adding intercoolers beyond those needed to maintain discharge temperature limits. The reasons are as follows:

- Reciprocating compressors are already highly efficient, and adding an intercooler adds pressure drop which offsets the power savings.
- Addition of a stage requires additional cylinder(s), pulsation dampers, knockout drum, and piping.

9.1.10.2 Rod loading

The rod load limit can affect the number of stages since the combined rod loading is related to the differential pressure across the cylinder. Increasing the number of stages obviously reduces the differential pressure of each stage. Quite often a rod loading problem can be solved by using two cylinders for one compression stage. In this case, the differential pressure would remain the same, but piston area, upon which the differential pressure acts, would be reduced. Rod loading is covered further later in this chapter.

9.1.10.3 Existence of a fixed sidestream pressure level

Sometimes a compressor application has more than one suction or discharge pressure level. For example, in an oil field gas system, the compressor may take different quantities of gas from the separator at two pressures, say 40 and 250 psig (275–1724 kPa). This machine could also be required to deliver a portion of the gas at 1000 psig (6895 kPa) for gas lift, and the remainder at 2500 psig (17,237 kPa) for injection back into the formation. In this case, these pressures would set the interstage pressures so that the sidestreams are accomplished. Note also that two stages might be required between the 40 and 250 psig (275–1724 kPa) levels (depending on suction temperature and k value) to stay below the discharge temperature limits.

9.1.10.4 Allowable working pressure

Occasionally a given pressure ratio might be achieved on one stage with satisfactory discharge temperature and rod loading, but an actual cylinder does not exist to handle both the capacity (ICFM) and pressure. In these situations, it is necessary to use two stages or use two similar single-stage cylinders depending on hardware and economics.

9.1.10.5 Procedure for determining the number of stages

The following procedure can be used to calculate the number of stages of compression and the power of a unit:

First, calculate the overall compression ratio using Eq. (9.5). If the compression ratio is under 5, consider using one stage. If it is not, select an initial number of stages so that the ratio per stage is <5. For initial calculations it can be assumed that the ratio per stage is equal for each stage.

Next, calculate the discharge gas temperature for the first stage using Eq. (9.7). If the discharge temperature is too high, $>300°F$ (149°C), either a large enough number of stages has not been selected or additional cooling of the suction gas is required. If the suction gas temperature to each stage cannot be decreased, increase the number of stages by one and recalculate discharge temperature.

Once the discharge temperature is acceptable, calculate the power required, suction pressure, discharge temperature, and horsepower for each succeeding stage.

If $R>3$, recalculate, adding an additional stage to determine if the addition of another stage could result in a substantial savings on power.

9.1.11 Rod load

The rods that produce the reciprocating motion of the piston can be subjected to high loads as the gas is compressed. The rod loads depend upon the action of the cylinder. Rod loads can be estimated (neglecting inertia loads) as follows:

Single-acting cylinder (head-end compression)

Field unit

$$\mathrm{RL}_c = a_p(P_d - P_u) + a_r P_u \tag{9.20a}$$

Metric unit

$$\mathrm{RL}_c = \frac{a_p(P_d - P_u) + a_r P_u}{1000} \tag{9.20b}$$

Field unit

$$\mathrm{RL}_t = a_p(P_u - P_s) + a_r P_u \tag{9.21a}$$

Metric unit

$$\mathrm{RL}_t = \frac{a_p(P_u - P_s) + a_r P_u}{1000} \tag{9.21b}$$

Single-acting cylinder (crank-end compression)

Field unit

$$\mathrm{RL}_c = a_p(P_u - P_s) + a_r P_s \tag{9.22a}$$

Metric unit

$$\mathrm{RL}_c = \frac{a_p(P_u - P_s) + a_r P_s}{1000} \tag{9.22b}$$

Field unit

$$\mathrm{RL}_t = a_p(P_d - P_u) + a_r P_d \tag{9.23a}$$

Metric unit

$$\mathrm{RL}_t = \frac{a_p(P_d - P_u) + a_r P_d}{1000} \tag{9.23b}$$

Double-acting cylinder

Field unit

$$\mathrm{RL}_c = a_p(P_d - P_s) + a_r P_s \tag{9.24a}$$

Metric unit

$$\mathrm{RL}_c = \frac{a_p(P_d - P_s) + a_r P_s}{1000} \tag{9.24b}$$

Field unit

$$\mathrm{RL}_t = a_p(P_d - P_s) + a_r P_d \tag{9.25a}$$

Metric unit

$$\mathrm{RL}_t = \frac{a_p(P_d - P_s) + a_r P_d}{1000} \tag{9.25b}$$

where;

RL_c = rod load in compression, Newton (lb)
RL_t = rod load in tension, Newton (lb)
a_p = cross-sectional area of piston, mm^2 (in^2)
P_u = pressure in unloaded end, kPa (psia)
P_d = stage discharge pressure, kPa (psia)
P_s = stage suction pressure, kPa (psia)
a_r = cross-sectional area of rod, mm^2 (in^2)

Good design requires load reversal (going from compression to tension) so that bearing surfaces will part to allow lubrication. For a double-acting cylinder at high ratios, RL_C may become larger than allowable. At low ratios, RL_t may become negative, indicating no load reversal.

The pressure in the unloaded end can be either P_s (if suction valves are removed), P_d (if discharge valves are removed), or atmospheric (if the cylinder is designed for single acting only). To get load reversal when single acting a double-acting cylinder, discharge pressure should be used for the unloaded end for head-end compression and suction pressure for crank-end compression.

9.1.12 Cylinder sizing

Cylinder displacement is easily understood with basic geometry. There are three cylinder configurations to consider:

- Single acting.
- Standard double acting.
- Double acting with tail rods

Refer to Fig. 9.2 for an illustration of a single-acting cylinder. Fig. 9.11 illustrates a double-acting configuration. Following are displacement rate equations.

9.1.12.1 Single acting

Cylinder displacement can be calculated from the following:

$$V_{\mathrm{cyl}} = \frac{\pi}{4} D^2 S = 0.785 D^2 S \tag{9.26}$$

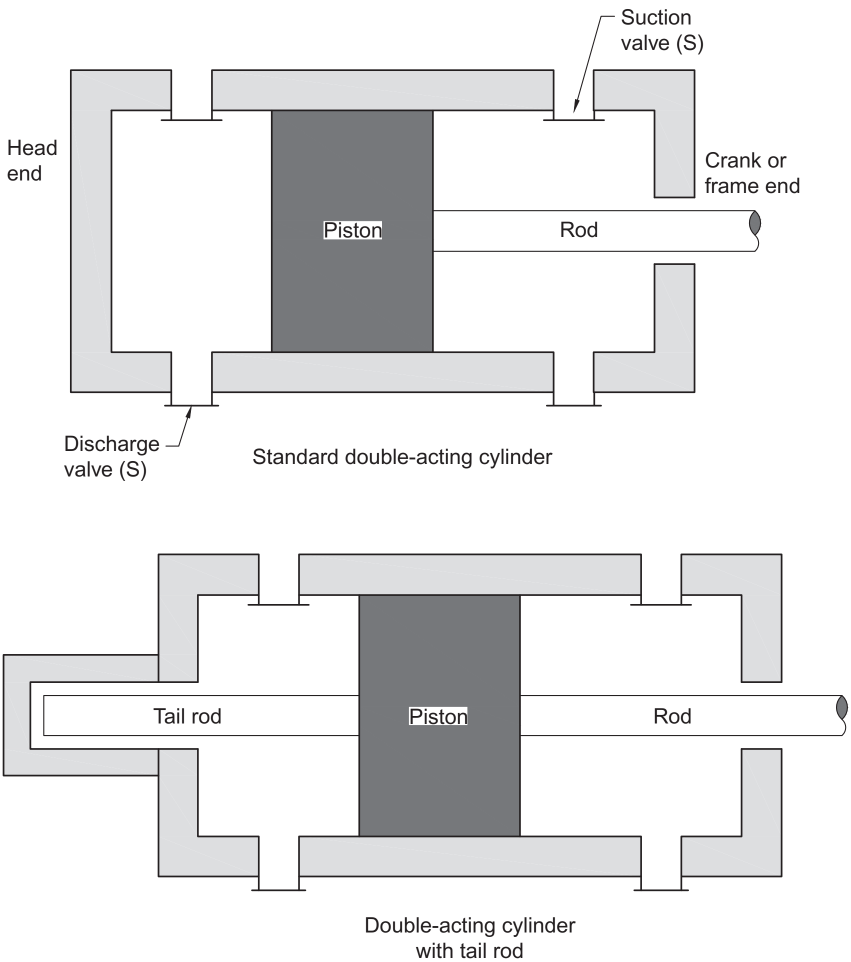


Fig. 9.11 Double-acting cylinders.

where:

V_{Cyl} = displacement, in^3
D = piston diameter, in
S = stroke, in

The displacement rate can be calculated from the following:

$$V_d = \left(\frac{\pi}{4}\right)\frac{D^2SN}{1728} = \frac{D^2SN}{2200} \tag{9.27}$$

where:

V_d = displacement, CFM
N = RPM

9.1.12.2 Standard double acting

Cylinder displacement is calculated from the following:

$$V_{\text{cyl}} = 0.785\left(2D^2 - d^2\right)S \tag{9.28}$$

where:

D = piston diameter, in
D = rod diameter, in

Cylinder displacement rate is calculated from the following:

$$V_d = \frac{(2D^2 - d^2)\text{SN}}{2200} \tag{9.29}$$

9.1.12.3 Double acting with tail rod

Cylinder displacement is calculated from the following:

$$V_{\text{cyl}} = 1.571\left(D^2 - d^2\right)S \tag{9.30}$$

Cylinder displacement rate is calculated from the following:

$$V_d = \frac{(D^2 - d^2)\text{SN}}{1100} \tag{9.31}$$

9.1.12.4 Estimating cylinder size to accommodate a given flow rate

Compressor cylinders are made in certain classes defined by:

- Maximum allowable working pressure (MAWP).
- Stroke.
- Number of valves per cylinder.
- Diameter range.

The diameter in a class is varied by changing the wall thickness of the inner cylinder or liner. Each class of cylinder has essentially the same clearance volume regardless of diameter. However, the percent clearance varies with diameter. Volumetric efficiency also varies since it is related to clearance. For example, assume that a 12-in. stroke double-acting cylinder has a 3-in. rod, a MAWP of 1000 psig, a speed of 360 RPM, a diameter range of 9 to 11 in., and a clearance of 15% when the diameter is 9 in. Further assume that the application has a pressure ratio of three, a molecular weight of 20, and a k-value of 1.25. Fig. 9.12 shows how present clearance and volumetric efficiency vary with diameter for this cylinder class.

The percent clearance varies from about 8% to well over 30% among the many classes of cylinders available. There is no rule of thumb to relate percent clearance to diameter with much accuracy. However, for rough estimates use:

Dia.[1]	Displ. rate, V_d, CFM	Displ., V_{cyl}, in^3	Cl. vol., in^3	% Clearance	E_V, %
9	300	1442	270	18.7	66.5
9-1/2	337	1616	270	16.7	69.3
10	375	1800	270	15.0	71.7
10-1/2	415	1993	270	13.6	73.6
11	458	2196	270	12.3	75.5

Fig. 9.12 Example: percent clearance and volumetric efficiency variance.

The common approach to cylinder sizing is to make an educated guess at E_V and then solve for the displacement rate using Eq. 9.4. Cylinder diameter can then be calculated using Eqs. (9.27), (9.29), and (9.31), as appropriate. This approach may have to be repeated two or three times to arrive at a combination that satisfies a given inlet flow quantity, Q.

As cylinder diameters do not come in an infinite number of increments, it is customary to select the next largest increment. In multistage machines, depending on the size of increments, oversizing of an initial stage is sometimes balanced by slightly undersizing the subsequent stage, assuming the interstage pressure level is not fixed (by a sidestream, for example).

9.2 Compressor types

There are two basic categories of reciprocating compressors. They are distinguished by the style of piston and the linkage between the piston and crankshaft. The first category covers the lighter duty machines having "trunk-type" (automotive-type) single-acting pistons lubricated by crankcase oil, with no crossheads. These machines are typically used for air compression to 125 psig (861 kPa), although cylinders are available for working pressures to 6000 psig (41,369 kPa), for various gases. They operate at speeds in the 1200 to 1800 RPM and have ratings to about 125 HP (93 KW). This type of machine is not often used in petroleum operations thus it is not described further in this text.

The second category of reciprocating compressor is a heavy-duty crosshead-type machine where each piston is usually double acting and is connected to the crankpin by a piston rod, crosshead, and connecting rod. The cylinders are lubricated by a force-feed lubricator. Refer to Fig. 9.13 for a cross-section of a typical horizontally opposed reciprocating compressor.

Under the second category there are three primary types of reciprocating compressors used in flammable and hazardous gas services in the petrochemical industry. These are the

- Horizontally opposed heavy duty.
- Integral engine.
- Packaged high speed.

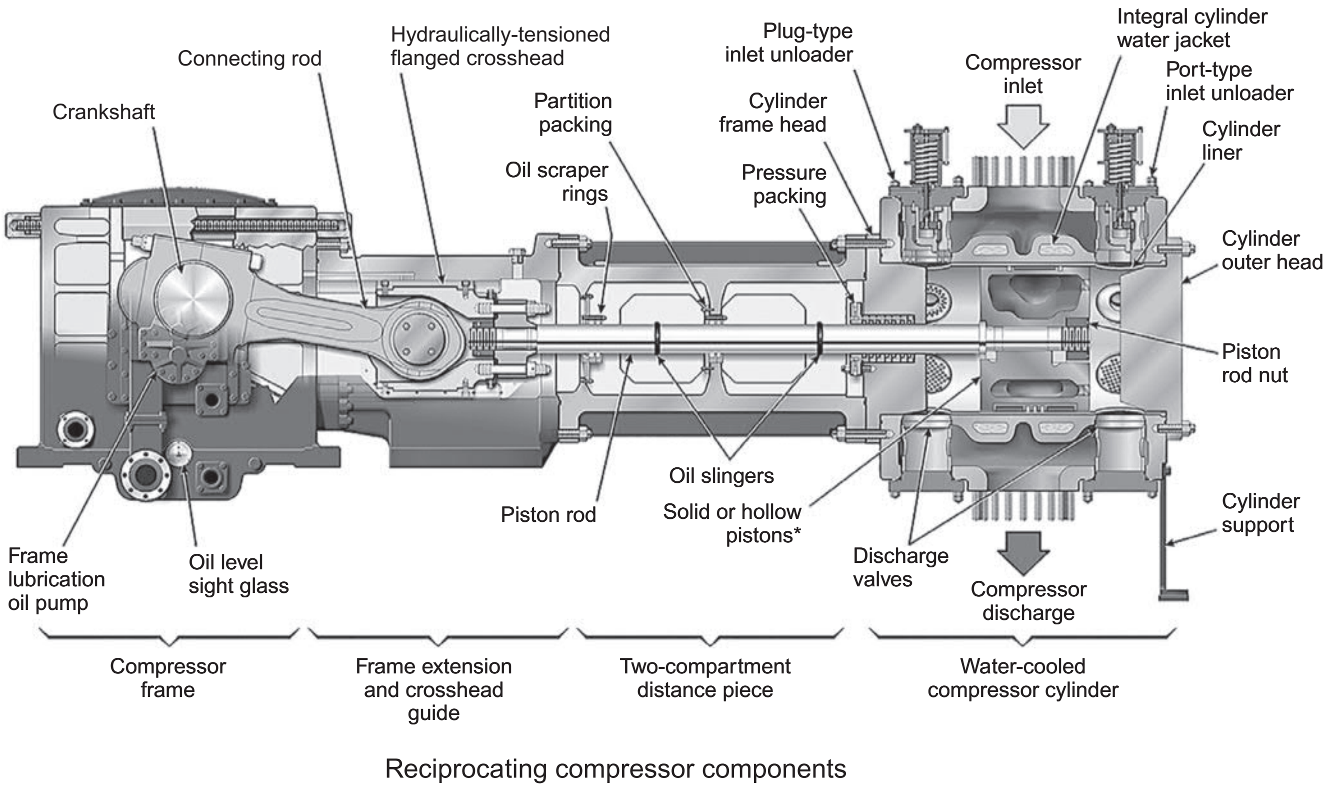


Fig. 9.13 Cutaway of a horizontally opposed reciprocating compressor.
Courtesy of Dresser-Rand Corporation.

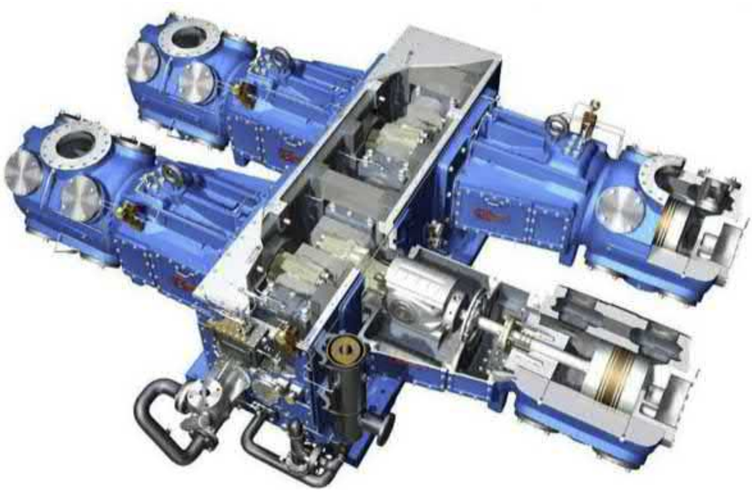


Fig. 9.14 Cutaway of a horizontally opposed four-cylinder reciprocating compressor. Courtesy of Ariel Corporation.

Horizontally opposed heavy duty type compressors are selected for most refinery and petrochemical plant reciprocating compressor services (refer to Fig. 9.14). They are slow-speed block-mounted machines supplied in accordance with API 618. They are generally driven by an engine-driven mounted synchronous motor. By engine mounted we mean that the inboard end of the motor is bolted to a hub on the end of the compressor crankshaft and is supported by the compressor main bearing on that end of the frame. The outboard of the motor is supported by a pedestal bearing.

Integral engine type compressors are generally found in large onshore oil producing fields and gas transmission applications, although they were used in many refinery applications in the past. (refer to Fig. 9.15). They are characterized by

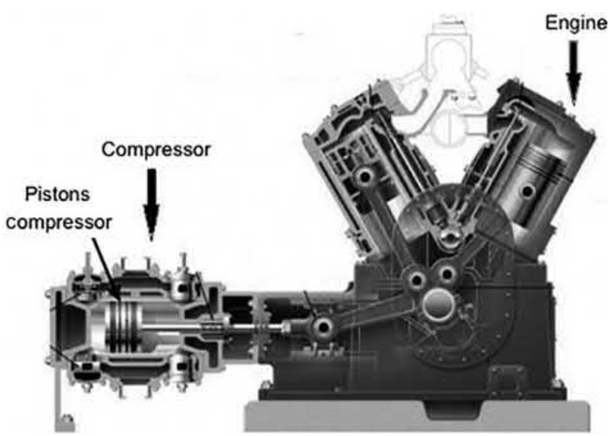


Fig. 9.15 Cutaway of an integral engine reciprocating compressor.

engine (or power) cylinders and compressor cylinders on the same crankshaft. Generally, the power cylinders are in a "vee" or inline configuration on the vertical centerline, and the compressor cylinders are mounted horizontally on one side of the frame. They were developed to be self-contained units that burn as well as compress field gas.

Packaged high-speed type compressors are used in onshore producing field services and in offshore services. They are small high-speed horizontally opposed compressors that are sold as complete equipment packages with driver and ancillary equipment on a skid. They can be motor or engine driven at 900 to 1000 rpm nominal running speed (refer to Figs. 9.16 and 9.17). Packaged high-speed units are supplied in accordance with API Specification 11P.

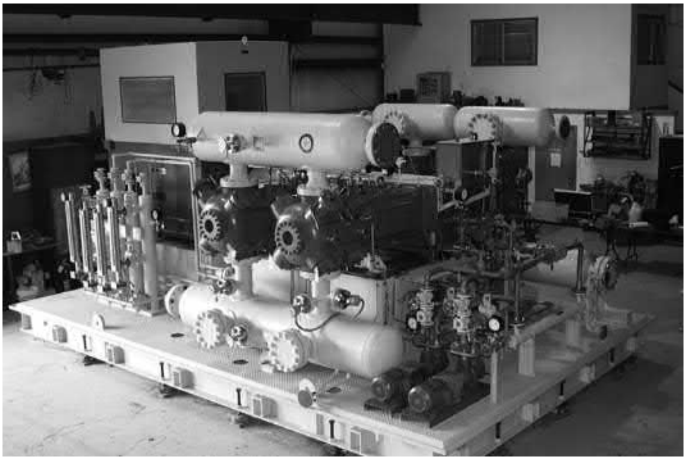


Fig. 9.16 Electric motor-driven reciprocating package.

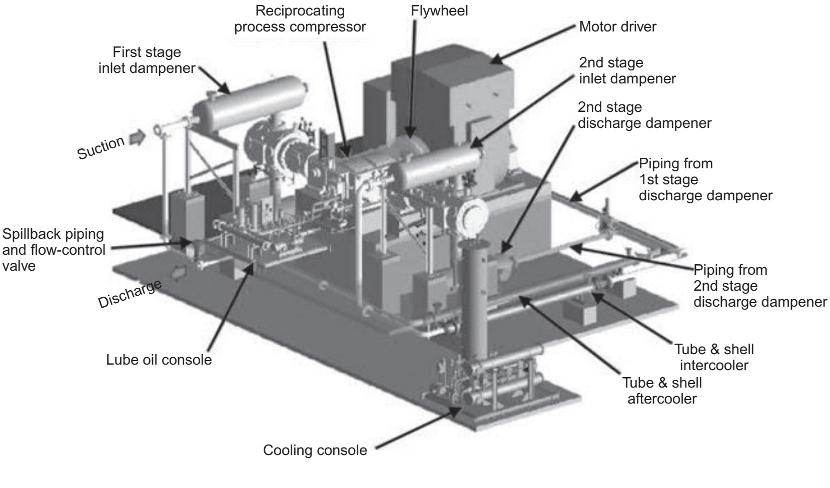


Fig. 9.17 Electric motor block-mounted reciprocating compressor.

9.3 Machine components

9.3.1 General considerations

In general, materials for construction of the compressor and auxiliaries are normally the manufacturer's standard for the specified operating conditions except as required by the data sheet or certain specifications. Reciprocating compressors are available in a variety of designs and arrangements. Fig. 9.18 is a cutaway view of a "separable" reciprocating compressor showing most of the components of a reciprocating compressor, specifically:

- Frame.
- Cylinder.
- Distance piece and packing case.
- Crankshaft.
- Connecting rod.
- Crosshead.
- Piston rod.
- Piston.
- Bearings.
- Valves.
- Capacity control devices.

9.3.2 Frame

Fig. 9.19 shows a compressor frame, which is a heavy, rugged casting containing all the rotating parts, and on which the cylinders and crosshead are mounted. All frames

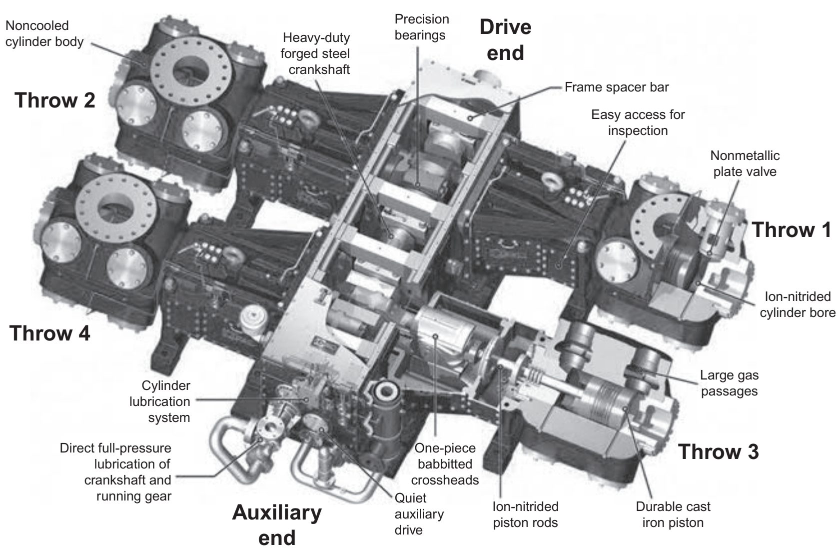


Fig. 9.18 Cutaway view of a "separable" reciprocating compressor.
Courtesy of McGraw-Hill Book Company.

Fig. 9.19 Example of a compressor frame.
Courtesy of Dresser-Rand.

are rated by compressor manufacturers for a maximum continuous power, which is determined by either the maximum power that can be transmitted through the crankshaft to the compressor cylinders or by the force load imposed on the frame by pressure differential between the two sides of the piston at the design speed.

9.3.2.1 Balanced opposed type

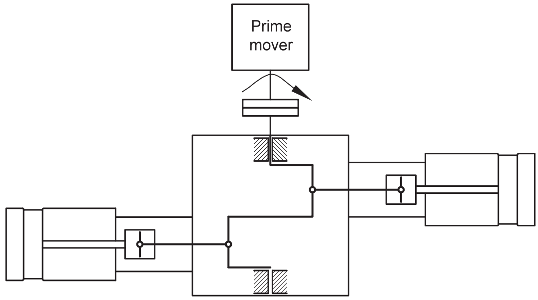
Balanced opposed type frames are characterized by an adjacent pair of crank-throws 180 degrees out of phase and separated only by a crank web. The frame is separate from the driver. Cranks are arranged so that the motion of each piston is balanced by the motion of the piston opposite it. Fig. 9.20 shows a balanced opposed compressor.

9.3.2.2 Integral type

Integral type frames are characterized by having compressor cylinders and power cylinders mounted on the same frame and driven by the same crankshaft. Engine cylinders may be vertical or at a v-angle to the compressor cylinders. Fig. 9.21 also shows an integral gas engine-driven compressor.

A typical gas compressor specification is shown in Fig. 9.22.

Fig. 9.20 Schematic diagram of a balanced-opposed type compressor frame.



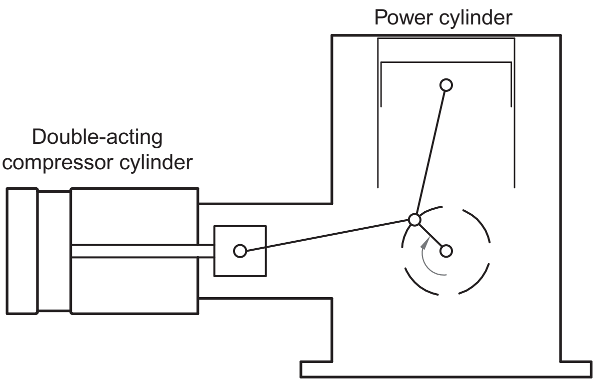


Fig. 9.21 Schematic diagram of an integral-type compressor frame.

Specifications

Model	W76	W74	W72	MW68	MW66	MW64	MW62
Frame horsepower (kW)[4]	4500 (3350)	3300(2460)	2250(1675)	2400(1790)	3000(2238)	2200(1641)	1300(970)
No. of cylinders	6	4	2	8	6	4	2
Stoke [in.(cm)]	7(17.78)	7(17.78)	7(17.78)	6(15.2)	6(15.2)	6(15.2)	6(15.2)
Rpm range	450–900	450–900	450–900	450–900	450–1000	450–1000	450–1000
Rod diameter [in.(cm)]	2.5(6.35)	2.5(6.35)	2.5(6.35)	2.55(5.7)	2.55(5.7)	2.55(5.7)	2.55(5.7)
Rod load [lb.(kg)]	45,000(20410)	45,000(20410)	45,000(20410)	35,000(16000)	35,000(16000)	35,000(16000)	35,000(16000)
Piston speed 1 [rpm(m/s)]	1050(5.33)	1050(5.33)	1050(5.33)	900(4.6)	900(4.6)	900(4.6)	900(4.6)
Crankshaft Pins/Journals [in.(cm)]	8(20.32)	8(20.32)	8(20.32)	7(17.8)	7(17.8)	6[2](15.2)	7(17.8)
Material	Forged Steel	Forged Steel	Forged Steel	Forged Steel	Forged Steel	Forged Steel	Forged Steel
Centerline from bottom [in.(cm)]	24(60.96)	24(60.96)	24(60.96)	18(45.7)	18(45.7)	18(45.7)	18(45.7)
Main bearing dia.[in.(cm)]	8(20.32)	8(20.32)	8(20.32)	7(17.8)	7(17.8)	6(15.2)	7(17.8)
Thrust bearing dia.[in.(cm)]	8(20.32)	8(20.32)	8(20.32)	7(17.8)	7(17.8)	6(15.2)	7(17.8)
Connecting rod bearing dia.[in.(cm)]	8(20.32)	8(20.32)	8(20.32)	7(17.8)	7(17.8)	6(15.2)	7(17.8)
Connecting rod center to center L[in.(cm)]	17(43.18)	17(43.18)	17(43.18)	14.5(36.8)	14.5(36.8)	14.5(36.8)	14.5(36.8)
Connecting rod bolts dia.[in.(cm)]	1.125(2.8575)	1.125(2.8575)	1.125(2.8575)	1(2.5)	1(2.5)	1(2.5)	1(2.5)
Crosshead dia.[in.(cm)]	13.75(34.29)	13.75(34.29)	13.75(34.29)	10.5(26.7)	10.5(26.7)	10.5(26.7)	10.5(26.7)
Crosshead shoe width [in.(cm)]	8.5(21.59)	8.5(21.59)	8.5(21.59)	6.75(17.1)	6.75(17.1)	6.75(17.1)	6.75(17.1)
Crosshead pin type	Tapered, Locked	Tapered, Locked	Tapered, Locked	Tapered, Locked	Tapered, Locked	Tapered, Locked	Tapered, Locked
Crosshead pin dia.[in.(cm)]	4.625(11.747)	4.625(11.747)	4.625(11.747)	3.75(9.5)	3.75(9.5)	3.75(9.5)	3.75(9.5)
Model	4.625(11.747)	4.625(11.747)	4.625(11.747)	3.75(9.5)	3.75(9.5)	3.75(9.5)	3.75(9.5)
Lube oil flow [gpm(l/min)]	53(132)	53(132)	18(68)	32(121)	24(91)	16(61)	8(30)
Sump capacity [gal(l)]	160(600)	80(304)	30(113)	105(397)	65(248)	40(151)	15(57)
Width3 [ft–in.(cm)]	15'0"(457)	15'0"(457)	15'0"(457)	11'0"(335)	11'0"(335)	11'0"(335)	11'0"(335)
Length3 [ft–in.(cm)	15'5"(470)	9'10"(300)	5'10"(178)	13'10"(422)	10'6"(320)	7'2"(218)	4'2"(127)
Frame weight–average [lb(kg)]	29,000(13156)	23,300(10570)	13,000(5900)	38,500(17460)	29,000(13150)	19,500(8840)	10.00(4535)

[1]Piston speed given at 900 rpm.
[2]7" Diameter available for applications heater than 1500 hp.
[3]Dimensions are approximate: not to be used for construction purposes.
[4]For higher horsepower applications consult superior's marketing department.

Fig. 9.22 A typical reciprocating compressor specification.

9.3.3 Cylinder

9.3.3.1 Overview

A cylinder is a pressure vessel that holds the gas during the compression cycle. There are two basic types:

- *Single-acting* (trunk pistons) cylinders are those in which compression takes place on only one of the two piston strokes per revolution. They are similar to pistons used in internal combustion engines. The piston is directly connected to the connecting rod. This arrangement cannot be positively sealed on the back side of the piston. They are usually found in small air and nontoxic gas compressors. Fig. 9.23 is a cutaway drawing of a single-acting cylinder. As the piston moves to the right, gas is discharged; as the piston moves to the left, gas is sucked in.

- *Double-acting* cylinders are those in which compression takes place on both of the piston strokes per revolution. They have a straight rod and a crosshead. The crosshead is connected to the crank-connecting rod. This allows pure axial movement which makes sealing possible on both sides of the piston. This arrangement is found in heavy duty, toxic, and flammable gas services. Fig. 9.24 is a cutaway view of a double-acting cylinder. As the piston moves to the right, gas is discharged from the head end of the cylinder and enters the cylinder at the crank end. As the piston moves to the left, gas enters the head end and is discharged at the crank end.

In addition, there are two other arrangements. The first is double acting with tail rod. In this case the rod goes beyond the head of the piston and penetrates the outboard cylinder head. This is done in some small high-pressure cylinders to balance the surface area and axial force acting on the piston. However, the arrangement adds another

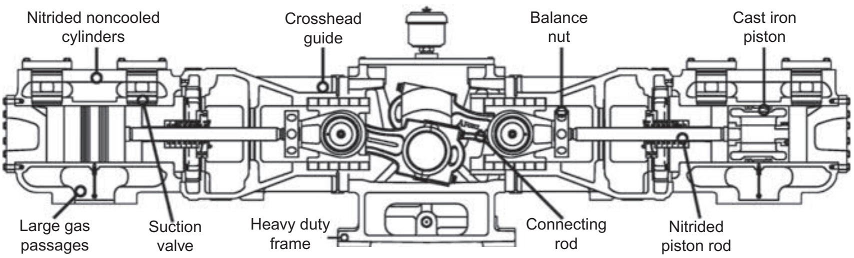


Fig. 9.23 Sectional view of a single-acting cylinder.
Courtesy of Dresser Rand Company.

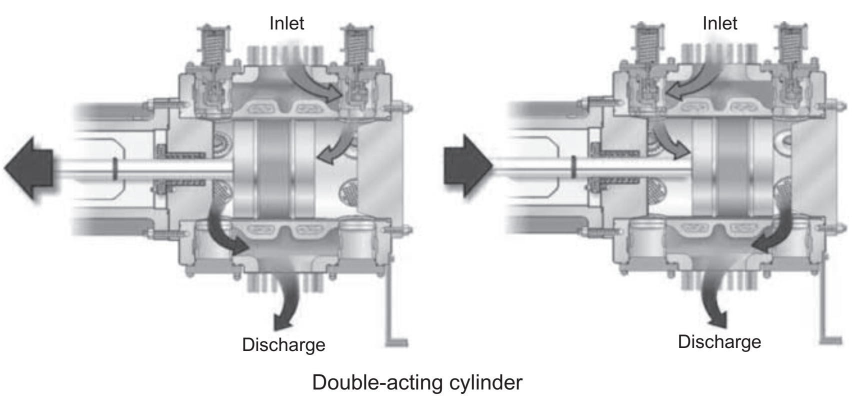


Fig. 9.24 Sectional view showing the flow through a double-acting cylinder.
Courtesy of Dresser-Rand.

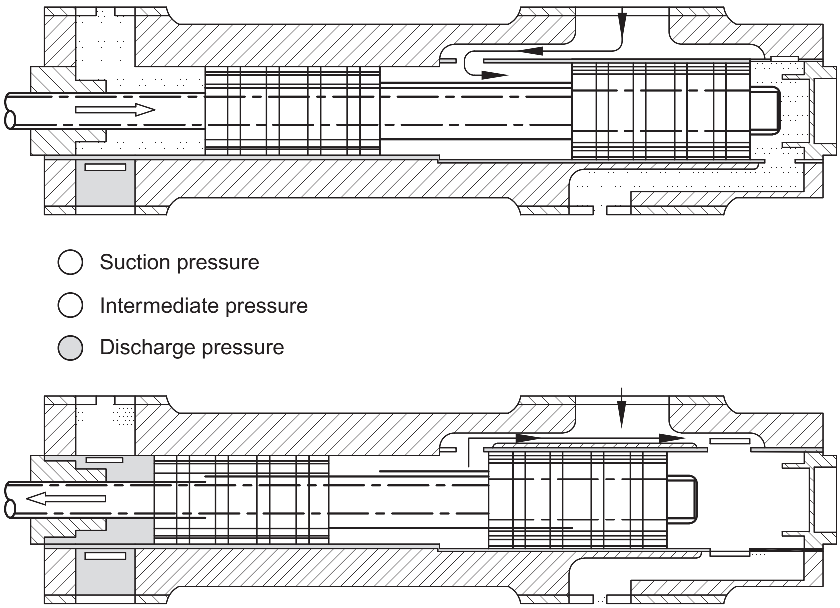


Fig. 9.25 Cutaway of a stepped-piston cylinder.

location where sealing is necessary. The second arrangement is a "stepped-piston" arrangement (refer to Fig. 9.25) which defines a cylinder that compresses gas at two different pressures. The cylinder compresses the lower pressure gas with the head end of the larger piston and the high-pressure gas with the crank end of the smaller piston.

9.3.3.2 Cylinder materials

The choice of material is usually selected based on operating pressure and/or resistance to thermal or mechanical shock. Cylinder material is also selected based on wear compatibility of rubbing parts (piston and cylinder bore, piston rod, and seal rings). The material will depend upon whether the cylinder is lubricated or nonlubricated. Thermoplastic rings or rider bands are used to prevent metal-to-metal contact for non-lubricated cylinders. The size of the cylinder also influences the method of construction.

For example, large cylinders are cast while smaller cylinders are forged. Large cylinders with many valves are limited to low pressures and are cast. As pressure increases, the cylinder is generally smaller and is forged.

Cast iron is generally used for cylinder operating pressures from 1000 to 1200 psig (6895 to 8274 kPa). Nodular iron is used for cylinder operating pressures up to 1500 psig (10,342 kPa), and cast steel is used for cylinder operating pressures in the 1000 to 2500 (6895 to 17,237 kPa). Forged steel is generally used for pressures above 2500 psig (17,237 kPa).

Like all pressure vessels, the cylinder has a maximum allowable working pressure (MAWP). Normally the maximum allowable working pressure of the cylinder determines the setting of the relief valve that is downstream of the cylinder; however, it is possible that the downstream piping or cooler will have a lower MAWP, which will determine the relief valve set pressure. The MAWP of the cylinder should be 10%, or at least 25 psif (172.4 kPa), greater than its operating pressure.

Typically, in specifying a unit, the pressures, power, inlet temperature, and gas properties are given. The actual sizing of the cylinders is usually left to the manufacturer from his specific combinations of standard cylinders, pistons, and liners. However, once a proposal is received from a manufacturer, sometimes it is beneficial to check the cylinder sizing and make sure that the compressor will perform.

Sometimes it is necessary to size a new cylinder for an existing compressor or to verify that an existing compressor will perform in a different service. The capacity of the cylinder is a function of piston displacement and volumetric efficiency. This in turn is a function of cylinder clearance, compression ratio, and gas properties.

9.3.3.3 Cylinder arrangements and staging

Fig. 9.26 illustrates two common types of cylinder arrangements. Figs. 9.27–9.29 illustrate single-, two-, and three-stage cylinder arrangements.

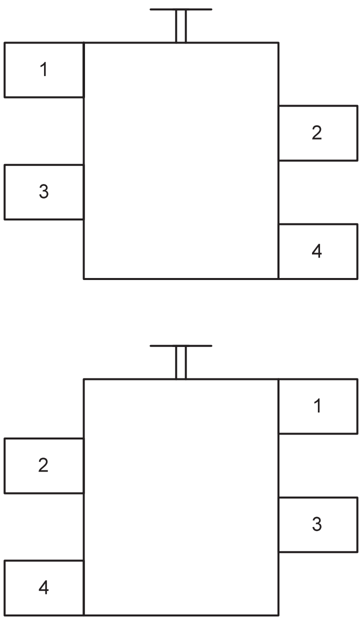


Fig. 9.26 Two common types of cylinder arrangements for a single-acting reciprocating compressor.

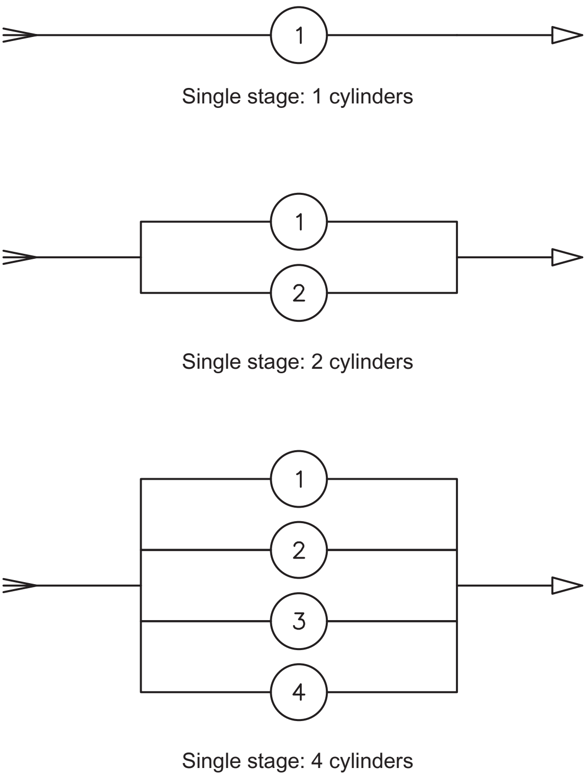


Fig. 9.27 Single-stage cylinder arrangements.

9.3.3.4 Cylinder liners

A cylinder liner such as those shown in Figs. 9.30 and 9.31 may be used to help prolong the life of the cylinder. Liners make repair less costly by taking the wear from the piston. It is often easier and less costly to replace a liner than to replace or remachine a cylinder.

Liners also allow the effective cylinder diameter to be changed without changing the complete cylinder. This provides flexibility for different conditions of pressure and flow rate.

Liners have the disadvantage of decreasing effective diameter and thus capacity for a given diameter cylinder, increasing clearance between valve and piston (thus decreasing capacity), and decreasing the effectiveness of jacket cooling.

9.3.4 Distance piece and packing case

9.3.4.1 General considerations

A distance piece provides the separation of the compressor cylinder from the compressor frame as shown in Fig. 9.32. It provides a housing for both wiper packing and pressure packing. The distance piece prevents entry of compressed gas into the crankcase

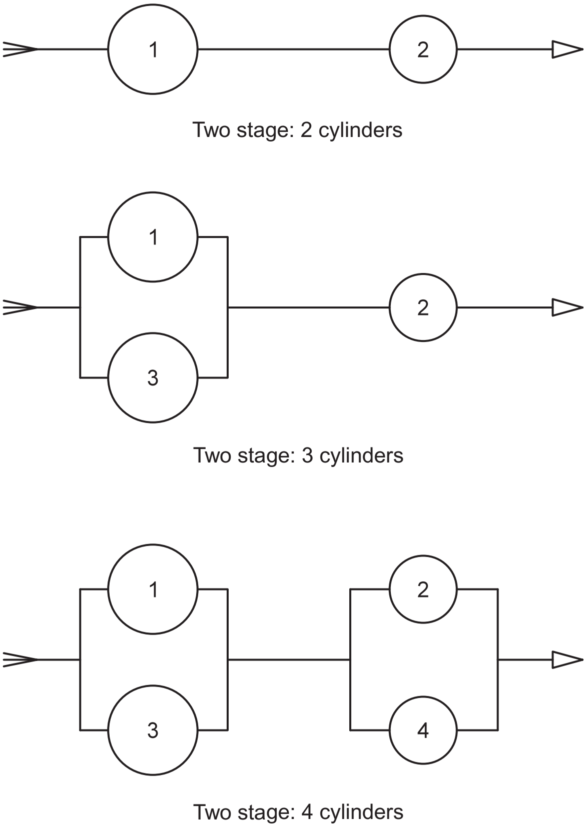


Fig. 9.28 Two-stage cylinder arrangements.

and provides access for maintenance of the packing and piston rod. Types of distance pieces, with corresponding diagrams, and their applications are covered in API 618. The three most commonly used distance pieces are shown in Fig. 9.33.

- *Type A* is a *standard (integral)* distance piece. The piston rod moves back and forth through packing that is contained within the distance piece. Any leaks in the packing are vented through a vent or drain (depending on whether they are gas or liquid leaks).

  The packing keeps compressed gas from leaking into the frame. As the rod passes through the packing it is lubricated. As it goes back and forth, it is in contact with the frame lubrication system and with the cylinder gas. Thus oil carryover can occur on the rod from cylinder to crankcase, and impurities that are picked up by the oil from the gas being compressed could contaminate crankcase oil.
- *Type B* is a *single-compartment* distance piece and both the frame end and the cylinder end contain packing. The distance between the cylinder packing and the frame diaphragm and packing is long enough to assure that no part of the rod enters both the cylinder and the frame. Therefore oil cannot carry contamination between the gas being compressed and the oil that is used to lubricate the crankcase.

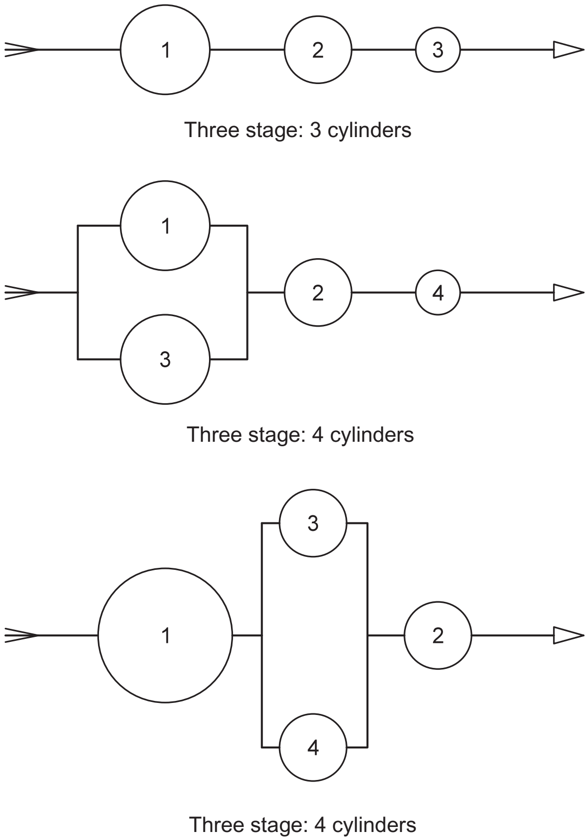


Fig. 9.29 Three-stage cylinder arrangements.

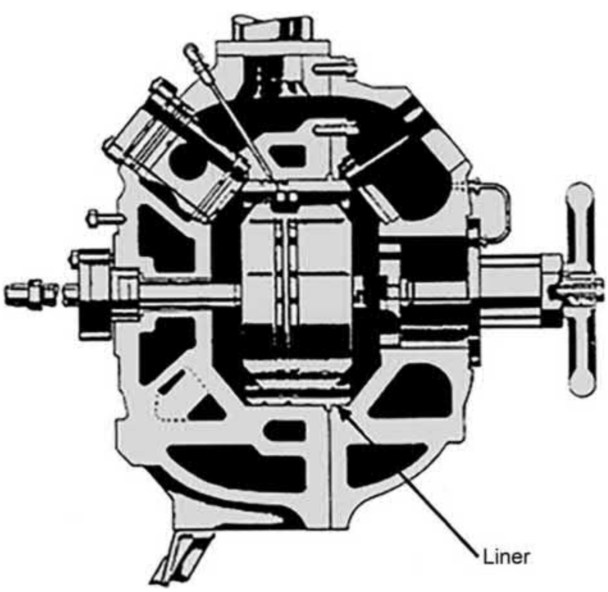


Fig. 9.30 Sectional view of a compressor cylinder showing the cylinder liner.

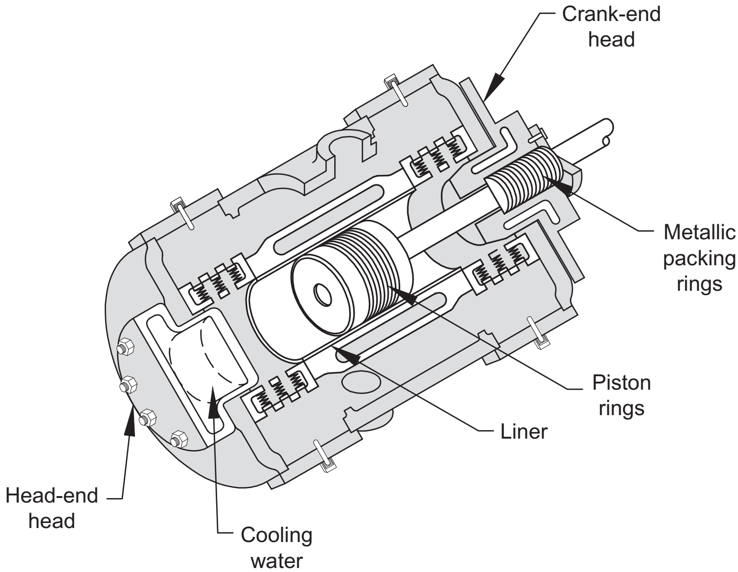


Fig. 9.31 Cutaway view of a double-acting cylinder showing cylinder liner, piston rings, and head-end cooling water.

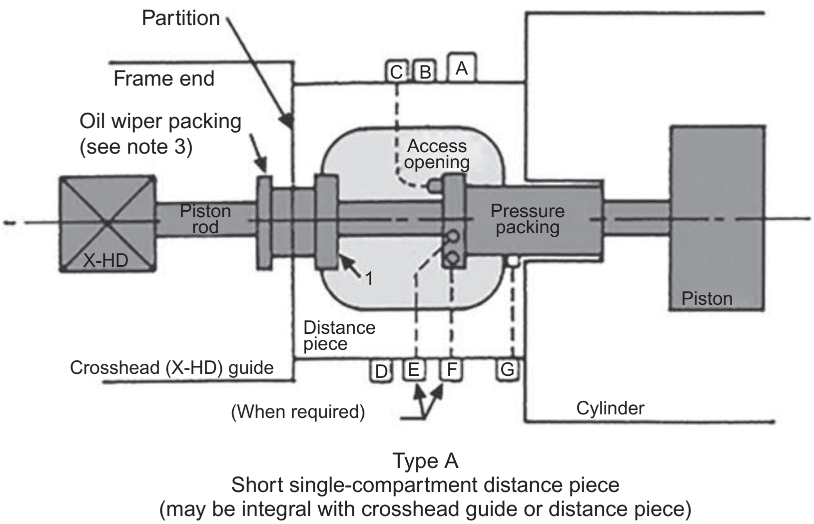


Fig. 9.32 Schematic diagram of a Type A short single-compartment distance piece.

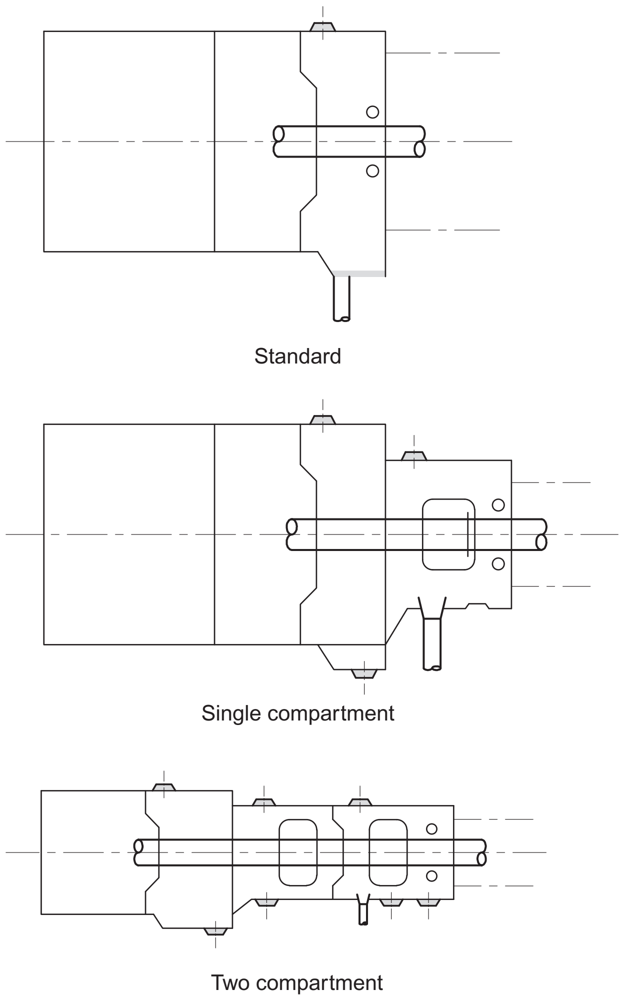


Fig. 9.33 API 618 types of distance pieces.

There are drains and vents off the distance piece and off the packing, so if there is a packing failure, the high-pressure gas has a place to vent and not build up pressure that could leak through the frame packing into the crankcase. An oil slinger to prevent packing lube oil from entering the compressor frames is also shown.

- *Type C* is a *two-compartment* distance piece that is sometimes used for toxic gases, but it is not very common. In this configuration, no part of the rod enters both the crankcase and the compartment adjacent to the compressor cylinder. That is, even if there were one failure, the crankcase oil cannot be contaminated with the toxic gas.

9.3.4.2 Vents and drains

Each compressor should have its own separate vent and drain system for distance pieces and packing. Vent and drain connections are shown on the API 618 diagrams

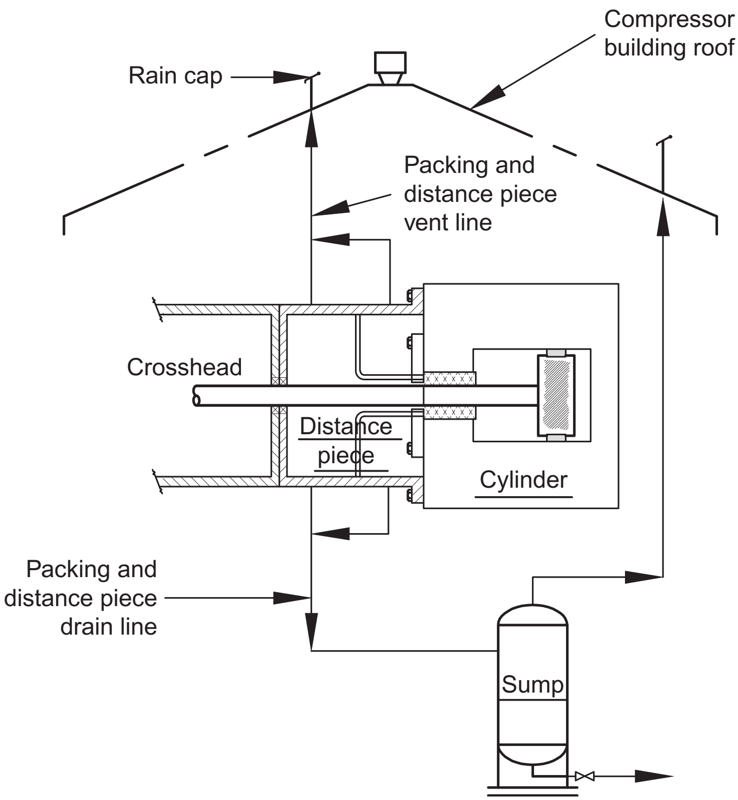


Fig. 9.34 API 618 distance piece packing and drain piping recommendations.

(refer to Fig. 9.34). No other vents should be tied to this system. Distance piece and packing vents should be piped into an open vent system that terminates outside and above the compressor enclosure (building) at least 25 ft horizontally from the engine exhaust. Distance piece drains should be piped into a separate sump that can be manually drained. Sump should be vented outside and above the compressor building. One should not drain contaminated oil to the frame reservoir. In some cases, lube oil can be mixed with crude oil; in other cases, it must be hauled off for disposal or to recycle facility.

9.3.5 Crankshaft, connecting rod, crosshead, and piston rod

Fig. 9.35 shows the crankshaft, connecting rod, crosshead, and piston rod.

9.3.5.1 Crankshafts

The crankshaft rotates about the frame axis, driving the connecting rod, crosshead, piston rod, and piston. Crankshafts are one-piece forgings or castings, although provisions are usually made for removable counterweights. Fig. 9.36 shows a crankshaft being lowered into a frame. Crankshafts are very expensive, long-delivery machined items.

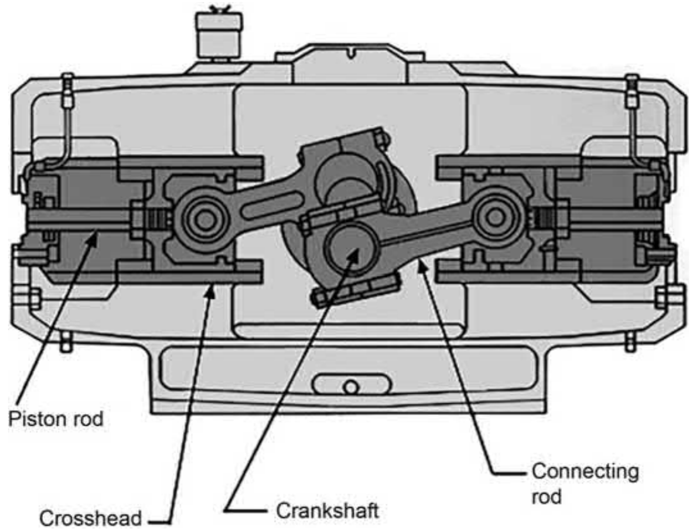


Fig. 9.35 Schematic diagram showing crankshaft, connecting rod, crosshead, and piston rod.

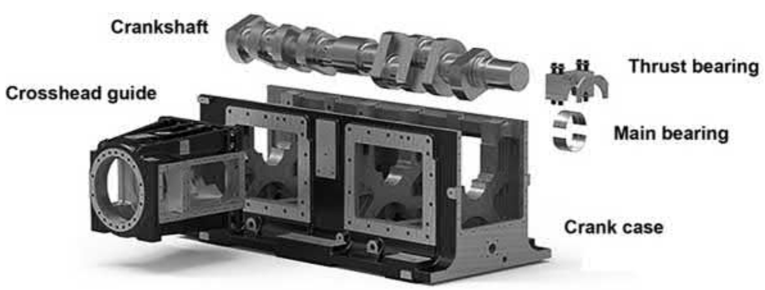


Fig. 9.36 Components of a reciprocating compressor: crosshead guide, crankshaft, thrust bearing, main bearing, and crank case.
Courtesy of Cooper Industries Energy Services Group.

9.3.5.2 Connecting rod and crosshead

The connecting rod connects the crankshaft to the crosshead. The crosshead converts the rotating motion of the connecting rod to a linear, oscillating motion, while driving the piston rod.

9.3.5.3 Piston rod

The piston rod drives the piston that compresses the gas in the cylinder. Rods are attached to pistons with a single extension of the rod through the piston or with multiple through-bolts as shown in Fig. 9.37. The advantages of multiple bolts are as follows:

- The smaller bolts are much easier to accurately tension.
- Adequate prestress levels are reliably maintained.
- Loading by the bolts is more evenly distributed in the piston.

This feature is especially useful for large diameter aluminum pistons with large diameter rods. Such pistons with the single through-bolt attachment sometimes have nut-loosening problems after a number of temperature cycles. Factors contributing to this looseness are the difficulty torqueing one large nut and possible nonsquareness of the nut's face with the piston surface.

9.3.6 Piston

9.3.6.1 General considerations

Pistons are located at the end of the piston rod (refer to Fig. 9.38). They act as the movable barrier in the compressor cylinder. Pistons are usually made from lightweight materials such as aluminum, cast iron, or steel with a hollow center (refer to Fig. 9.39).

Fig. 9.40 illustrates the three styles of pistons:

One-piece design is made of cast iron or steel for small diameters and high differential pressures. It is also sometimes used when necessary to add weight for balancing to reduce reciprocating shaking forces and thus reduce bearing pressure in the bore thus minimizing wear.

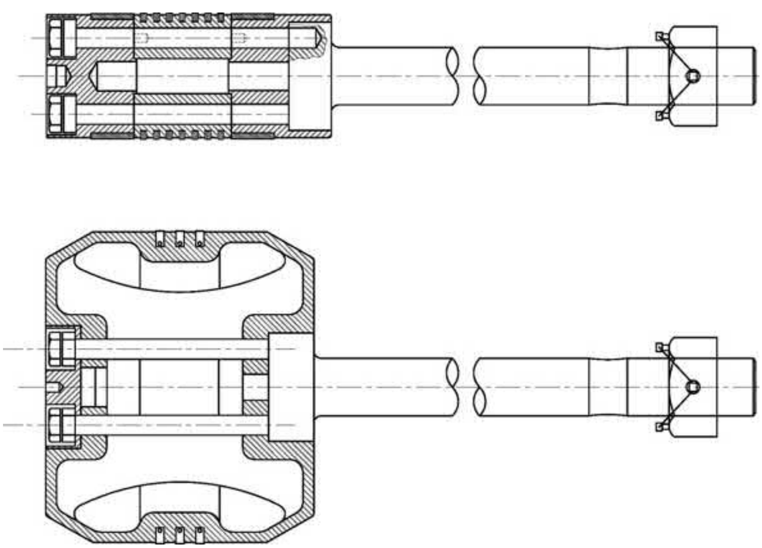


Fig. 9.37 Pistons with multiple through-bolts.
Courtesy of the Cooper Cameron Corporation.

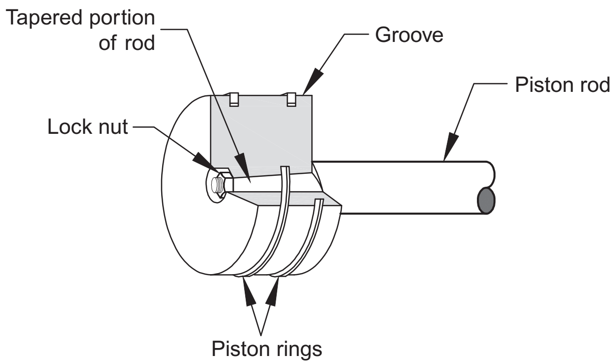


Fig. 9.38 Sectional of a compressor piston.

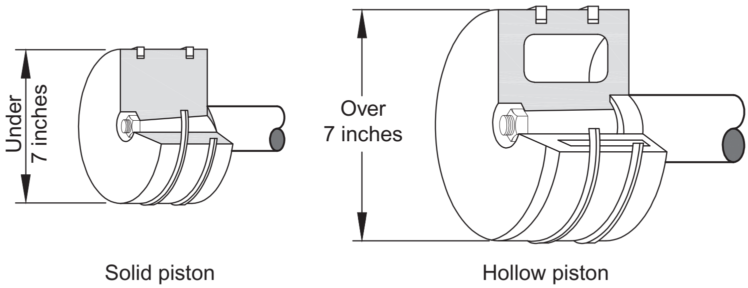


Fig. 9.39 Piston construction.

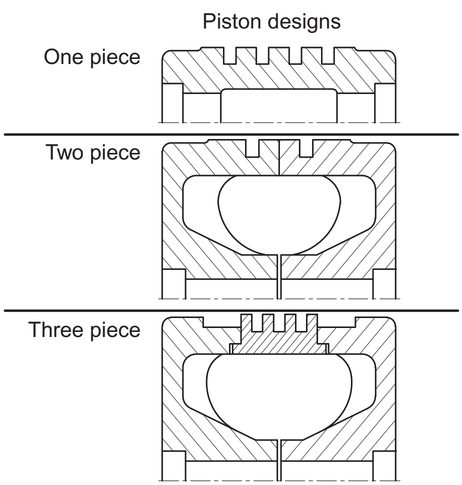


Fig. 9.40 Piston designs.
Courtesy of Dresser-Rand.

Two-piece design is used for ease in casting and weight control. It is made of aluminum or cast iron and is generally applied for diameters above 10 in. Aluminum is used to reduce reciprocating mass.

Three-piece segmental design incorporates a ring carrier. It is used to facilitate installation of rider rings (wear bands) which, when required, are placed on each side of the carrier. In this way, the rider band can be thicker because it does not have to be stretched over the outside diameter of the piston.

9.3.6.2 Piston rings and rider rings

The purpose of piston rings is to prevent the blowby of gas from one end of the piston to the other. Rider rings or wear bands (refer to Fig. 9.41)

- Support the weight of the piston.
- Help guide the piston in the bore.
- Prevent rubbing of the piston on the cylinder wall.

For many years, piston rings were made from nonabrasive, relatively soft metallic materials. Cast iron was the most common material, later largely replaced by bronze. Metallic rings were favored because of their good heat transfer characteristics. However, much development of nonmetallic piston rings and rider rings occurred when nonlube applications became popular. Today PTFE (Teflon) with various filler materials is the favored material because of its excellent low-friction characteristics. PTFE is used exclusively for lubricated and nonlubricated services. Bronze is still used on rare occasions for clean and dry lubricated service when good heat transfer is needed.

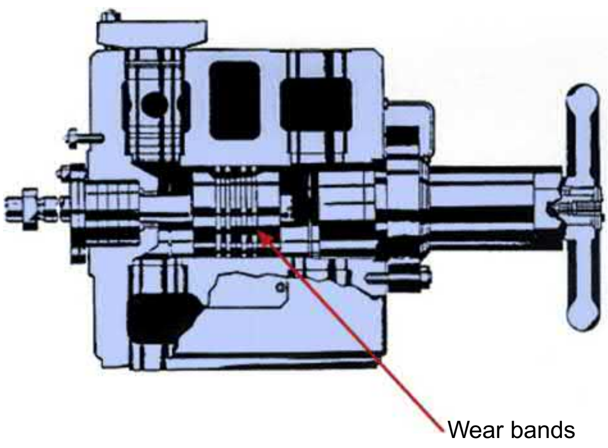


Fig. 9.41 Sectional view of a compressor cylinder showing the wear bands. Courtesy of Worthington Compressors.

Metal piston rings are made in either one piece with a gap or in several segments (Fig. 9.42). The shapes of piston and rider rings are shown in Fig. 9.43. The "angle" cut is generally preferred and is the most commonly used. For smaller lower pressure cylinders, the "step" cut is used, although care must be taken in the design to avoid joint breakage. The "seal" cut provides the best seal but is more expensive.

Pressure in the cylinder acts on the piston rings, and assuming that the ring does some sealing, there will be a pressure drop from one side of the ring to the other. This pressure difference results in a net "pressure-induced force" holding the ring against the side of the piston groove and outward against the cylinder bore (refer to Fig. 9.44).

Figs. 9.45–9.47 provide some typical dimension ranges for piston rings and piston clearance. The latter is governed mainly by the coefficient of thermal expansion of the piston material. Generally, the ring should not protrude from the piston groove by $>25\%$ of its thickness.

Rider rings and piston rings are almost always of the same material. Rider rings must be designed so that they do not act as a piston ring. Otherwise, wear will occur too rapidly. Solid rider rings are not prone to outward expansion but cut rider rings must be vented with holes or slots to bleed off pressure. Figs. 9.48 and 9.49 are examples of typical thickness for solid and cut rider rings versus cylinder diameter.

Rider ring width is determined by the bearing pressure. Fig. 9.50 shows piston ring and rider ring arrangements on the rider ring. The bearing pressure is generally limited to 5 psi for PTFE in nonlube services and 10 psi for lubricated cylinders (see API 618). These pressures are based on the weight of the piston plus one half the weight of the rod divided by 0.87DW (where D is the piston diameter and W is the width of all rider rings on the piston).

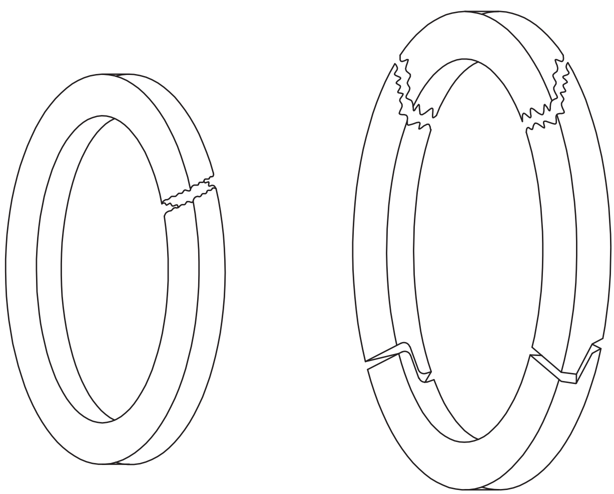


Fig. 9.42 Metal piston ring configurations.

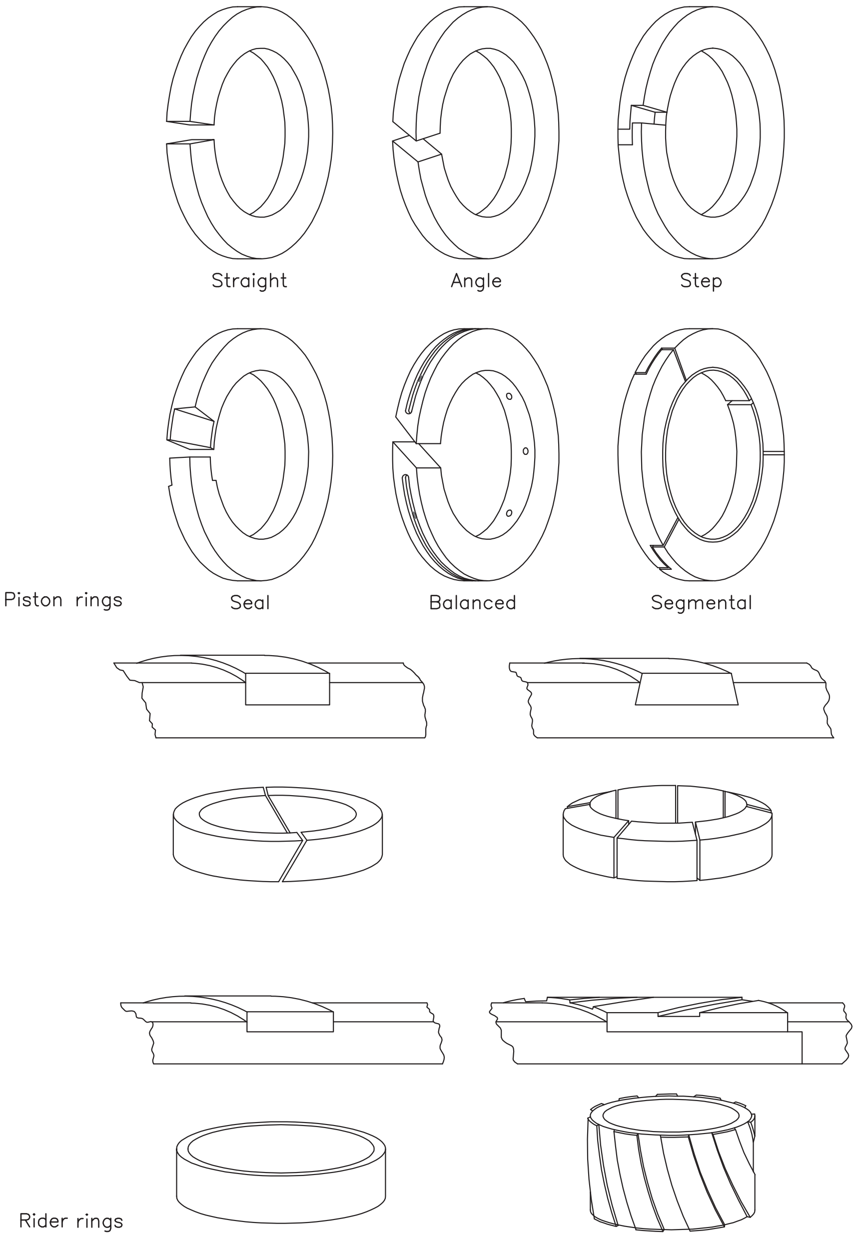


Fig. 9.43 Piston rings and rider rings.
Courtesy of the Cooper Cameron Corporation.

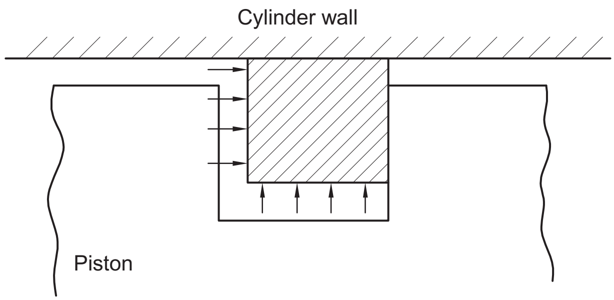


Fig. 9.44 Pressure-induced forces acting on a typical compressor piston ring.
Courtesy of American Society of Mechanical Engineers (ASME).

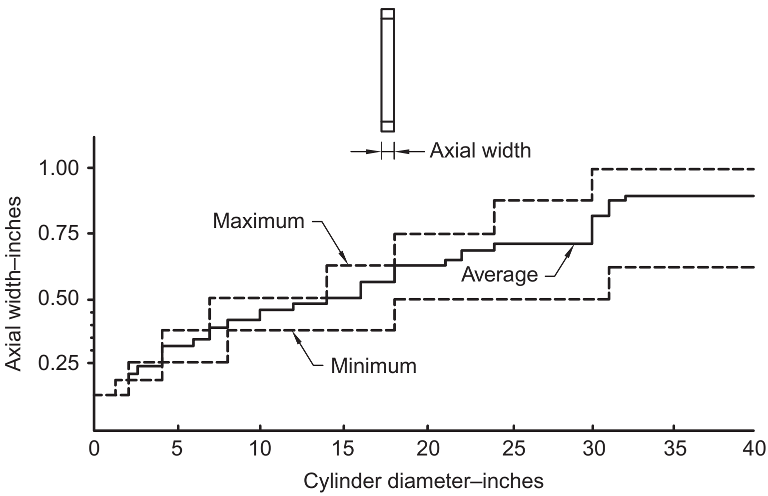


Fig. 9.45 Compilation compression ring—axial width.
Courtesy of American Society of Mechanical Engineers (ASME).

9.3.6.3 Packing

Packing provides the dynamic seal surrounding the piston rod as it penetrates the cylinder. It consists of a series or rings mounted in a packing case. The case is bolted to the cylinder. The piston rod must be sealed to reduce gas leakage from inside the cylinders. This seal is called pressure packing. It is of the full-floating design so that the packing rings follow any lateral motion of the piston rod.

The packing rings work in pairs and are designed and installed for automatic compensation for wear. The amount of pressure differential one set of rings can withstand is limited. Therefore several pairs must be installed to handle typical oil field gas compression applications. Packing ring materials may be Teflon impregnated with

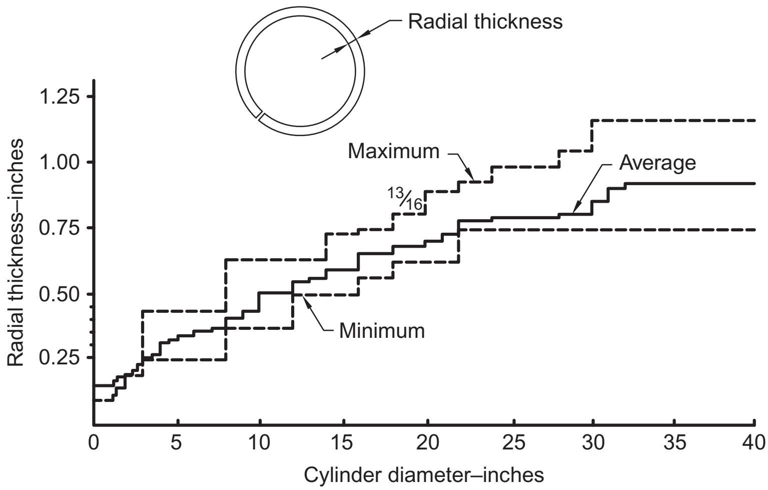


Fig. 9.46 Compilation compression ring—radial thickness.
Courtesy of American Society of Mechanical engineers (ASME).

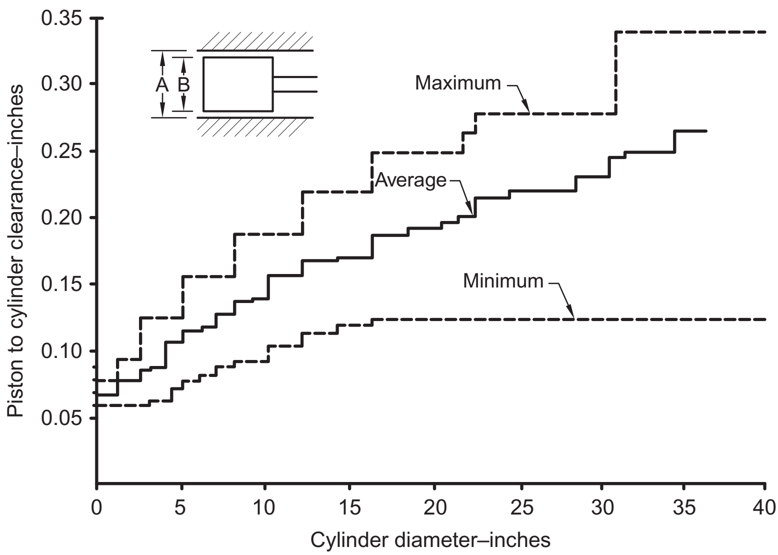


Fig. 9.47 Compilation piston to cylinder clearance—A-B.
Courtesy of American Society of Mechanical Engineers (ASME).

fiberglass, carbon fibers, or soft metal (brass, etc.); each pair of rings consists of one radial cut ring and one tangential cut ring as shown in Fig. 9.51.

The radial cut ring is installed toward the cylinder (pressure) side. Gas flows around the front face of the radial cut ring and then around the OD of both rings. Since the ring

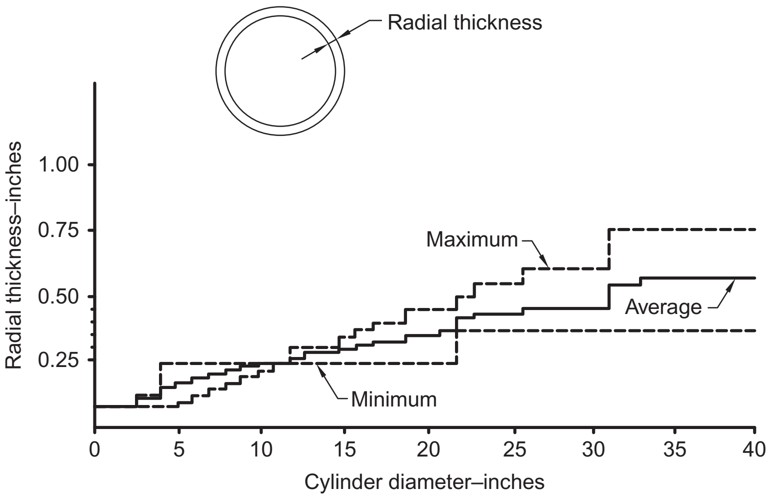


Fig. 9.48 Compilation band-type (solid) rider rings radial thickness.
Courtesy of American Society of Mechanical engineers (ASME).

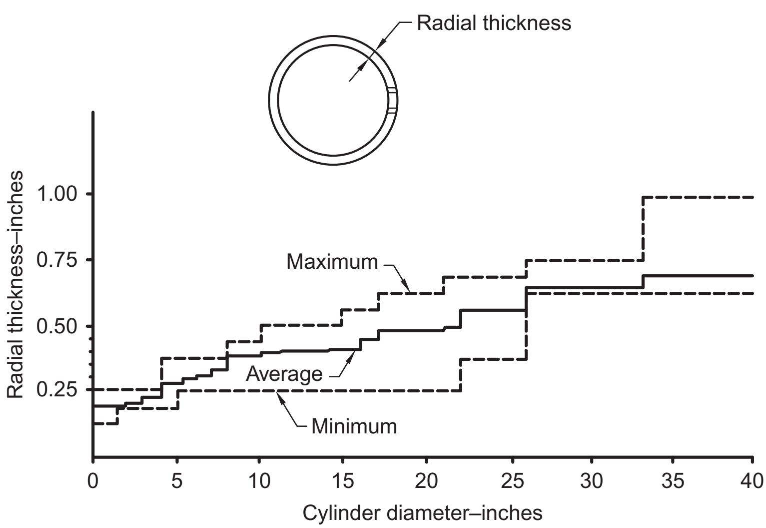


Fig. 9.49 Compilation JOINED (CUT) RIDER RING RADIAL THICKNESS.
Courtesy of American Society Engineers (ASME).

OD is greater than the ring ID, a squeezing force will be exerted on the rod. This seals the path between the rings and the rod. The radial cuts are positioned in the ring assembly so that they do not line up with the tangential cuts. Cylinder pressure forces the ring assembly against the packing case lip, thus preventing flow around the rings.

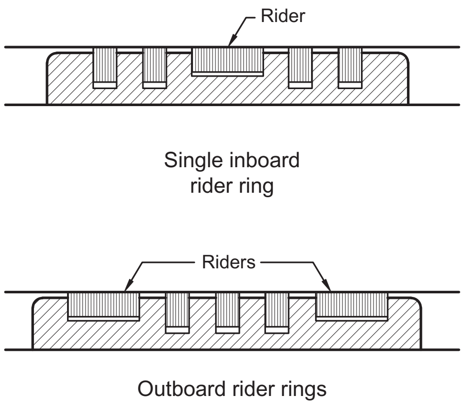


Fig. 9.50 Typical arrangements for piston and rider rings.
Courtesy of American Society of Mechanical Engineers (ASME).

As with piston rings, PTFE is used extensively for packing rings. Fig. 9.52 shows the forces on a packing ring. Fig. 9.53 shows a typical arrangement of packing rings. The backup ring limits deformation of the packing ring and is usually not required below 500 psi (3447 kPa). The backup ring is sometimes made of bronze for better heat dissipation.

Packing lubrication reduces friction and provides a cooling effect. It must be finely filtered to prevent grit from entering the case. Lubrication is generally injected in the second ring assembly, where pressure moves the oil along the shaft.

Additional cooling may be required for pressures above 3000 psi (20,684 kPa), at which a large amount of frictional power can be generated between the packing and the piston rod. Cooling may also be required at high compression ratios and where long packing cases are installed. Fig. 9.54 shows a packing case with passages for coolant.

Packing cases with vent and buffer arrangements are shown in the Appendix of API 618.

9.3.7 Bearings

The majority of bearings used in oilfield applications are hydrodynamic lubricated (journal) bearings. As shown in Fig. 9.55, oil enters into the bearing from supply holes strategically placed along the bearing circumference. The oil then flows axially and circumferentially along the bearing and out the ends. As the oil flows through the bearing, the load compresses the oil film, which generates a high pressure within the bearing supporting the load. For the crosshead shoe the principle is identical, except the motion is linear and oscillating. The bearing locations in a reciprocating compressor are as follows:

- *Main bearings*: Located between crankshaft and frame.
- *Crank pin bearing*: Located between crankshaft and connecting rod.

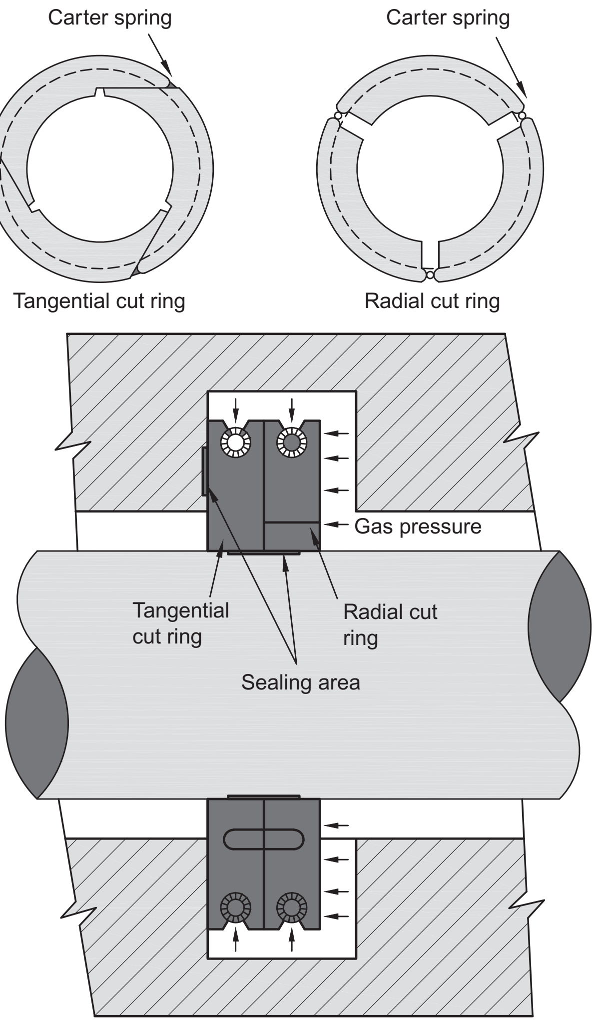
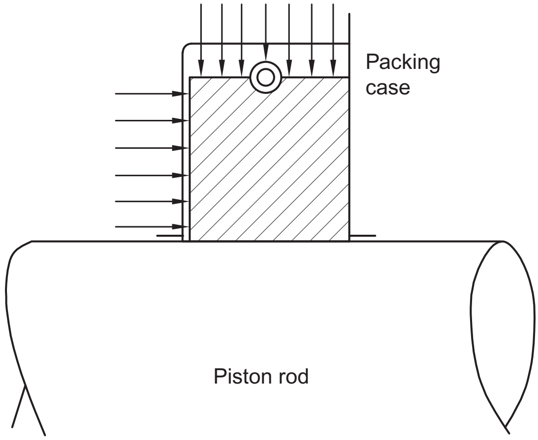


Fig. 9.51 Sectional view of the packing arrangement.
Courtesy of Dresser-Rand.

- *Wrist pin bearing*: Located between connecting rod and crosshead.
- *Crosshead bearing (shoe)*: Located underneath the crosshead (refer to Fig. 9.56).
- *Sleeve bearings*: Both ends of the connecting rod are equipped with heavy-duty sleeve bearing that can be adjusted by adjusting bolts (refer to Fig. 9.57).

Main and connecting rod bearings for the larger heavy-duty frames are split-sleeve precision insert type. The most common materials are cast iron/Babbitt. Occasionally, aluminum bearings are used. Aluminum bearings require better oil filtration as they are sensitive to dirt.

Fig. 9.52 Pressure-induced forces acting on a typical ring.
Courtesy of American Society of Engineers (ASME).



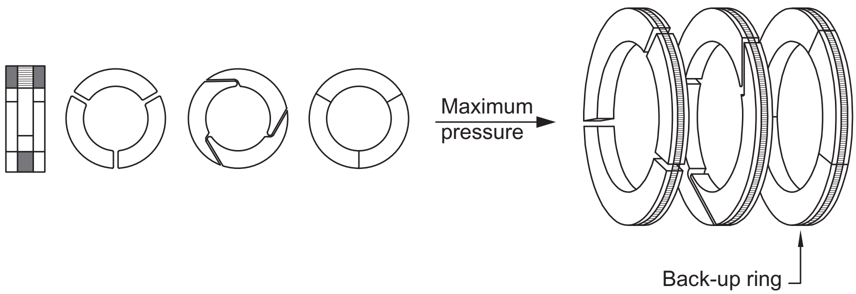


Fig. 9.53 Typical radial cut and tangent cut packing sets with back-up ring.
Courtesy of American Society of Mechanical Engineers (ASME).

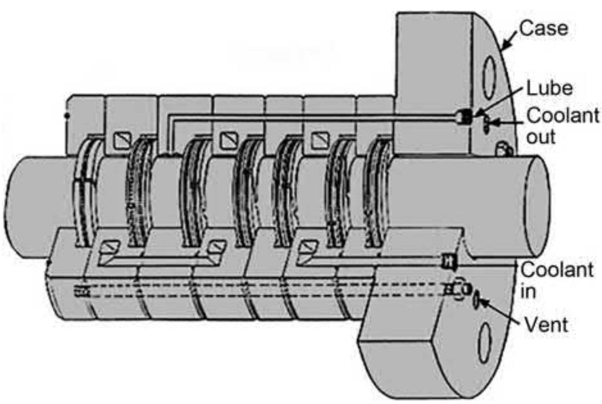


Fig. 9.54 Pressure packing case with coolant passages.
Courtesy of C. Lee Cook, Dover Resources Company.

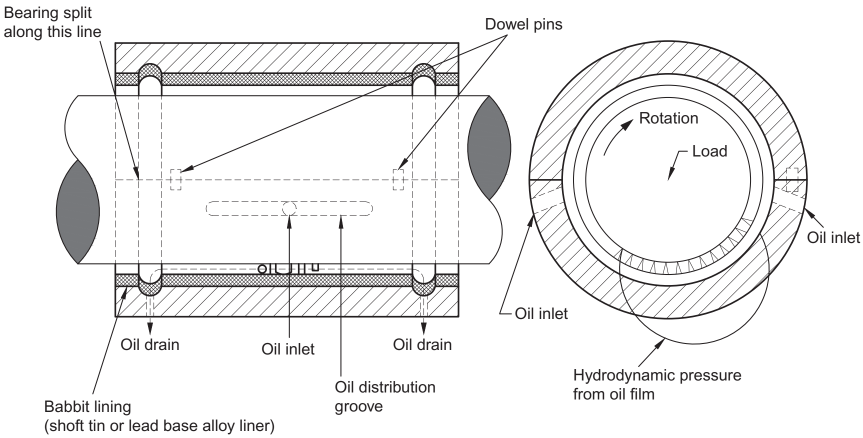


Fig. 9.55 Schematic diagram of a journal bearing.

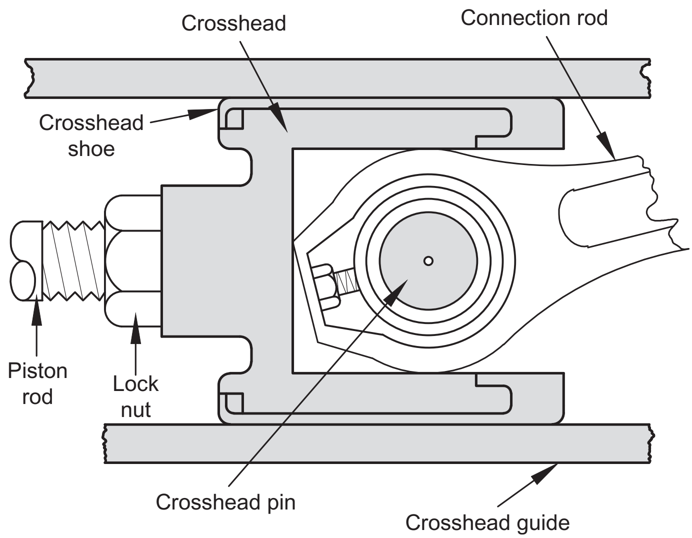


Fig. 9.56 Schematic showing the crosshead bearing.

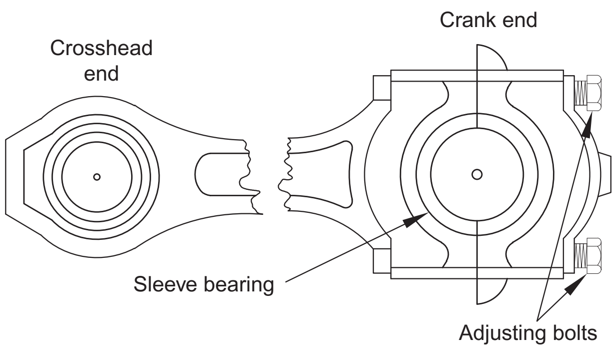


Fig. 9.57 Schematic showing sleeve bearings on connecting rod.

Although sleeve bearings are the most common, API 618 allows rolling-element (antifriction) main bearings for machines up to a rating of 200 HP. Rolling-element bearings are used in machines for ratings in excess of 1000 HP.

9.3.8 Compressor valves

9.3.8.1 General considerations

Valve devices are used to permit unrestricted flow of gas in one direction but blocks all flow in the opposite direction. Each operating end of a cylinder must have two valves or sets of valves—one to admit gas before compression, and the other to discharge gas after compression. Valves are used over a wide range of conditions. The gas may be clean or dirty, wet or dirty or corrosive. The gas may be light as hydrocarbon or as heavy as Freon. Compression ratios may be as low as 1.1 or as high as 8.0 or higher.

Valves are highly stressed wearing parts that account for the majority of compressor downtime. Even though reciprocating compressors represent a small percentage (typically <5%) of the machinery within a facility they account for a larger percentage (typically >15%) of a facilities maintenance budget as significant percentage of the reciprocating compressor maintenance is attributable to compressor valves.

Although liquid, dirt, or other process contaminants often cause valve failures, design factors are often a major contributor. In addition, valve design can also reduce the effects of contamination in some cases. Valve life in some services has been as short as 5 days when the wrong combination of valve lift and materials was specified.

Even though new materials and a better understanding of valve dynamics have greatly improved reliability, valves continue to have a major impact on overall compressor availability. For example, a large Indonesian gas plant's compressor was shut down for valve replacement and it reduced the facility's production rate by 60%, which cost $1500,000 (2017 dollars) in lost production. Even small improvements in valve life which postpone valve repairs can have a significant impact on a plant's profitability.

9.3.8.2 Valve types

Compressor manufacturers are normally responsible for valve design. There are many types of compressor valves. Almost all are spring loaded and gas actuated. As stated earlier compressor valves control the flow of gas into and out of the compressor cylinder. There are three primary types of valves:

- Plate (includes ring and ported plate).
- Channel.
- Poppet.

Valves are composed of these components:

- Seat.
- Guard.
- Valve sealing elements.
- Springs.
- Retaining hardware.

The three types of valves are described as follows:

9.3.8.2.1 Plate valves

Figs. 9.58 and 9.59 show a typical "ring" plate valve. This type of valve is actuated by unbalanced pressures on either side of the valve. The valve plates or elements are held against the ports in the valve seat by spring force. The gas pressure overcomes the spring force, the elements lift away from their seats and stop against the guard, opening the valve.

The "ring" type plate valve can be fitted with plastic elements, which is an advantage in corrosive services. The concentric ring valve can be used over the widest range of compressor applications and can withstand the most extreme operating conditions. Concentric ring valves have been used for pressures as high as 60,000 psi (413,685 kPa) with differential pressures >10,000 psi (8948 kPa) and temperatures in excess of 500°F (260°C).

Ported plate valves are very similar to concentric ring valves except that the individual rings are joined to form one or two larger plates. Their chief advantage is ease of manufacturer and simpler assembly. Ported plate valves are shown in Fig. 9.60.

9.3.8.2.2 Channel valves

Channel valves (Fig. 9.61) are the least expensive and are common for standard applications. Channel valves use channel-shaped plates instead of flat plates. Above each channel is a bowed, steel tension spring. The springs hold the channel over the slots in the lower portion of the valve. As pressure increases under the valve, the channels lift and allow flow up through the valve. A disadvantage is that they cannot be used with plastic elements and are thus not very tolerant of dirt or liquids in the gas stream. They have good flow areas and are relatively inexpensive. Channel valves have been used for pressures as high as 1500 psi (10,342 kPa) with maximum differential pressure of 500 psi (3447 kPa) and maximum temperatures of 350°F (177°C). Strip valves can be used in compressors with speeds up to 1800 rpm.

Fig. 9.58 Cutaway of an Ariel plate valve.
Courtesy of Ariel Corporation.

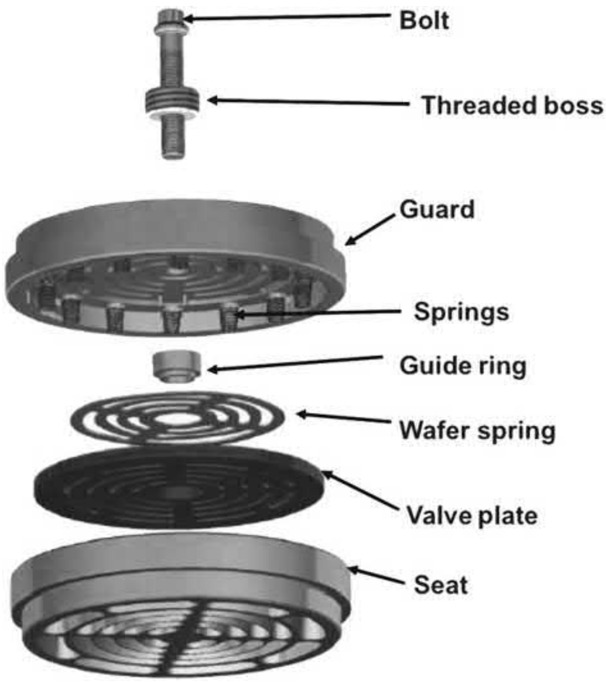


Fig. 9.59 Components of an Ariel plate valve photograph of an Ariel plate valve. Courtesy of Ariel Corporation.

Fig. 9.60 Ported plate valve. Courtesy of Dresser-Rand.

9.3.8.2.3 Poppet valves

Poppet valves (Figs. 9.62 and 9.63) are the most expensive but are also the most efficient. They have an effective lift area approximately 50% greater than that provided by the same size concentric ring valve. Poppet valves can operate with lifts as great as ¼ inch and are used extensively in natural gas pipeline booster applications. As the

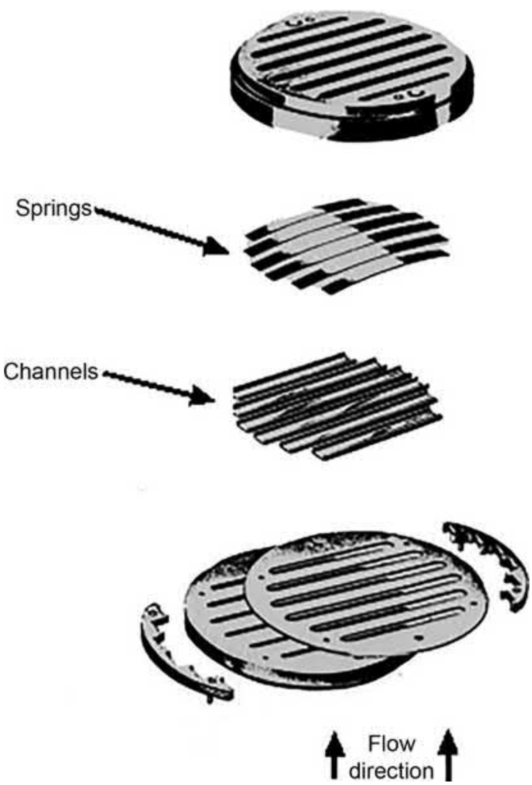


Fig. 9.61 Example of a channel valve assembly.
Courtesy of Dresser-Rand.

Fig. 9.62 Poppet valve.

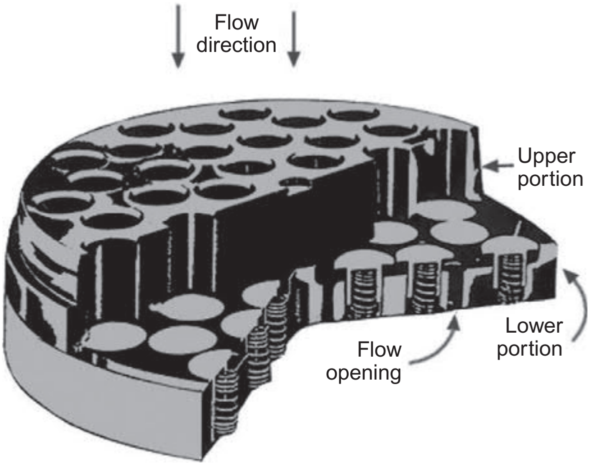


Fig. 9.63 Schematic of poppet valve assembly. Courtesy of Cooper Industries Energy Services Group.

pressure builds above each of the individual poppets, they open, allowing gas to pass through the openings in the lower portion. The greater the distance between the upper and lower portions, the smaller the resistance to flow.

Poppet valves utilize a mushroom-shaped element made from a variety of materials. The sealing element material determines the range of application. Valves with metallic poppets can withstand pressures up to 3000 psi (20,684 kPa) and temperatures up to 500°F (260°C). However, metallic poppets are seldom used due to inertial effects. Nonmetallic poppets are limited to 450°F (232°C) and 800 psi (5516 kPa), with speeds up to 1800 rpm. Nylon, Torlon, and PEEK are used for the poppet material due to their lightweight and conformability to the valve seat.

Figs. 9.64 and 9.65 show the Dresser-Rand "Magnum" valve. Unlike other poppet valves the Magnum can be used at high compressor speeds. The valve's unique

Fig. 9.64 Cutaway of a dresser-rand "magnum" poppet type valve.

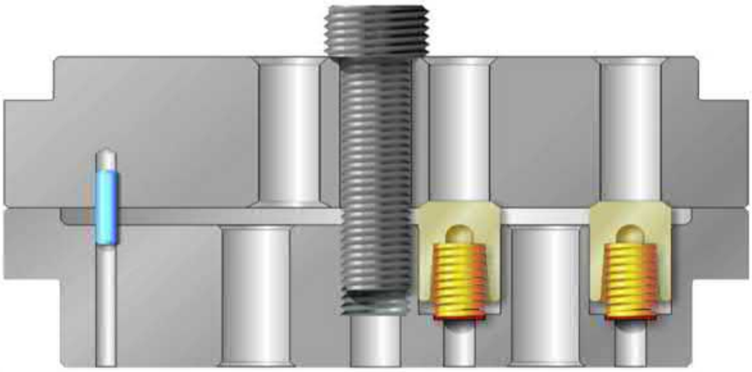


Fig. 9.65 Sectional view of a dresser-rand "magnum poppet type valve.

element design minimizes tensile stresses. The streamlined flow path with optimized seat, guard and lift areas, maximizes valve flow area and is more tolerant of particles and liquids in the gas. The valve is specifically designed for high molecular weight applications at both low and high compressor speeds.

9.3.8.3 Valve materials

Table 9.1 lists typical guard and seat materials for compressor valves.

Table 9.2 lists typical valve plate materials.

9.3.8.4 Valve velocity

Valve types and sizes are normally specified by the compressor manufacturer. An important parameter in selecting valves is the average velocity, which can be calculated from:

Field unit

$$V_g = \frac{288\mathrm{PD}}{A} \tag{9.32a}$$

Metric unit

$$V_g = 556\frac{\mathrm{PD}}{A} \tag{9.32b}$$

where:

V_g = average gas velocity, m/s (ft/min)
PD = piston displacement, m^3/h (ACFM)
A = the total product of the actual lift in service and the valve opening for all inlet valve per cylinder, mm^2 (in^2)

Table 9.1 Typical guard and seat materials

Material	Application
1141	Light duty noncorrosive service
Heat Treated 1141	Light to medium noncorrosive service
Ductile Iron	Light to medium service—resistance to some chemical attack
4140	Medium to high strength—resistance to some chemical attack
Heat Treated 4140	High strength service—resistance to some chemical attack
400 Series Stainless Steel	Corrosive service
300 Series Stainless Steel	Extreme corrosive service
17-4 PH Stainless Steel	

Table 9.2 Typical valve plate material

Material	Application
Glass-Filled Nylon Thermoplastic	Good impact and corrosion resistance 270°F temperature limit
Peek—Polyetheretherketone	High strength—high temperature (up to 375°F)
Linen-Based Phenolic Laminate	Clean gas: low compression ratios 225°F temperature limit
Laminated Cloth-Based Phenolic	High-temperature applications up to 400°F—available for ported plate application only
410 Stainless Steel	Moderate corrosion resistance Good Impact resistance
17-7 pH Stainless Steel	Moderate corrosion resistance Good impact resistance
Inconel X-750	High corrosion resistance and high strength properties in high-temperature applications

At lower velocities, valves have less pressure drop, and thus have less associated maintenance. Velocities calculated from this equation can be used to compare compressor designs.

9.3.9 Capacity control devices

The capacity of a reciprocating compressor cylinder may be controlled on the basis of suction pressure, discharge pressure, flow, or a combination of these variables. Capacity can be adjusted using:

- Inlet valve unloaders.
- Clearance pockets.
- Varying speed.

9.3.9.1 Inlet valve unloader

An inlet valve unloader holds the inlet valves open during both the suction and discharge strokes, so that all the gas is pushed back through the intake valves on the discharge stroke. Valve unloaders can be either plug or depressor type. Plug valve unloaders consist of an external plug valve that separates the cylinder from the suction side. Plug valves can be operated automatically or manually. Fig. 9.66 shows a plug-type valve unloader.

Depressor valve unloaders consist of a piston connected to a number of fingers or valve depressors. When activated, this piston is forced downward, forcing the depressors downward until the valve is lifted off its seat; Fig. 9.67 shows a depressor valve unloader. This type of valve unloader is actuated hydraulically or pneumatically. Care must be taken to ensure "fail-safe" operation in the event of control system failure.

The number of cylinders deactivated at any operating condition should be minimized, since heating and vibration problems could occur. When a cylinder or a cylinder end is deactivated, the pulsation levels in the piping system can increase significantly. Unloaded conditions must be considered during the acoustic study of the piping system.

9.3.9.2 Clearance pockets

Clearance (the volume of the cylinder not swept by the piston) can be adjusted by changing fixed clearance or by the use of clearance pockets. The greater the clearance, the lower the volumetric efficiency and the lower the capacity of the cylinder. End clearance is required to keep the piston from striking the compressor head or crank end. It is usually 3/32 in (2.38 mm).

Some small clearance is also required under suction and discharge valves so that the valves can be removed and reinstalled. These clearances are called fixed clearances and can be adjusted by:

- Removing a small portion of the end(s) of the compressor piston.
- Shortening the projection of the cylinder heads into the cylinder.
- Installing spacer rings between cylinder head and body or under valves.
- Changing from single-deck to double-deck valves. Single-deck valves reduce clearance.

9.3.9.3 Normal head-end clearance (Fig. 9.68)

Normal Head-End clearance volume is the space within the cylinder where gas can be trapped during a stroke of the piston. This space is filled with gas which contributes to the reexpansion part of the stroke and affects compressor capacity. Clearance volumes includes:

- The ports of the suction valve guards and the discharge valve seats
- The space between the face of the piston and the head
- The space between the inside diameter of the cylinder and the outside diameter of the piston from the faces of the ends back to the piston rings.

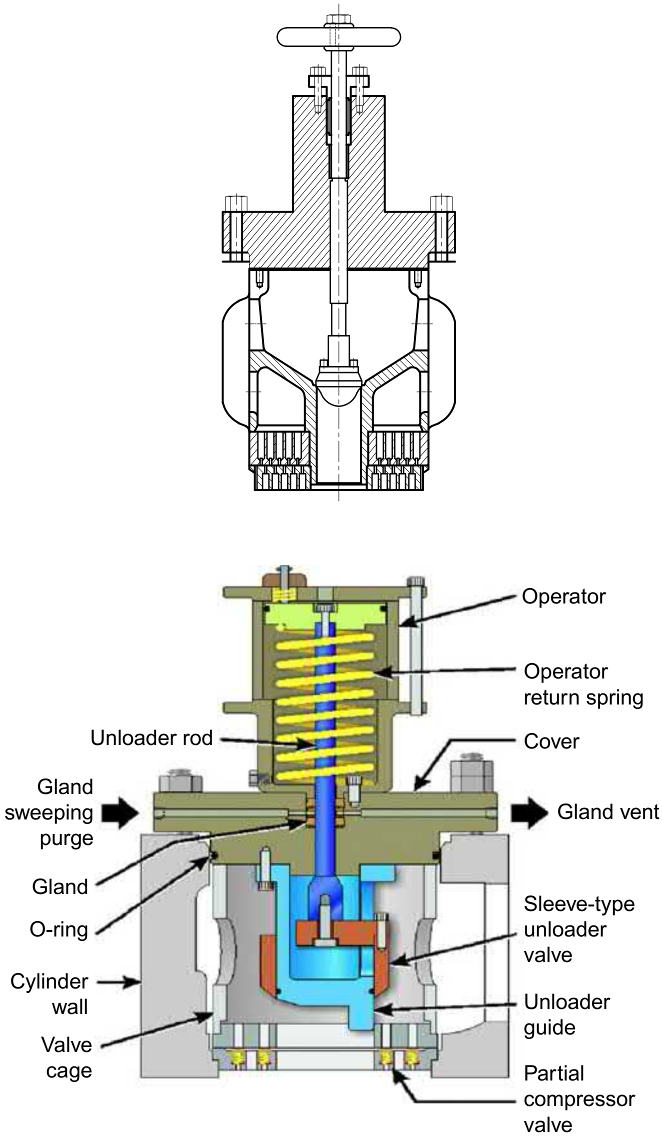


Fig. 9.66 *Top*: Sectional of a plug-type suction valve unloader—manual operation; *Bottom*: Sectional view of a Plug-type pneumatic unloader.
Top: Courtesy of Cooper Industries Energy Services Group.

9.3.9.3.1 Normal crank-end clearance (Fig. 9.69)

Normal Crank-End clearance is similar to head-end clearance except for the following factors:

- The space that the piston rod occupies reduces the swept volume in the crank end of the cylinder
- Additional space exists around the piston rod to the packing rings

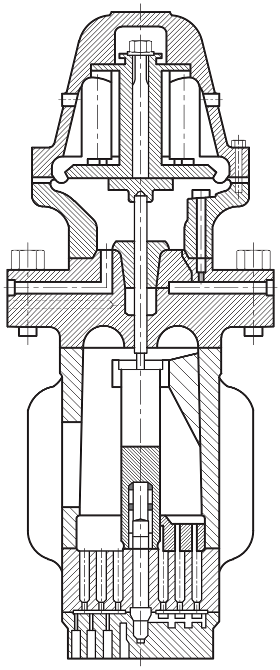


Fig. 9.67 Sectional view of a depressor type valve unloader. Courtesy of Cooper Industries Energy Services Group.

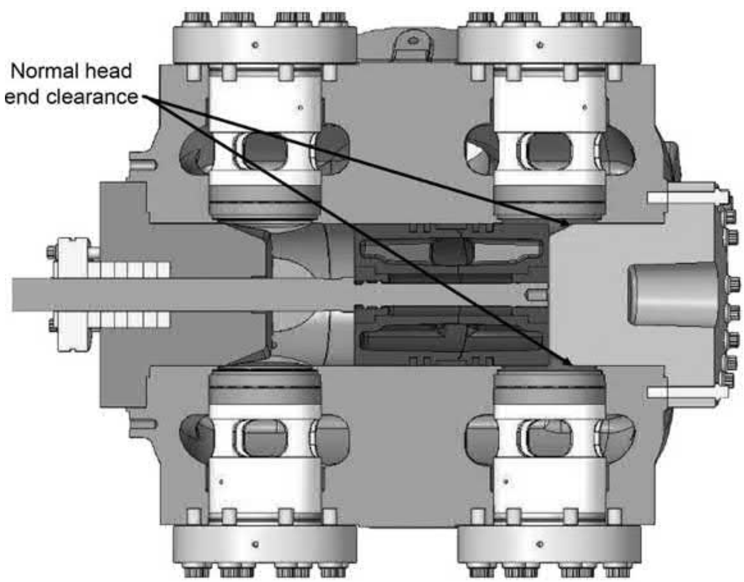


Fig. 9.68 Sectional view illustrating the normal head-end clearance.

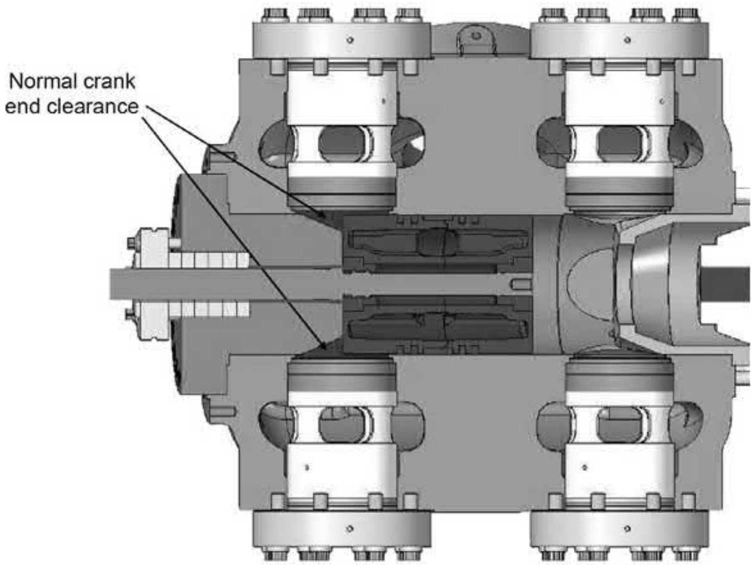


Fig. 9.69 Sectional view illustrating the normal crank-end clearance.

Clearance can also be changed by attaching fixed or variable pockets to the cylinder. Fig. 9.70 is an example of a fixed clearance pocket mounted on the cylinder. This type is separated from the cylinder by a valve that can be opened and closed from the outside allowing flow into a fixed volume chamber. Several of these can be installed on the same cylinder so that different combinations can be used to achieve the desired clearance.

Some fixed clearance pockets do not have valves. To change the performance of the cylinder the clearance can be changed by shutting down the compressor, unbolting one pocket, and installing another pocket with a different volume. In this way it is easy to add or subtract clearance if the cylinder is set up to receive clearance pockets.

More flexibility can be obtained with a variable-volume clearance pocket (VVCP) such as that shown in Fig. 9.71. This pocket is a plug built into the cylinder wall, and it forms part of the cylinder boundary. When the plug is moved, the volume within the cylinder changes. Variable clearance pockets are generally limited to the head end of the cylinder and are not normally found on the frame (or crank) end. VVCPs are manually operated and may be adjusted while the compressor is running. To adjust the VVCP for optimum compressor operation (Fig. 9.72):

- Loosen the locking handle.
- Turn the adjusting screw clockwise (CW) for less clearance and counterclockwise (CCW) for more clearance.
- Retighten the locking handle.

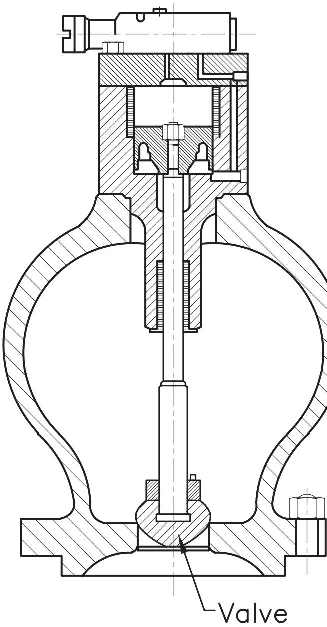


Fig. 9.70 Sectional view of a fixed clearance pocket.
Courtesy of Cooper Industries Energy Services Group.

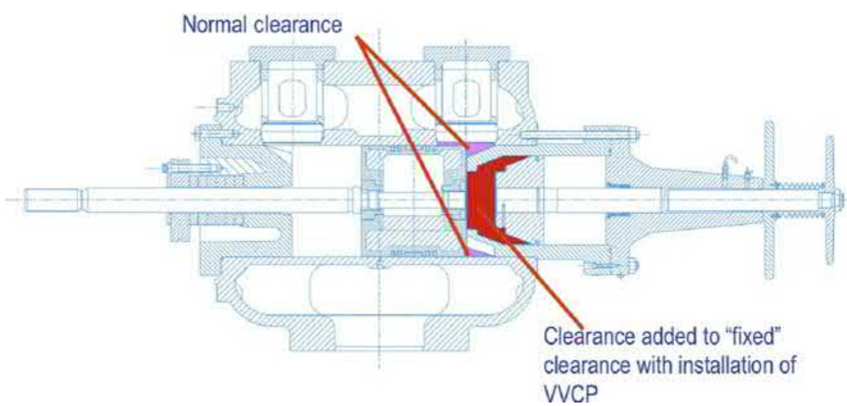


Fig. 9.71 Sectional view of a compressor cylinder with a variable volume clearance pocket.

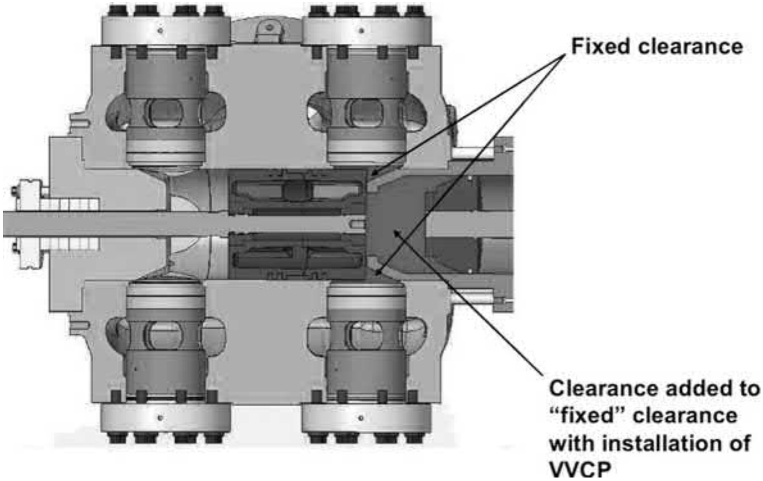


Fig. 9.72 Sectional view of a variable volume clearance pocket.

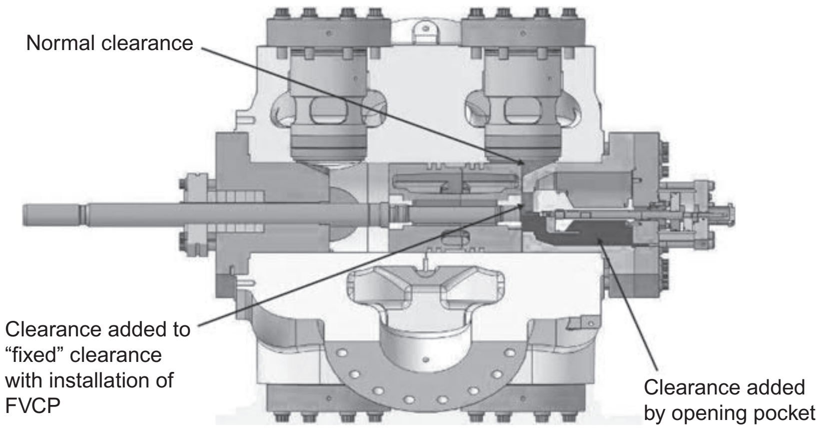


Fig. 9.73 Sectional view of a pneumatic fixed volume clearance pocket.

Fixed volume clearance pockets (FVCP) are either open or closed and can be operated from a remote location. Refer to Figs. 9.73 and 9.74.

9.3.9.4 Variable speed

Whenever possible, constant-speed operation is recommended in order to avoid possible excitation of torsional or acoustical resonances. This is especially important on complex compressors with three or more stages, or when the unit is installed offshore.

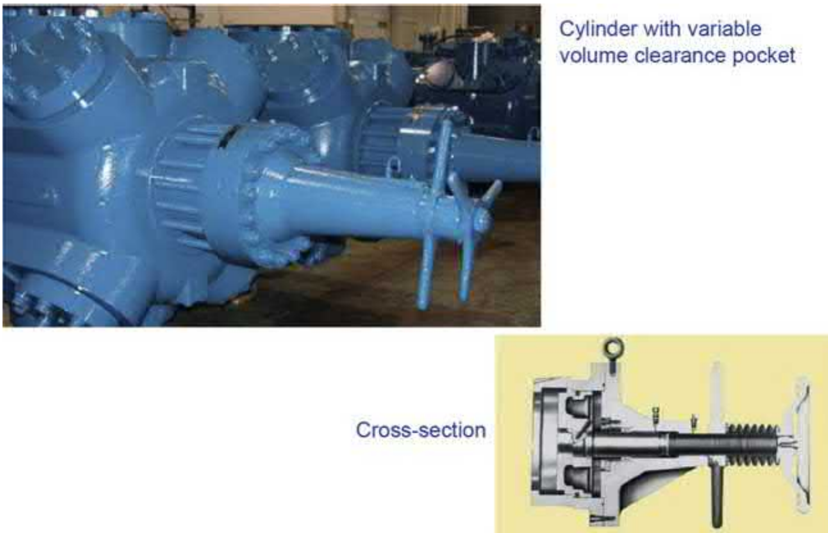


Fig. 9.74 Compressor cylinder with variable volume clearance pocket.

Acoustical simulation is difficult to perform with variable-speed machines, as does torsional vibration analysis.

9.4 Pulsation and vibration

9.4.1 Overview

Acoustical pulsations occur in reciprocating compressors because suction and discharge valves are open during only part of the stroke, requiring the gas in the piping and associated equipment leading to and away from the cylinder to accelerate and decelerate. Pulsations can represent a high amplitude cyclic excitation force in a mechanical system such as vessels and/or piping. As a piston moves back and forth, it generates pressure and volume fluctuations, called pulsations. This is known as sinusoidal wave phenomena. Pulsation waves travel at the speed of sound in the compressed fluid and reflect back toward the source when they hit a significant change in cross-sectional area in the flow path. The reflected waves and outbound waves are vectorially additive. Pulsation waves are potentially harmful when the acoustic length of the system is "tuned" such that the peak amplitudes of outbound and reflective waves occur at the same time. This is called acoustic resonance.

High stresses may result. In addition, the mechanical system may vibrate excessively when its natural frequency matches some multiple of the excitation frequency.

When this happens, the system may vibrate in response and high stresses may result, high enough to cause a mechanical failure. The highest stresses will usually occur at a vessel nozzle weld, at an elbow weld, or at some other location where there is a major change in stiffness.

9.4.2 Pulsation reduction

A three-pronged approach may be taken in the design of a facility utilizing reciprocating compressors.

1. Large volumes called pulsation bottles are placed at the suction and discharge nozzles of each stage to reduce the amplitude of pulsations on the piping side of the bottles.
2. An analog or digital acoustic study may be performed to acoustically model the system. This allows the designer to select an acoustic filter to alter the frequency of pulsations, or to alter the acoustic length of various portions of the system, or both.
3. A mechanical study may be performed to model the mechanical response of the system. This study allows the designer to alter the mechanical stiffness of the system so that the natural frequency can be detuned from any multiple of the acoustic excitation frequency. These studies are often collectively referred to as an "analog" study.

9.4.2.1 Pulsation dampeners

The most common way to minimize pulsations is to install volume bottles (sometimes called pulsation chambers, surge drums, etc.) mounted directly on or very close to the suction and discharge nozzles of the compressor cylinder. Sometimes a large diameter pipe joining the connections of two or more compressor cylinders operating in parallel can serve as a volume bottle. Often it is necessary to install a choke tube between two volume bottles to adequately dampen high frequency pulsations.

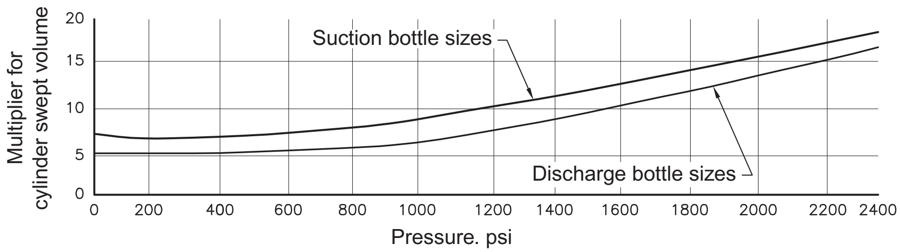
Volume bottles generally work satisfactory; however, in critical installations it is necessary to analyze the pressure pulsations using digital or analog computers. However, for most low power (<1000 HP or 746 kW), high-speed (>900 rpm) applications, most manufacturers have sufficient operating experience to be able to size the volume bottles using empirical techniques. One empirical method that can be used to provide a reasonable rule of thumb for a volume bottle is to provide a volume that is a multiple of the swept volume of the cylinder. Fig. 9.75 can be used to obtain the multiplier. The swept volume is given by:

For single-acting cylinders—Head end:

$$\mathrm{SV_{HE}} = 0.785\ d_c^2(s) \tag{9.33}$$

For single-acting cylinders—Crank end:

$$\mathrm{SV_{CE}} = 0.785\ \left(d_c^2 - d_r^2\right)(s) \tag{9.34}$$



Note: Suction bottle sizes correlate to suction pressures
discharge bottle sizes correlate to suction pressures.

Example: P_s = 400 psi
Bottle size = 7 x PD

Fig. 9.75 Approximate bottle sizing chart.
Courtesy of GPSA.

For double-acting cylinders (sum head end and crank end)

$$SV_{HE+CE} = 0.785\left(2d_c^2 - d_r^2\right)(s) \tag{9.35}$$

where:

SV_{CE} = swept volume, crank end, mm^3 (in^3)
SV_{HE} = swept volume, head end, mm^3 (in^3)
SV_{HE+CE} = swept volume, head end and crank end, mm^3 (in^3)
d_c = diameter of cylinder, mm (in)
s = stroke length, mm (in)
d_r = diameter of rod, mm (in)

When more than one cylinder is connected to the bottle the sum of the individual swept volumes is the size required for the common bottle.

For a specified volume there are an infinite number of choices of cylinder diameter and length that could be chosen. It is desirable to make the length as short as possible. However, the shorter the length, the larger the inside diameter. For a given design pressure, the larger the diameter, the thicker the wall thickness. Thus a large diameter bottle tends to cost more than a smaller diameter bottle containing the same volume. One rule of thumb is to make the volume bottle diameter 1.5 times the diameter of the largest size cylinder to which it is connected. Often space and piping limitations determine the dimensions of the volume bottle.

Once either a diameter or length is selected, the other can be calculated from Eqs. (9.33)–(9.35) substituting the required volume for SV, the inside diameter of the vessel for d_c, and the seam-to-seam length of the vessel for S. Welding caps are often used on the ends of volume bottles. Table 9.3 presents the volume contained in these caps. Often this volume is neglected in computing the required inside diameter and seam-to-seam length of the volume bottle.

Table 9.3 (A) Oilfield units: welding caps. (B) SI units: welding caps

(A)

	Standard weight		Extra strong		Double extra strong	
Pipe size	**Volume**	**Length**	**Volume**	**Pipe size**	**Volume**	**Length**
in	**in^3**	**in**	**in^3**	**in**	**in^3**	**in**
4	24.2	2 1/2	20	2 1/2	15	3
6	77.3	3 1/2	65.7	3 1/2	48	4
8	148.5	4 11/16	122.3	4 11/16	120	5
10	295.6	5 3/4	264.6	5 3/4		
12	517	6 7/8	475	6 7/8		
14	684.6	7 13/16	640	7 13/16		
16	967.6	9	911	9		
18	1432.6	10 1/16	1363	10 1/16		
20	2026.4	11 1/4	1938	11 1/4		
24	3451	13 7/16	3313	13 7/16		

(B)

	Standard weight		Extra strong		Double extra strong	
Pipe size	**Volume**	**Length**	**Volume**	**Length**	**Volume**	**Length**
mm	**10^5 mm^3**	**mm**	**10^5 mm^3**	**mm**	**10^5 mm^3**	**mm**
101.6	3.965	63.5	3.277	63.5	2.458	76.2
152.4	12.667	88.9	10.766	88.9	7.866	101.6
203.2	24.335	119.1	20.041	119.1	19.664	127
254	48.44	146	43.327	146		
304.8	84.271	174.6	77.839	174.6		
355.6	112.186	108.4	104.877	198.4		
406.4	158.561	228.6	149.286	228.6		
457.2	234.761	255.6	223.356	255.6		
508.9	332.067	285.8	317.581	285.8		
609	565.518	341.3	542.903	341.3		

The Southern Gas Association appointed Southwest Research Institute (SWRI) to investigate installations that experienced intolerable piping vibration and failures. The result was the development of an analog computer to simulate the acoustical interaction of one or more compressors and the associated piping. At first, the analog results were analyzed only from the standpoint of pressure pulsation amplitudes and frequencies. Later, the piping systems were analyzed for mechanical interaction between the gas pulsations and piping vibration. Due to this research, in many cases pulsation dampers with proprietary internals replaced volume bottles except in low-pressure applications.

9.4.2.2 Digital/analog pulsation studies

Analog and digital simulators that can establish the pulsation performance of any compressor piping system, no matter how complex, are available. These simulators provide a more accurate sizing of volume bottles than the rule-of-thumb procedures described in the previous section. Simulators can also be used in the design of other devices such as choke tubes and orifice plates between volume bottles.

API STD 618 (Reciprocating Compressors for General Refinery Service) recommends the following design approaches for pulsation suppression within a compressor station:

- *Design Approach 1*: Oversized piping for compressors of 150 hp. and under. Otherwise, pulsation suppression devices designed using proprietary or empirical techniques. Proprietary techniques include volume bottles, choke tubes, and orifice systems. Simplified analysis may be performed to determine critical piping lengths that may be in resonance with acoustical harmonics. An acoustical piping simulation is not required.
- *Design Approach 2*: Pulsation suppression devices and piping system response control using proven acoustical simulation techniques. Includes evaluation of interaction of acoustical forces among compressor, suppression devices, and piping.
- *Design Approach 3*: Similar to Design Approach 2, with the addition of a mechanical analysis of compressor manifold and associated piping, including study of interaction between mechanical and acoustical system responses.

For the different design approaches, different levels of pressure pulsation are allowed by API STD 618.

For Design Approach 1:

Field unit

$$\mathrm{PP} = \frac{10}{P^{\frac{1}{3}}} \le 2 \tag{9.36a}$$

Metric unit

$$\mathrm{PP} = \frac{19.033}{P^{\frac{1}{3}}} \le 2 \tag{9.36b}$$

where:

PP = allowable pulsation level (peak to peak), percent of average pressure
P = average line pressure, kPa (psia)

In Eq. (9.36), "PP" is the allowable pulsation level after filtering. In addition, the unfiltered peak-to-peak pulsation level at the compressor cylinder flange is limited to 7% of the average line pressure or the level determined by:

$$P_{cf}(\text{percent}) = 3R \tag{9.37}$$

where:

P_{cf} = maximum allowable unfiltered peak-to-peak pulsation level expressed as a percentage of average absolute line pressure at the compressor cylinder flange
R = compression ratio P_d/P_s of a compressor stage

For those special cases where pulsation levels exceed these values and a reasonable increase in volume bottle size will not satisfy the requirement of 7% or the level determined by Eq. (9.37), higher limits may then be used as agreed upon by the purchaser and the vendor.

For Design Approach 2 or 3:

The most severe operating conditions should be simulated. These conditions generally are maximum flow and any single-acting cylinder condition for a compressor.

Pressure pulsation levels obtained by analog simulation should be less than or equal to those determined by analog or digital models.

Analog or digital modeling studies should be performed early in the project so that changes required to detune the piping can be accommodated. Once the simulation has been completed, design changes should be minimized.

Field unit

$$\mathrm{PP} = \frac{300}{\left(Pd_i f_p\right)^{\frac{1}{2}}} \tag{9.38a}$$

Metric unit

$$\mathrm{PP} = \frac{3970}{\left(Pd_i f_p\right)^{\frac{1}{2}}} \tag{9.38b}$$

where:

PP = allowable pulsation level (peak to peak) percent of average pressure
P = average line pressure, kPa (psia)
d_i = pipe internal diameter, mm (in)
f_p = pulsation frequency, cycles/s

The pulsation frequency peaks will be at multiples of compressor speed.

$$f_p = R_c \times \frac{N}{60} \tag{9.39}$$

where:

f_p = pulsation frequency, cycles/s
N = 1,2,3…, cycles/rev
R_c = compressor speed, rpm

In practice, many operators do not use "Analog" compressors of 1000 HP (746 kW) or lower, but rather rely on extrapolations from proven designs. For larger power sizes or where unusual conditions (e.g., unloading and loading cylinders) exist, an analog is recommended. Pressure pulsation levels are normally limited to the allowable pulsation levels shown before. Many producing locations employ analog studies (from SWRI) when purchasing new equipment. These studies are expensive, however, and may not always be appropriate. Analog studies have the disadvantage of not being very flexible. If changes in piping, vessels, or operating conditions are made after the analog study, the entire analog must be redone at additional cost.

9.4.3 Vibration

As a rule of thumb, the first-stage suction line should be sized for a pressure drop not to exceed ½ psi (3.4 kPa) or 1% of operating pressure, whichever is smaller, and a maximum actual velocity of 30 ft./s. (9 m/s). The interstage and discharge piping should be sized for a pressure drop not to exceed 2% of operating pressure or 5 psi (34.5 kPa), whichever is less, and a maximum actual velocity of 50 ft./s. (15 m/s).

To minimize pipe vibrations, it is necessary to design pipe runs so that the "*acoustic length*" of the pipe run does not create a standing wave that amplifies the pressure pulsations in the system. The acoustic length is the total overall length from end point to end point including all elbows, bends, and straight pipe runs. Typical pipe runs with respect to acoustic length are considered to be:

- Pipe length from scrubber to suction volume bottles
- Pipe length from discharge volume bottles to cooler
- Pipe length from cooler to scrubber
- Pipe length from discharge scrubber to pipeline.

The end of a pipe run can be classified as either "*open*" or "*closed*." Typically, the end of a pipe is considered "closed" when the pipe size is dramatically reduced, as at orifice plates and at short length flow nozzles. The end is considered "open" when the pipe size is dramatically increased.

When the pipe run contains similar ends (closed-closed or open-open), prohibited pipe lengths are:

$$0.5\lambda,\ \lambda,\ 1.5\lambda,\ 2\lambda, \ldots$$

where:

λ = acoustic wave length, m (ft)

When the pipe run contains dissimilar ends (closed–open or open–closed), prohibited pipe lengths are:

$$0.25\lambda,\ 0.75\lambda,\ 1.25\lambda,\ 1.75\lambda$$

The wavelength may be calculated from:

Field unit

$$\lambda = 13.382 \frac{\left(\frac{kT}{\text{MW}}\right)^{\frac{1}{2}}}{R_c} \tag{9.40a}$$

Metric unit

$$\lambda = 91.3 \frac{\left(\frac{kT}{\text{MW}}\right)^{\frac{1}{2}}}{R_c} \tag{9.40b}$$

where:

λ = acoustic wave length, m (ft)
k = ratio of gas specific heats, dimensionless
T = gas temperature, K ($^{\text{O}}$R)
MW = molecular weight of gas
R_c = compressor speed, rpm

Normally, if the pipe support spacing is kept short, the pipe is securely tied down, the support spans are not uniform in length, and fluid pulsations have been adequately dampened, then mechanical pipe vibrations will not be a problem. It is good practice to ensure that the natural frequency of all pipe spans is higher than the calculated pulsation frequency.

The pulsation frequency is given by:

Field unit

$$f_p = \left(\frac{R_c}{60}\right) n_p \tag{9.41a}$$

Metric unit

$$f_p = R_c n_p \tag{9.41b}$$

where:

f_p = pulsation frequency, cycle/s
n_p = 1 for single-acting cylinder and
= 2 for double-acting cylinder
R_c = compressor speed, rps (rpm)

The natural frequency of pipe spans can be estimated from Fig. 9.76.

For smaller high-speed compressors, the pipe sizing rule of thumb discussed before, in conjunction with pulsation bottles sized from Fig. 9.75, should be sufficient to satisfy the requirements of Eq. (9.37).

These rules of thumb can also be used for preliminary sizing of piping and bottles in preparation for an analog study.

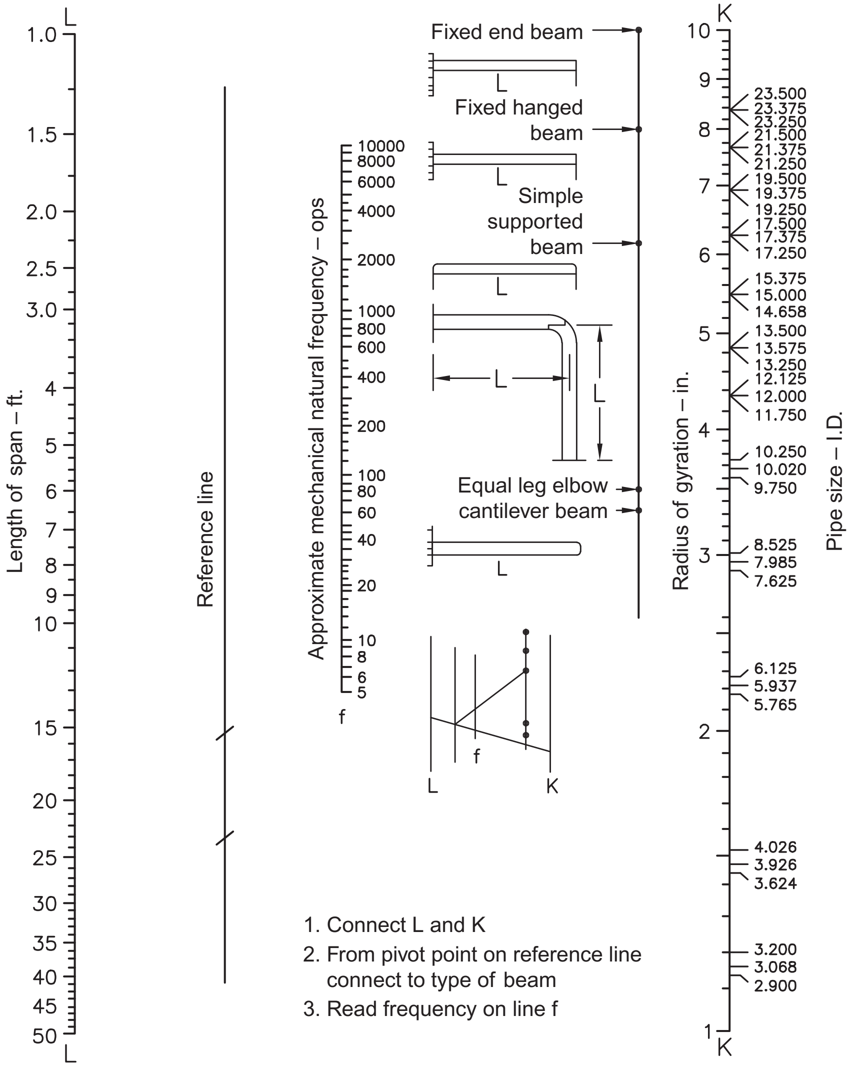


Fig. 9.76 Approximate mechanical natural frequency of uniform steel beams. Courtesy of Southwest Research Institute (SWRI).

9.5 Process piping considerations

9.5.1 General piping considerations

Reciprocating compressors are sensitive to dirt. Therefore the suction piping to each stage, from the knockout drum to the suction pulsation damper, should be “pickled” (chemically cleaned). Temporary strainers should be installed and left in place for several days during initial operation. These strainers are usually of the truncated cone or conical type and made of perforated steel plate with a double overlay of stainless steel wire mesh screen (often 100 mesh over 30 mesh). The screen is attached to the outside of the strainer basket so that the flow encounters the screen first. The strainer is typically located at the inlet to the suction pulsation damper.

Liquid is also a significant problem. Suction piping must be configured so that liquids cannot be trapped in low spots. Liquids that collect in "pockets" in the piping can "slug" the compressor causing extensive damage. The suction line immediately attached to the knockout drum (scrubber) should be slightly sloped so that liquid drains back to the vessel. Inadequacy of liquid separators is a common complaint in plants. They may be too small, unable to take slugs, or located too far from the compressor. The latter is a very common failing. This aspect of facility design should be studied very carefully. Very often the design is based on preliminary dew point calculations for a given gas, but later the composition changes. Accordingly, the design should include a generous safety factor.

Where there is any possibility of liquid condensation, the suction lines should be heat traced. A second line of defense is to include a liquid separation chamber in the pulsation damper (refer to API 618). Pulsation dampers can be heat traced with a "plate coil" for steam or with electrical heaters. Bayonet heaters can also be supplied.

The distance between the pulsation damper and the compressor cylinder should be held to a minimum.

Piping design should be analyzed to assure that forces and moments exerted on the cylinder flanges are within the vendor's tolerances. "Cold Springing" should not be allowed. (Cold springing is forcing pipe to the machine in a cold condition in order to afford relieved stresses as the pipe heats to operating temperature.) (Refer to Volume 3 for a detailed discussion.)

9.5.2 Process piping considerations

Fig. 9.77 is a generalized process flow diagram for a single reciprocating compressor. It shows the relationship of various devices that should be considered in the development of a process flow diagram for a reciprocating compressor. No one compressor needs all of the devices discussed.

Suction gas is routed first to a suction scrubber to eliminate any liquids that may condense in the suction line due to cooling and to trap liquid carryover due to failure of upstream equipment. Pulsation bottles are shown on the inlet and outlet of the compressor cylinder. A temporary strainer is shown on the inlet. Unless all lines leading to the compressor are thoroughly cleaned, on start-up some small amount of trash may be swept toward the compressor. The strainer traps this material before it can damage the compressor. The strainer should be removed after a few days of operation to avoid the possibility of a sudden strainer failure that would allow metal fragments to enter the cylinder.

As the gas is compressed, it is heated. The discharge temperature increase is determined by the ratio between suction and discharge pressure. If the compressor is a low

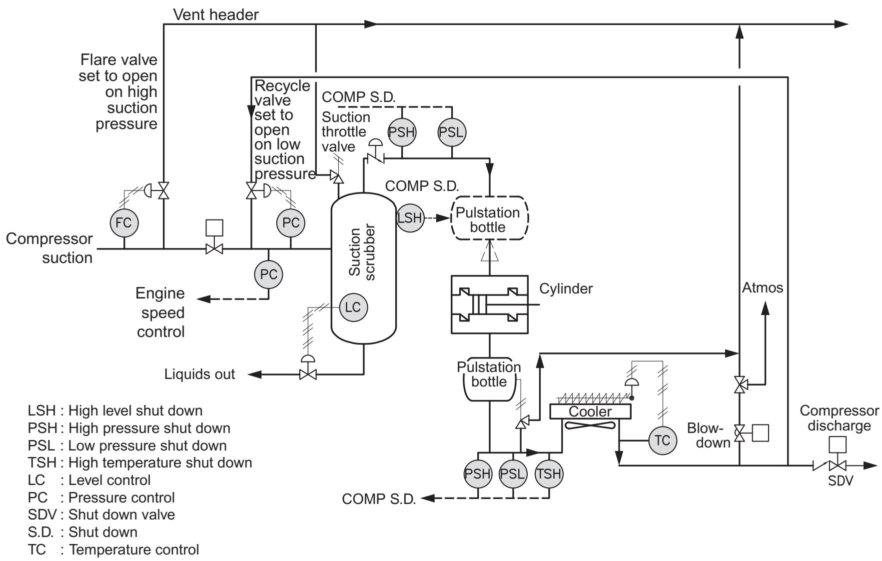


Fig. 9.77 Schematic flow diagram for a reciprocating compressor system.

ratio single-stage unit it may not require a gas cooler, but most units have a temperature increase sufficient to require cooling of the discharge gas. Cooling is normally accomplished with an aerial cooler as shown, although heat exchange with any cold media (e.g., seawater) could also be used. The discharge piping delivers the gas for sales, for injection, or for suction in another stage of compression.

9.5.2.1 Suction and discharge piping

The piping that conveys the gas to the compressor and between scrubber, cylinders, and coolers should be sized using the techniques used for Facilities Piping. In addition, to aid in reducing pressure pulsations, the guidelines discussed in this presentation should be considered.

Due to the vibrations and pulsations associated with reciprocating compressors, many operators prefer not to use threaded piping. Line sizes $<$50.8 mm (2 in) in diameter often are precluded in hydrocarbon service, and wall thicknesses consistent with Schedule 80 are required. Some operators allow 1-in and 1.5-in pipe but require it to be Schedule 160 and socket welded.

Some gas piping may be subjected to large temperature changes requiring that attention be paid to piping flexibility. Sufficient direction changes often are present in the piping design so that special flexibility studies are not required. However, for large compressors containing large diameter piping, sufficient flexibility may not be available and detailed pipe stress calculations may be required.

9.5.2.2 Suction, discharge, and blowdown valves

These valves are found on all compressors. They provide suction and discharge line isolation and the ability to depressure the unit for malfunctions, emergencies, start-up, and maintenance. The flowsheet shows automated shutdown and blowdown valves. Some operators of smaller onshore units have manually operated valves on suction, discharge, and blowdown.

9.5.2.3 Suction and discharge valves

Many operators require that these valves be automated and sequenced to open and close in a particular order upon unit shutdown or start-up. The decision to automate suction and discharge block valves is not influenced by unit size or power but rather by unit location, safety considerations, type and volume of gas being compressed, and unit investment. The decision to operate manually or automate these critical valves should be considered carefully. If the compressor is installed inside a building, suction and discharge block valves should be located outside the building.

On large compressors with large diameter actuated suction and discharge valves (or valves with high differentials across them upon opening), a smaller, automated equalizing valve is often installed. This valve equalizes upstream and downstream pressure before opening the large diameter valve. This permits the opening of valves with lower differential pressures across the valve. This practice can protect valve seats and permit use of smaller actuators in some instances.

9.5.2.4 Blowdown valves

A blowdown valve relieves trapped pressure when the compressor is shut down. Most operators use automatic blowdown valves to reduce the hazards of trapped gas. Some operators use manual blowdown valves on smaller units for ease of restarting, depressuring the unit only for maintenance. (On larger units the compressor should be started with the compressor cylinders “unloaded” or discharging to atmosphere, and thus the blowdown valve would be open during a start sequence, then closed to permit gas to discharge to line pressure.)

If the blowdown valve has been tied into a closed vent system to route gas to a safe area for flare or vent, an additional blowdown valve is required. This valve should vent directly to atmosphere, with nothing else tied into its line. This prevents backflow from the closed vent system while the compressor is down for maintenance and permits small leaks of gas from the compressor to vent safely. The flowsheet shows a manual 3-way valve that accomplishes this isolation.

9.5.2.5 Flare valve

As the flow rate to the compressor increases, the suction pressure rises until the volume of gas compressed by the cylinder at actual conditions of temperature and pressure equals the volume required by the cylinder. As the suction pressure increases, the flow rate at standard conditions increases. In most instances, the net effect is to increase the power required of the driver. A flare valve is needed to keep the suction pressure from rising too high and overloading the driver.

The flare valve also allows production to continue momentarily if a compressor shuts down automatically. Even in booster service, it may be beneficial to allow an operator to assess the cause of compressor shutdown before shutting in the wells. In flash gas or gas lift service, it is almost always beneficial to continue to produce liquids while the cause of the compressor shutdown is investigated. If the compressor cannot be restarted within a matter of minutes, the wells can then be shut in to avoid continued venting or flaring of hydrocarbon gases, as dictated by pollution and conservation regulations or by the economics of the particular installation. The flare valve must always be installed upstream of the suction shutdown valve.

9.5.2.6 Suction pressure throttle valve

A suction pressure throttling valve can also be installed to protect the compressor from too high a suction pressure. This is typically a butterfly valve or v-ball that is placed in the suction piping. As the flow rate to the compressor increases, the valve closes slightly and maintains a constant suction pressure. This automatically limits the flow rate to the cylinder to a maximum rate of gas measured at standard conditions (i.e., a maximum suction pressure) determined by compressor limitations on power.

When the suction throttle valve is limiting pressure to the cylinder, the pressure upstream of the valve will increase until sufficient backpressure is established on the wells or equipment feeding the compressor to reduce the flow to a new rate in equilibrium with that being handled by the cylinder. If this does not occur, eventually the flare valve or relief valve will open.

Suction throttle valves are commonly used in gas lift service to minimize the action of the flare valve. Flow from gas-lift wells decreases with increased backpressure. If there were no suction valve, the flare valve would have to be set at a low pressure to protect the compressor. With a suction throttle valve, it may be possible to set the flare valve at a much higher pressure (slightly below the working pressure of the low-pressure separator). The difference between the suction valve set pressure and the flare valve set pressure provides a surge volume for gas and helps even the flow to the compressor.

Suction throttle valves are also common where several compressors are operating in parallel. If the clearances in the units are set to load the compressors fully (demand maximum power available) when all compressors are running, then when one compressor goes down, the suction pressure on the remaining compressors will increase and they will become overloaded. This situation results in high maintenance costs and, in severe cases, can lead to serious accidents. Suction throttling valves automatically balance the flow to a higher suction pressure equilibrium point equal to the gas volume compressed by the remaining "on-line" units. Suction pressure throttle valves should be selected and sized carefully.

9.5.2.7 Recycle valve

At constant speed and operating conditions, a constant volume of gas (at suction conditions of pressure and temperature) will be drawn into the cylinder. As the flow rate to the compressor decreases, the suction pressure decreases until the available gas expands to satisfy the actual volume required by the cylinder. When the suction pressure decreases, the ratio per stage increases and therefore the discharge temperature increases. Low suction pressure can also cause high rod loads. In order to keep from having too high a discharge temperature or rod load, the recycle valve opens to help fill the compressor cylinder volume and maintain a minimum suction pressure.

Most gas lift, flash gas, and vapor recovery compressors require a recycle valve because of the unsteady and sometimes unpredictable nature of the flow rate. Indeed, there may be periods of time when there is no flow at all to the compressor.

9.5.2.8 Speed controller

A speed controller can help extend the operating range and efficiency of the compressor. As the flow rate increases, the compressor speed can be increased to handle the additional gas. Compressor speed will stabilize when the actual flow rate to be compressed equals the required flow rate for the cylinder at the preset suction pressure. As the flow rate decreases, the compressor slows down until the preset suction pressure is maintained.

A speed controller (or for that matter, any capacity control device) does not eliminate the need for a recycle valve, flare valve, or suction throttling valve, but it does minimize their use. The recycle valve and suction throttling valve add arbitrary loads to the compressor, and thus increase fuel usage. The flare valve leads to a direct waste of reservoir fluids, and thus loss of income. For this reason, engine speed control is recommended for most medium to large size (ò 372.8 kW, or 500 hp) reciprocating compressors for which a constant flow rate cannot be assured by the process.

Some operators do not like to use speed control in gas lift service because the highly variable flows normally encountered tend to make the engine speed cycle on a frequent basis. They believe this causes higher engine maintenance that is not warranted by the savings in fuel efficiencies.

9.5.2.9 Pressure relief valve

Each cylinder discharge line should have a relief valve located upstream of the cooler. As with all reciprocating positive displacement devices, the piston will continue to increase pressure if flow is blocked. The relief valve assures that nothing is overpressured. It must be located upstream of the coolers as hydrates can form in the coolers, blocking the flow of gas. Likewise, the need for suction pressure relief valves should be examined if the possibility of overpressuring suction piping and cylinders exists due to control system failure.

9.5.2.10 Discharge coolers

Coolers are used to decrease the temperature of the gas after it has gone through a stage of compression. It is necessary to cool interstage gas before returning it to a scrubber for another stage of compression. Coolers may also be required to cool the discharge gas prior to gas treating or dehydration, or to meet pipeline specifications. Typically, aerial coolers are used in these situations. Coolers are normally designed for an 11°C (20°F) approach to maximum ambient temperature, although on large units and where there are special process considerations, coolers with a 5.6°C (10°F) approach to ambient may be used.

Cooler exit temperatures are controlled using louvers that control the air flow across the cooling heat exchange surface. These louvers may be manually or pneumatically adjusted to achieve the desired cooling effect.

Several types of fan drives are used. Field units are mostly belt driven from the compressor engine. Larger units can be electric motor driven or hydraulic motor driven from an engine-mounted hydraulic pump.

Normally, the pressure drop through the coolers is limited to 34.5 kPa (5 psi) for coolers with a capacity of up to 1724–2068 kPa (250 to 300 psig) discharge pressure; it is limited to 34.5–68.9 kPa (5 to 10 psi) for higher pressure units.

Shell-and-tube water coolers or shell-and-tube gas-to-gas coolers are often used as alternate methods of gas cooling.

9.5.2.11 Suction scrubbers

Suction scrubbers are required on the unit to catch any liquid carryover from the upstream equipment and any condensation caused by cooling in the lines leading to the compressor. They are also required on all the other stages to remove any condensation after gas cooling. On each suction scrubber a high level shut-down is required so that if any liquids do accumulate in the suction scrubber, the compressor will automatically shut down before liquids carry over to the compressor cylinders.

These gas scrubbers are normally designed for removal of droplets on the order of 400–500 μm and contain mist extractors designed for removing 10-μm droplets.

9.6 Control and instrumentation

9.6.1 Compressor control systems

Compressor control systems use the following to regulate capacity:

- Suction pressure
- Discharge pressure
- Flow
- External process signal

Control systems are normally pneumatic or electronic. The capacity of a compressor is adjusted by:

- Actuating unloaders and clearance pockets
- A control valve in a bypass
- Changing speed

The control system can be manual or automatic. Many systems are programmable. This section will briefly describe the control devices and instruments normally supplied with a reciprocating compressor.

9.6.2 Capacity control

Five-step unloading means capacity control in approximately 25% steps from zero to 100%. Thus if two 50% compressor units are used, capacity control is available in 12.5% steps. Suction valve or plug unloaders can only deactivate each cylinder to zero or 50%. If 25% intermediate step is desired, it is necessary to use clearance pockets. It is possible to obtain five-step unloading in increments other than 25%.

When the pressure ratio is very low, say less than approximately 1.7, the volume of the clearance pocket becomes very large. The actual volume is dependent on the piston displacement and "k" value of the gas. Refer to the formula for volumetric efficiency. API 618 also recommends that volumetric efficiency remain above 40% since performance prediction is generally unreliable below that value. That said, values as low as 25% have been used when less precise capacity control is acceptable.

Note that when the cylinder bore is not much larger than the piston rod, unloading the crank end of the cylinder results in a capacity reduction much $<50\%$.

Some general-purpose machines, such as air compressors, use a three-step control (100%, 50%, and 0%) to control capacity. Three-step control requires more cycle actuation of the unloaders than five-step control. Therefore three-step is more detrimental to machine components, especially valves.

When precise capacity turndown is required, a bypass with a control valve is necessary. Depending on the system requirements, the bypass may only be across the first stage. But, more often the bypass spills back across all stages unless the differential pressure is too high to be handled by a single control valve. The tie-in point for the bypass must be downstream of a cooler so that cooled gas spills back to the suction. If there is no cooler in the discharge, a cooler may be placed in the bypass line. In any case, the bypass should tie-in upstream of a suction knockout drum so that any condensate resulting from the expansion cannot enter the compressor.

When a bypass is used in combination with step control, five-step operation is more efficient than three-step.

9.6.3 Instrumentation

9.6.3.1 Load-less starting

Most motor-driven compressors are equipped with suction valve or plug-type unloaders on both ends of all cylinders on the frame to permit load-less starting. Load-less starting is not mandatory, but facilitates start-up and reduces disturbance in the electrical system. It is also a convenient and less punishing feature for all types of drivers.

9.6.3.2 Alarms and shutdowns

Table 9.4 presents a typical list of alarm and shutdown functions for the compressor frame and cylinders. Local preferences may call for additions to or deletions from this list. The data sheets in API 618 list many other functions. These data sheets may also be used as a good checklist.

If vibration detection devices are used on the frame, an accelerometer-type detector is recommended to provide continuous measurement. Ball-and-seat or magnetic-type switches are unreliable. Vendors should be consulted regarding the best location for the device. Sometimes it is necessary to install two devices, one for transverse vibration and one for axial vibration. Whenever alarms and shutdowns are chosen, it is important to make sure they are installed with provisions to allow testing.

9.6.3.3 Gages

Table 9.5 presents a typical list of gages. Location of the gages is a matter of plant preference. They can be mounted locally on the compressor or piping or mounted in a local panel. Gages mounted on the compressor or attached piping may be subject to vibration. Operations and maintenance personnel should be consulted for best location for these gages.

Table 9.4 Typical list of alarm and shutdown functions

Function	Alarm	Shutdown
Low lube oil pressure	X	X
High discharge temp, (each cyl.)	X	X
High oil filter diff. Pressure	X	
High cyl. Jacket water temp.		X
Low cyl. Jacket water pressure	X	
Low lubricator flow		X
High vibration	X	X
High liquid level, gas separator	X	X

Table 9.5 Limitations for MAWP based on cylinder materials

Material	MAWP, PSIG API 618	MAWP, PSIG oil Field ≤ 8 inch diameter	8 inch diameter
Gay Cast Iron, ASTM A278	1000	1600	1000
Cast Nodular Iron ASTM A395	1000[a]	2500	1500
Cast Steel, ASTM A216	2500	2500	2500
Forged Steel, ASTM A668	>2500	7500	–

[a]May be quoted to 1500 PSIG as separate option.

9.6.3.4 Monitoring

Continuous monitoring is not widely used on reciprocating compressors. Eutectic bearing temperature safety devices are available for main and connecting rod bearings. These devices are spring loaded and they vent control air to alarm or shutdown the unit when high temperature melts the eutectic material. Main bearing thermocouples or RTDs are also available.

The rod drop monitor is gaining more acceptance. This device can be a eutectic sensor that melts when rubbed by the rod allowing a signal to be transmitted for alarm or shutdown. This function can also be accomplished with a proximity probe.

9.7 Lubrication and cooling systems

9.7.1 Frame lubrication system

The frame lubrication system circulates oil to the frame bearings, connecting rod bearings, and crosshead shoes. It can also supply oil to the packing and cylinder lubrication system. Splash lubrication systems are the least expensive and are used in small air compressors. Forced-feed systems are used for almost all oil field gas compression applications.

9.7.1.1 Splash lubrication system

Fig. 9.78 shows a splash lubrication system. An oil ring rides loosely and freely on the rotating shaft, dipping into the oil sump as it rotates. The ring rotates because of its contact with the shaft, but at a slower speed. The oil adheres to the ring and drips onto the shaft.

9.7.1.2 Forced-feed lubrication system

Fig. 9.79 is a schematic of a typical forced-feed lubrication system. In a forced-feed lubrication system, a pump circulates lubricating oil through a cooler and filter to a distribution system that directs the oil to all the bearings and crosshead shoes.

The details of any one system will vary greatly. Major components and considerations of a forced-feed lubrication system are the main oil pump, auxiliary pump, prelube pump, oil cooler, oil filter, and day tank.

The main oil pump is driven from the crankshaft and is normally sized to deliver 110% of the maximum anticipated flow rate. The auxiliary pump serves as a backup for the main oil pump. Often it is electric motor driven and starts automatically when supply pressure falls below a certain level. The prelube pump may be either manual or automatic. It is designed to lubricate the bearings prior to start-up to prevent the bearings from running without proper lubrication on start-up. Some larger units are equipped with postlube pumps to assure no loss of vital lubrication on unit shutdowns as well as proper cool down of bearing surfaces. Either of these pumps may be electric, pneumatic, or hydraulic motor driven.

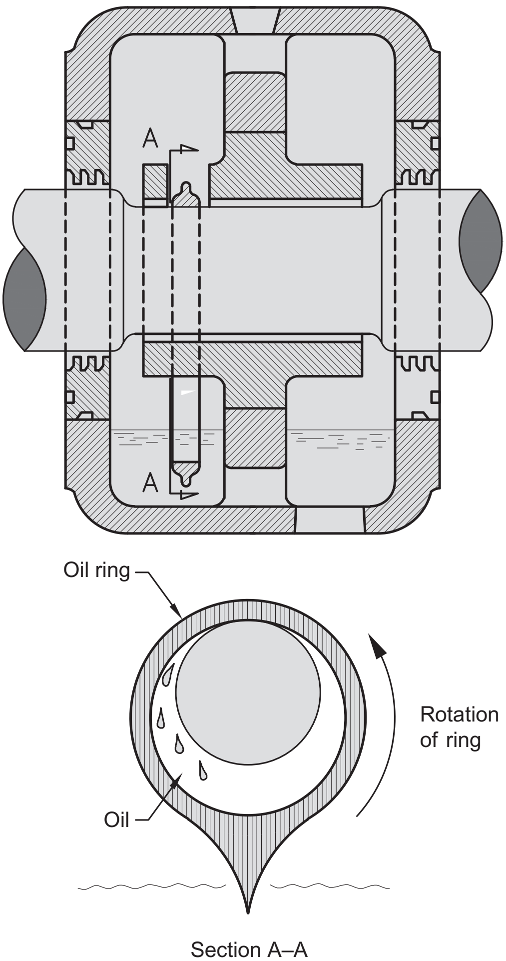


Fig. 9.78 Schematic diagram of a splash lubrication system.

The oil cooler ensures that the oil is kept below a specified temperature (120°F (48.9°C). Jacket cooling water can be used or the oil can be routed directly to a section of the gas aerial cooler. Coolers should be sized for 110% of the maximum anticipated duty.

A dual full-flow oil filter, with isolation valves arranged so switching can occur without causing a low-pressure shutdown, is normally used.

The overhead day tank is normally sized to handle 1 month of oil consumption. It should be equipped with a level indicator.

The lube oil piping is usually stainless steel downstream of filters. Socket weld fittings or other pockets that can accumulate dirt downstream of the filters should be avoided. Carbon steel lines are often used upstream of the filters. These

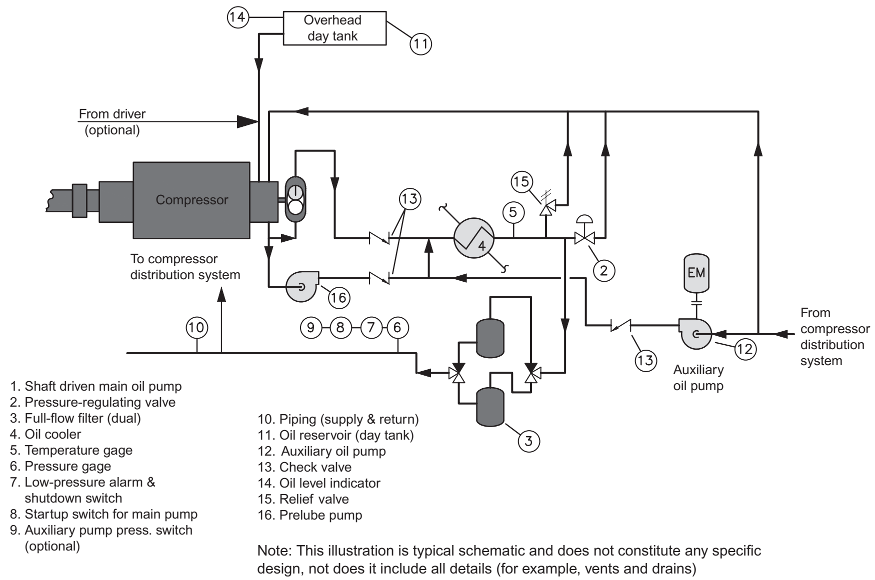


Fig. 9.79 Schematic flow diagram of a forced-feed lubrication system.

should be pickled and coated with rust inhibitor to prevent lube system contamination with corrosion products.

Once fabrication is complete, the lube oil system from pump discharge to the distribution system should be flushed with lube oil at 71°C to 82°C (160°F to 180°F). Oil should flow across a 200-mesh screen. Flushing should cease when no more dirt or grit is found on the screen.

Combustion gases from an engine driver can sour the lube oil if a common system is used for compressor and driver. This practice should be examined closely for specific applications; on larger units, a separate engine lube oil system is often used.

9.7.2 Cylinder/packing lubrication system

Packing/cylinder lubrication can be provided from a forced-feed compressor lube oil system. For very cold installations, immersion heaters and special lube oils must be considered. If the lube oil temperature gets too cold, the oil becomes too viscous and does not flow and lubricate properly.

The flow required to lubricate the packing and cylinders is quite small. The pressure necessary to inject the lubricant at these locations is quite high, so small plunger pumps (force-feed lubricators) such as shown in Fig. 9.80 are used.

The lubricator assembly with reservoir is mounted on the end of the compressor. The piston in each lubricator is driven by the camshaft contained in the reservoir. This camshaft is belt driven off the compressor crankshaft. On the piston downstroke, oil is drawn into the lubricator cylinder from the sight well. The void created in the airtight

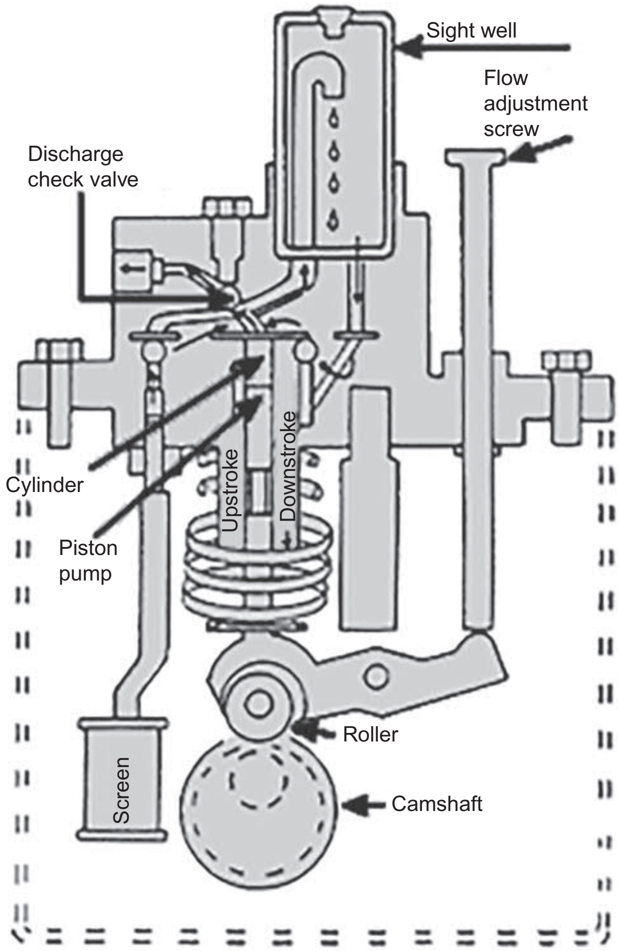


Fig. 9.80 Schematic diagram of a force-feed lubricator.

sight well is filled by oil drawn up from the reservoir. On the piston upstroke, the oil in the cylinder is injected out through the discharge check valve to the distribution system. The number of drops seen falling into the sight well is the amount of oil discharged by the pump. The design should provide extra lubricator slots in the reservoir as well as a low oil reservoir level or no-flow shutdown valves.

Each pump feeds one level of compression (i.e., first stage, second stage, third stage, etc.). Divider blocks are used to distribute the flow equally from the lubricator, between cylinders and packing cases. These blocks consist of a series of cylinders and pistons all mounted on a common shaft, as shown in Fig. 9.81.

As the high-pressure oil entering the block strokes the common piston, different inlet and exhaust ports of the blocks line up. When the ports line up, a preset amount of oil flows through the ports of the divider block. The block and system are typically designed by the compressor manufacturer. Oil is supplied to this system from the

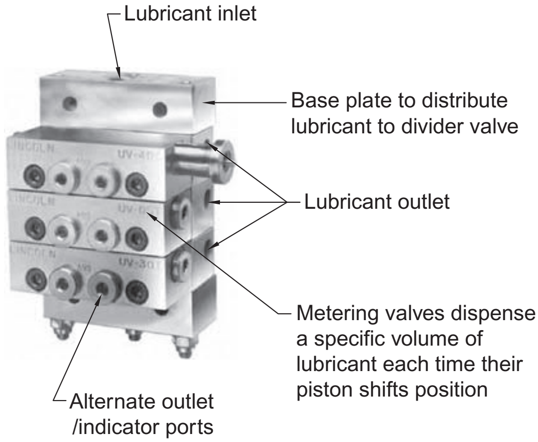


Fig. 9.81 Example of a divider block.
Courtesy of Lincoln Controls.

frame lube oil system or from an overhead tank. This oil comes in contact with (and is contaminated by) the gas being compressed. Gas/lube oil compatibility should be checked.

9.7.3 Compressor cylinder cooling system

It is necessary to provide some cooling to the compressor cylinder to promote longer parts life and reduced maintenance. Cooling also reduces losses in capacity and power due to suction gas preheating and removes heat from the gas, lowering discharge temperature slightly.

In very low power and low-ratio services (especially small vapor recovery compressors), it is possible to cool the cylinders by installing fins and relying on natural air convection. However, most installations require a cooling water system. If the cylinder is too cool, liquids can condense from the suction gas stream. Thus it is desirable to keep the cylinder temperature 10°F (5.6°C) higher than that of the suction gas. If the cylinder is too hot, however, gas throughput capacity is lost due to the gas heating and expanding. Therefore it is desirable to limit the temperature to <30°F (16.7°C) above that of the suction gas.

Fig. 9.82 includes schematics of several types of liquid coolant systems. Forced coolant systems are the most common for reciprocating compressors. A pump circulates the liquid through the cylinder and an aerial cooler. Normally the pump is connected to the drive shaft, and the lubrication cooler is a section of the aerial cooler used to cool the discharge gas. It is very important to use a forced liquid system to dissipate heat generated by friction if the cylinder operates unloaded. On the other hand, thermo-siphon systems use the density differences between the hot and the cold coolant to establish flow. They are typically used when the change in temperature is <150°F (83.3°C), and the discharge temperature is <200°F (93.3°C). Static systems, which employ a liquid and air-cooled system, are not frequently used.

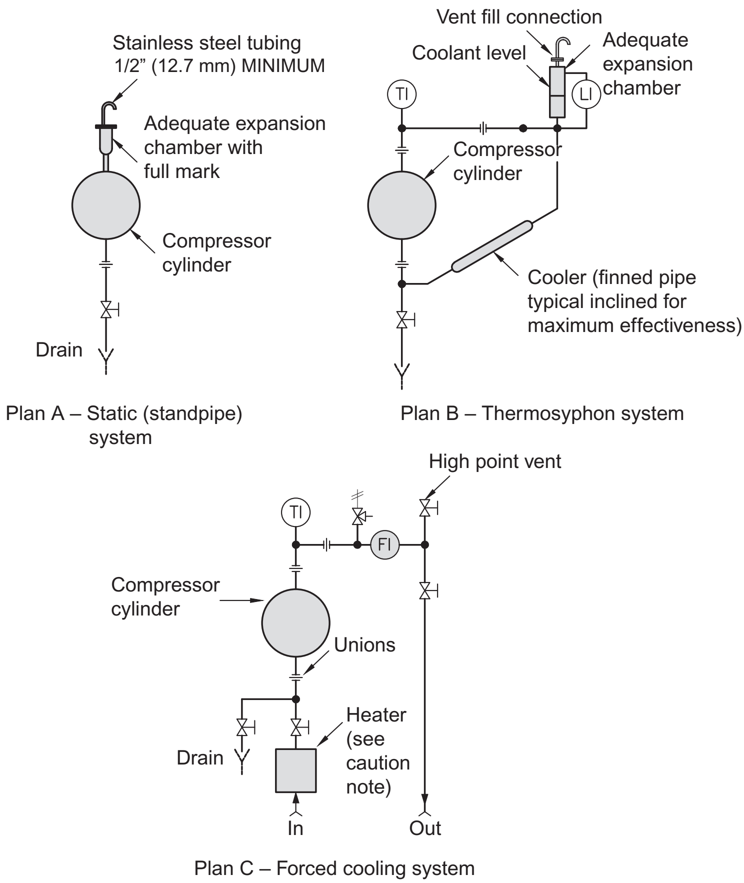


Fig. 9.82 Schematic diagrams of different types of liquid cooling systems.
Courtesy of API.

9.8 Materials of construction

The following is a description of the materials available for the major components of a reciprocating compressor.

9.8.1 Crankshafts

Crankshafts are commonly one-piece forgings or castings. Steel or nodular iron castings are sometimes used for machines up to 1500 HP. The advantages of castings are that counterweights can be an integral part of the shaft. Most companies prefer forged steel for ratings of 200 HP and higher. A typical material designation is ASTM A688 Class F.

9.8.2 Piston rods

The most common material used is heat-treated AISI 4140 steel with a maximum Rockwell "C" hardness of 40 at the core and a minimum of 50 at the surface. If stress corrosion is a design factor, this material is annealed to a hardness of 22C maximum (core) and 50C minimum (surface). AISI 8620 with the same hardness provides higher working stresses for stress corrosion applications.

Rods of 17-4 pH stainless steel are used for corrosive services. Core and surface hardness are 40-50C for standard applications. When stress corrosion is present, the through-hardness is limited to 33C.

9.8.3 Crossheads

Crossheads are available in cast gray iron, nodular iron, or steel. Most companies prefer cast steel for all high-horsepower applications. It is recommended that cast gray iron be allowed only on smaller machines with ratings less than about 200 HP.

9.8.4 Connecting rods

Connecting rods should be forged steel. A typical material designation is ASTM A235.

9.8.5 Compressor cylinders

The maximum allowable working pressure (MAWP) corresponding to materials is typically limited to the valves given in Table 9.5. Note that nodular iron may be used in machines above 1000 psig only in special cases. Nodular iron is an excellent engineering material, but homogeneity of the material throughout the casting can sometimes be a problem. Thus the yield strength may not be as high as anticipated. API 618 calls for specimen testing and other NDE in an effort to ensure the quality of nodular iron castings.

9.8.6 Compressor valves

Valve materials are selected for both durability (long-term operation) and compatibility with the gas being handled. Table 9.6 presents materials for valve guards and seats. Valve plates are available in various types of stainless steels and thermoplastics, as given in Table 9.7. Table 9.8 presents the wide variety of spring materials available, from music wire to Inconel. Other super alloys, such as Elgiloy and Haynes 25, are being used to avoid hydrogen embrittlement for springs.

Surface finish and parallel face surfaces are the most serious considerations for metallic plates. Dimensional stability or thermoplastic plates in humid and high-temperature environments is essential. One disadvantage to the use of thermoplastic plates is their affinity for moisture, called hygroscopicity. Newer materials, such as PEEK and TORLON, have lower absorption rates (0.06%) than nylon glass

Table 9.6 Typical guard and seat materials for compressor valves

Material	Application
1141	Light duty noncorrosive service
Heat Treated 1141	Light to medium noncorrosive service
Ductile Iron	Light to medium service—resistance to some chemical attack
4140	Medium to high strength—resistance to some chemical attack
Heat Treated 4140	High strength service—resistance to some chemical attack
400 Series Stainless Steel	Corrosive service
300 Series Stainless Steel 17-4 PH Stainless Steel	Extreme corrosive service

Table 9.7 Typical valve plate materials

Material	Application
Glass-filled nylon thermoplastic	Good impact and corrosion resistance 270°F temperature limit
Peek—Polyetheretherketone	High strength—high temperature (up to 375°F)
Linen-Based Phenolic Laminate	Clean gas. low compression ratios. 225°F temperature limit
Laminated cloth-based phenolic	High-temperature applications up to 400°F—available for ported plate application only
410 Stainless steel	moderate corrosion resistance, good impact resistance
17-7 PH stainless steel	Moderate corrosion resistance, good impact resistance
Inconel X-750	High corrosion resistance and high strength properties in high-temperature applications

Table 9.8 Typical valve spring materials

Material	Application
Music Wire	Low corrosion resistance. good durability in clean gas environments and low temperatures
302 Stainless Steel	Moderate corrosion resistance. Average durability in moderate temperatures
17-7 FH Stainless Steel	Excellent corrosion resistance, high strength properties in moderate/high temperatures (700°F Max)
Inconel X-750	High corrosion resistance, high strength properties in high-temperature applications (1100°F Max)

composites (1%). Also, some of the materials have a lower coefficient of thermal expansion. A low thermal expansion factor makes the plate more resistant to deformation at higher temperatures and better able to hold dimensional integrity.

9.9 Foundations

9.9.1 General considerations

For complex offshore structures or places where foundations may be critical, finite element analysis computer programs can be used to calculate the foundation's natural frequency and the forced vibration response. In most onshore locations the foundation design is not critical. It is normally less expensive to provide a greater than necessary mass of concrete than it is to design a more economical foundation (Fig. 9.83).

A well-balanced compressor unit is preferred. The concrete foundation weight should be 2 to 3 times the compressor and driver weight. The static soil bearing should be <50% of the allowable bearing strength. Generally, it is better to increase the length and/or width rather than depth to meet the weight requirements. For integrals, a pedestal on a rigid mat spreads out the load and provides a longer moment arm to resist unbalanced forces. The mat should be at least 30 in. (75 cm). For rectangular block, at least 40% of height (but not <18 in. (45 cm)) should be embedded in undisturbed soil.

9.9.2 Concrete

Concrete should be poured into a "neat" excavation without formed side faces. The concrete should have crushed rock aggregate (not river gravel) and a 28-day strength of 35,000 psi (241,316 kPa).

9.9.3 Steel

The side and end faces should have a minimum 12-in. (30 cm) by 12-in. (30 cm) grid of shrinkage steel. The steel area should be 0.002 times the cross-section. The maximum spacing of reinforcing bars should be 18 in. (45 cm). For mat/pedestal foundations, vertical dowels (#6 minimum at 12 in. (30 cm) around pedestal periphery) with one-foot horizontal right-angle hook. The top of dowel should extend without splice to within 2 in. (10 cm) of top of pedestal. The natural frequency of support beams must be at least 20% different than compressor running speed at any harmonic. Hex-head type anchor bolts should be used. Safety jacking crews are used to facilitate leveling.

9.9.4 Grouting

It is suggested to use an expert grout supply firm. Soleplates should be grouted to the foundation, and the compressor and driver bolted to the soleplates. Allow for a minimum of 1/8-in. (3.175 mm) shim between soleplate and compressor to bring unit into final alignment. After 7 days of curing, chip ½ inch off the top of the surface.

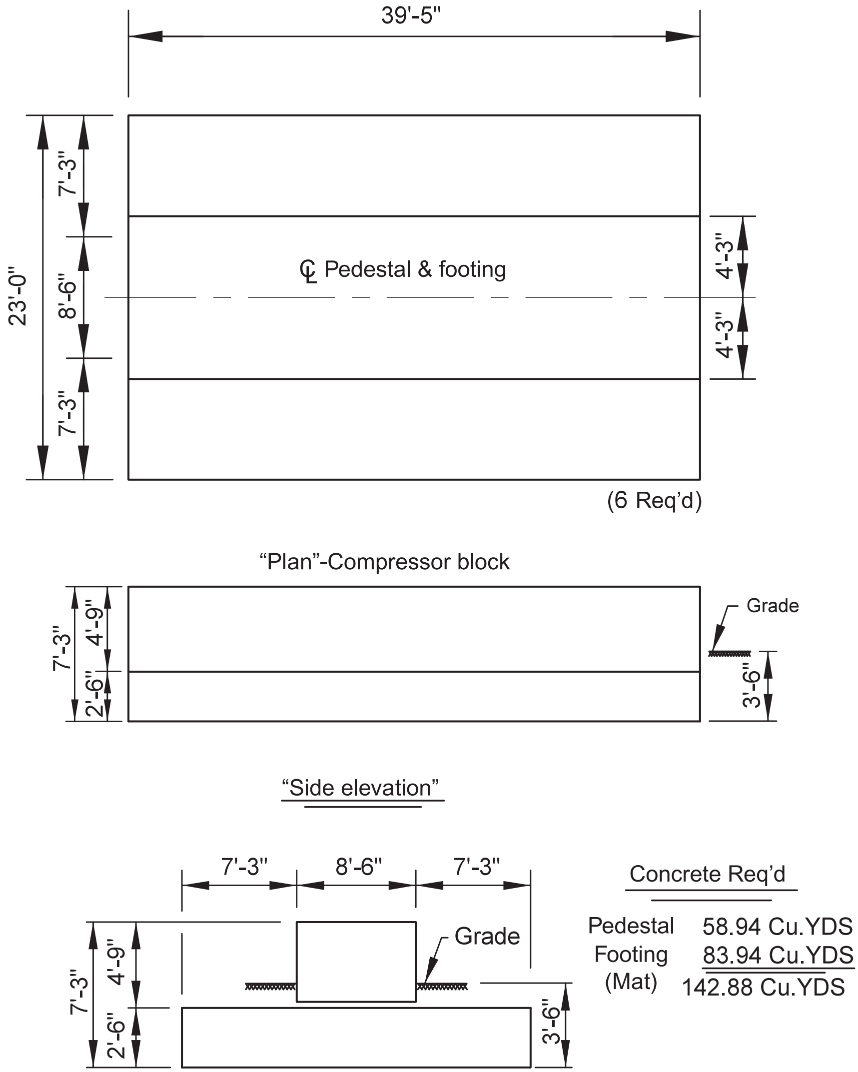


Fig. 9.83 Typical foundations design details.

Generally, epoxy grout should be used. Nonshrink cement should be used only for short (6 months) duration projects. Figs. 9.84–9.86 are examples of typical foundation design details.

9.9.5 Unbalanced forces

Due to the basic design of the machine, its rotating and reciprocating mass produce inertia forces that cannot be eliminated completely by balancing. The manufacturer has the ability to minimize the forces' magnitude by extending the cranks, adding

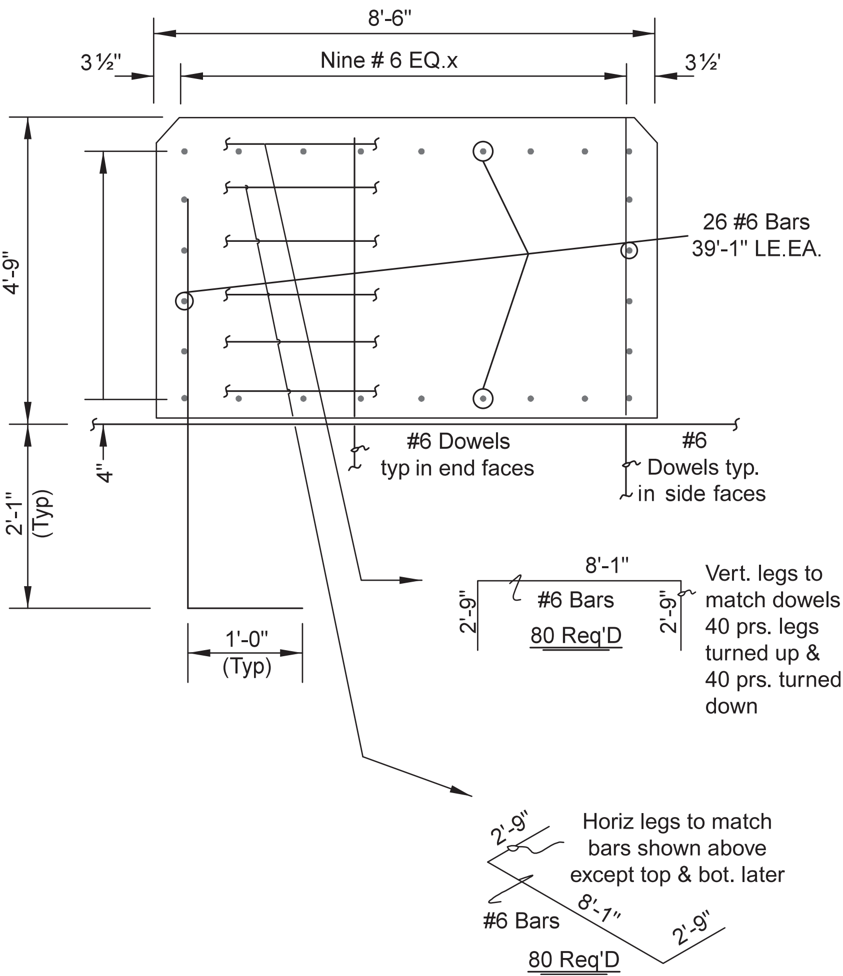


Fig. 9.84 Typical foundations design details.

counterbalance weights to the crosshead, and phasing the crank-throws. The magnitude of these forces must be provided by the manufacturer.

It is beyond the scope of this tutorial to describe the procedures used to design a foundation for unbalanced forces. The following comments are meant to familiarize the project engineer with the concepts involved.

The forces produced by unbalance act at each crank-throw, and while the summation of forces may be small, the magnitude of each may be quite large. Therefore forces from each crank-throw must be provided by the manufacturer for a detailed design of unbalanced forces. The moments generated by the crank-throws act about the center of gravity.

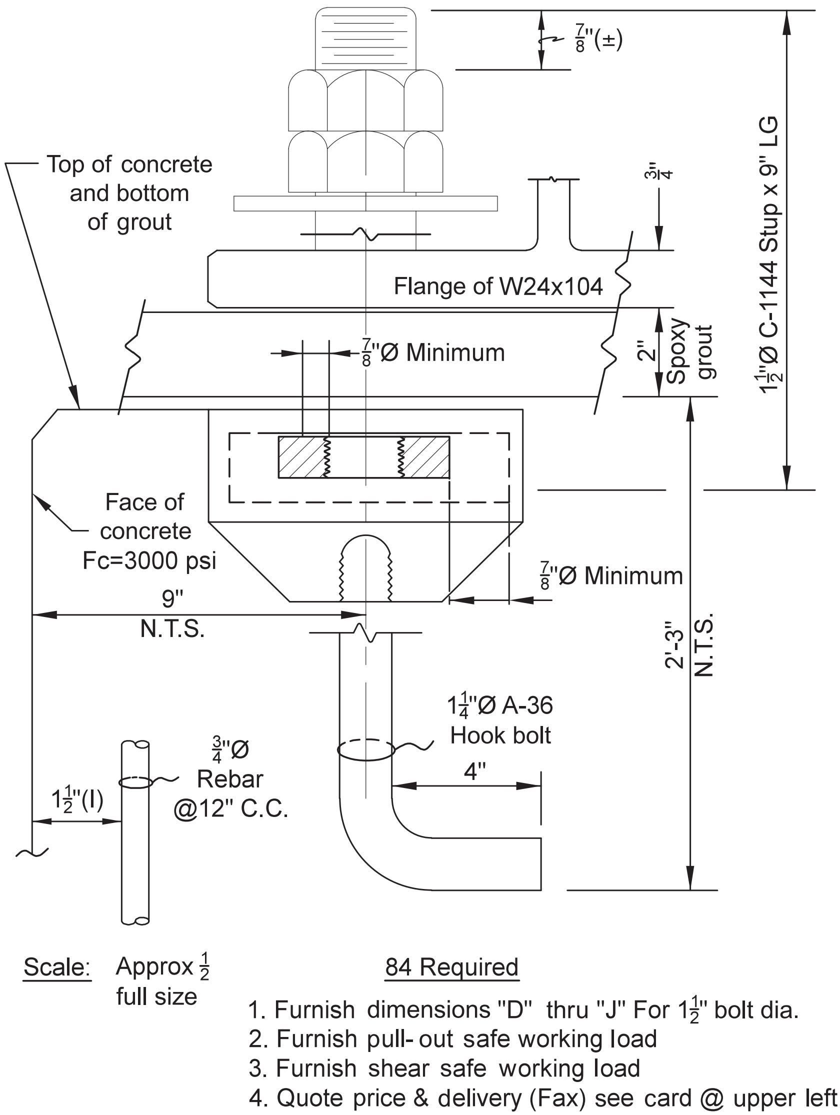


Fig. 9.85 Typical foundations design details.

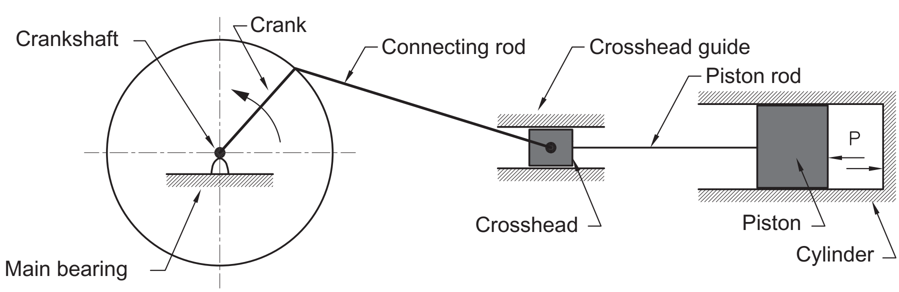


Fig. 9.86 Schematic diagram showing mechanical linkage of a reciprocating compressor.

Fig. 9.87 shows the mechanical linkage of a reciprocating compressor whose purpose is to drive the compressor piston back and forth in a strictly linear manner. The linkage can be considered as consisting of three separate point masses connected by shafts whose mass is negligible. This is represented by Fig. 9.88:

where:

m_1 = rotating mass of the crank acting at d, kg (lb)
m_2 = rotating mass of the connecting rod acting at radius r, kg (lb)
m_3 = mass of piston, piston rod, and cross head, kg (lb)
ω = rotative speed of compressor. Rad/s
r = crank radius, mm (in)

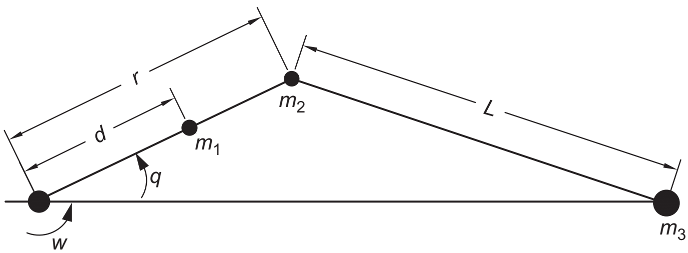


Fig. 9.87 Schematic diagram of a reciprocating compressor crank-throw.

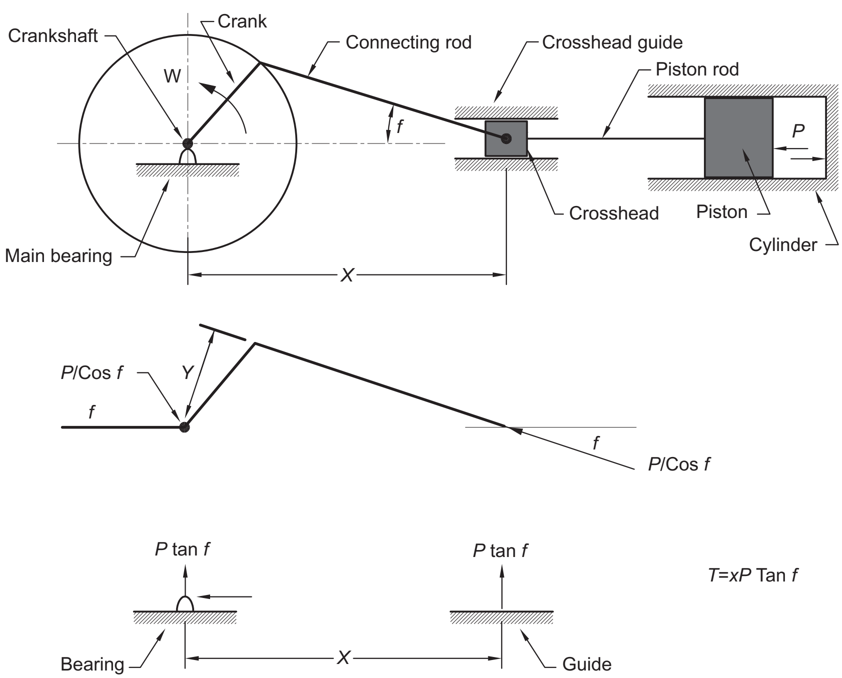


Fig. 9.88 Schematic diagram of a gas torque in a reciprocating compressor.

L = connecting rod length, mm (in)
d = distance from main bearing to mass center of crank, mm (in)

Since the compressor piston velocity is changing in magnitude and direction, the horizontal acceleration of m_3 can be approximated by Eq. (9.42). This is an approximation since it is derived from an infinite series and then differentiated. As is characteristic of most connecting rod/crank pin systems, there is no vertical acceleration for m_3.

$$a_{m3x} = r\omega^2 \cos\theta + \omega^2 \frac{r^2}{L} \cos 2\theta \tag{9.42}$$

The acceleration components of m_1 and m_2 in the horizontal and vertical directions have been derived from the theory of angular acceleration due to rotating masses. The acceleration term for each mass is defined as follows:

a_{mlx} = vertical acceleration of $m_1 = \omega^2 d \sin\theta$
a_{mly} = horizontal acceleration of $m_1 = \omega^2 d \cos\theta$
a_{m2x} = horizontal acceleration of $m_2 = \omega^2 r \sin\theta$
a_{m2y} = horizontal acceleration of $m_2 = \omega^2 r \cos\theta$

Using Newton's second law, F = ma, the forces for each compressor cylinder are thus obtained and listed as follows. It should be noted that primary forces are those whose frequency is equal to running speed and secondary forces are those whose frequency is twice the rotational speed of the compressor.

$$F_x = \omega^2 (m_1 d + m_2 r + m_3 r) \cos\theta + m_3 \omega^2 \frac{r^2}{L} \cos 2\theta \tag{9.43}$$

$$\sum F_y = \omega^2 (m_1 d_1 + m_2 r) \sin\theta \tag{9.44}$$

$$\text{Primary horizontal force} = \omega^2 (m_1 d + m_2 r + m_3 r) \tag{9.45}$$

$$\text{Secondary horizontal force} = \omega^2 \cos\theta + \omega^2 \frac{r^2}{L} m_3 \tag{9.46}$$

$$\text{Primary vertical force} = \omega^2 (m_1 d + m_2 r) \tag{9.47}$$

$$\text{Secondary vertical force} = 0 \tag{9.48}$$

It is possible to add weights to m_1 to balance the horizontal primary loads (see Eq. 9.43). However, this will cause a large imbalance in the vertical direction (Eq. 9.45) and will not balance the secondary horizontal loads. Thus the compressor shaft, frame, and foundation have to be designed for load imbalances.

Balanced-opposed compressor frames have cylinders on both sides of the frame; they are counterweighted so that one cylinder balances the primary load of the other, minimizing both horizontal and vertical load imbalances. However, in plain view, the cranks for each cylinder are offset, creating an imbalance couple on the crankshaft, which must be handled in the shaft, frame, and foundation design.

The inertial force of accelerating the piston and rod described before add to the gas rod loads discussed before. Fortunately, the maximum inertia load does not occur at the same time as the maximum gas load, and gas loads alone can be normally used to check against maximum allowable rod load. However, when cylinders are single acted, a check should be made with the manufacturer to assure that the combination of both gas and inertia loads is acceptable.

9.9.6 Gas torque

This is the torque created by the gas pressure forces as the compressor goes through a revolution. This torque has a primary component ($1 \times$ rpm) and a secondary ($2 \times$ rpm) component. See Fig. 9.89 for schematic of gas torque in reciprocating machinery.

The gas pressure in the cylinder exerts a horizontal force on the ends of the cylinder which is balanced by the gas rod loads, and thus there is no net horizontal load on the foundation. However, because of the angle of the connecting rod, a vertical force is imparted on the crosshead guide. This is balanced by a vertical force on the main bearing forming a couple that will tend to rock the compressor.

Usually, the manufacturers will check the performance of the compressor due to the gas torque created. Analysis tests are run to determine torque limits such as critical torques, maximum torques, and foundation stiffness, which must resist this moment.

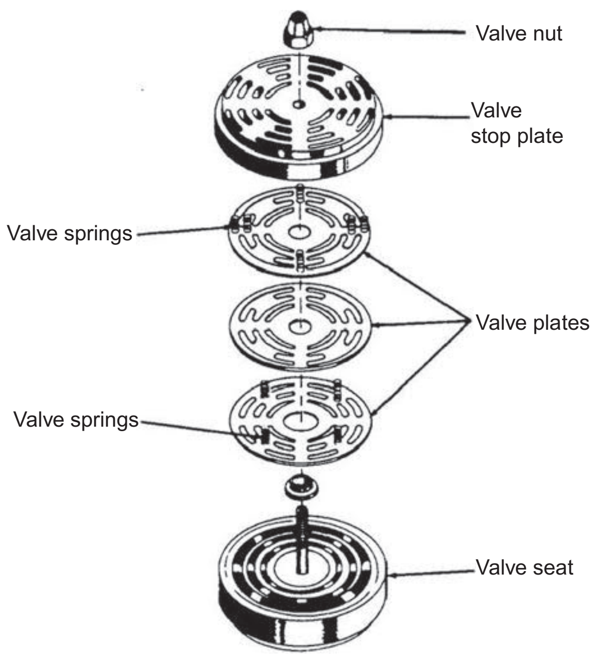


Fig. 9.89 Parts of a channel valve for Exercise # 9.

9.10 Troubleshooting

Table 9.9, from GPSA, gives some hints for troubleshooting reciprocating compressors and may prove useful during start-up and operation.

9.11 Specifications

There are two API specifications for reciprocating compressors. API STD 618 (used for general refinery service) and API SPEC 11P (used for field gas compressors). API STD 618 results in a higher quality and more expensive compressor than does 11P, and in production facilities it is normally used only for high power units or offshore units.

Table 9.9 Probable causes of reciprocating compressor trouble

Trouble	Probable cause(s)
Compressor will not start	1. Power supply failure 2. Switchgear or starting panel 3. Low oil pressure shutdown switch 4. Control panel
Motor will not synchronize	1. Low voltage 2. Excessive starting torque 3. Incorrect power factor 4. Excitation voltage failure
Low oil pressure	1. Oil pump failure 2. Oil foaming from counterweights striking oil surface 3. Cold oil 4. Dirty oil filter 5. Interior frame oil leaks 6. Excessive leakage at bearing shims and/or tabs 7. Improper low oil pressure switch setting 8. Low gear oil pump bypass/relief valve setting 9. Defective pressure gage 10. Plugged oil sump strainer 11. Defective oil relief valve
Noise in cylinder	1. Loose piston 2. Piston hitting outer head or frame end of cylinder 3. Loose crosshead lock nut 4. Broken or leaking valve(s) 5. Worn or broken piston rings or expanders 6. Valve improperly seated/damaged seat gasket 7. Free air unloader plunger chattering
Excessive packing leakage	1. Worn packing rings 2. Improper lube oil and/or insufficient lube rate (blue rings)

Continued

Table 9.9 Continued

Trouble	**Probable cause(s)**
	3. Dirt in packing
	4. Excessive rate of pressure increase
	5. Packing rings assembled incorrectly
	6. Improper ring side or end gap clearance
	7. Plugged packing vent system
	8. Scored piston rod
	9. Excessive piston rod run-out
Packing overheating	1. Lubrication failure
	2. Improper lube oil and/or insufficient lube rate
	3. Insufficient cooling
Excessive carbon on valves	1. Excessive lube oil
	2. Improper lube oil (too light, high carbon residue)
	3. Oil carryover from inlet system or previous stage
	4. Broken or leaking valves causing high temperature
	5. Excessive temperature due to high pressure ratio across cylinders
Relief valve popping	1. Faulty relief valve
	2. Leaking suction valves or rings on next higher stage
	3. Obstruction (foreign material, rags), blind, or valve closed in discharge line
High discharge temperature	1. Excessive ratio on cylinder due to leaking inlet valves or rings on next higher stage
	2. Fouled intercooler/piping
	3. Leaking discharge valves or piston rings
	4. High inlet temperature
	5. Fouled water jackets on cylinder
	6. Improper lube oil and lube rate
Frame knocks	1. Loose crosshead pin, pin caps, or crosshead shoes
	2. Loose/worn main, crankpin, or crosshead bearings
	3. Low oil pressure
	4. Cold oil
	5. Incorrect oil
	6. Knock is actually from cylinder end
Crankshaft oil seal leaks	1. Faulty seal installation
	2. Clogged drain hole
Piston rod oil scraper leaks	1. Worn scraper rings
	2. Scrapers incorrectly assembled
	3. Worn/scard rod
	4. Improper fit of rings to rod/side clearance

Most standard field gas compressor packages have compressors constructed in accordance with API SPEC 11P, which results in a much less costly compressor, but one that probably requires higher operating and maintenance costs.

The basic requirements of API STD 618 and API SPEC 11P are summarized as follows for comparison purposes and to provide some guidance in making selections of various options:

9.11.1 API STD 618-reciprocating compressors for general refinery services

1. *Discharge temperature*—limited to 300°F (148.9°C).
2. *Vibration and critical speeds.*
   - No torsional natural frequencies within 10% of any shaft speed and 5% of twice any shaft speed.
   - Vendor states all critical speeds.
3. *Compressor cylinders*
   - Must be rated the greater of 10% or 15 psi (103.4 kPa) above discharge pressure.
   - Horizontal cylinders with bottom discharge for saturated gases.
   - Replaceable liners preferred but not required.
   - Cylinder heads, stuffing boxes, clearance pockets, and valve covers mounted with studs.
4. *Cylinder cooling*
   - Static cooling if $T_d < 190°F$ (87.8°C) and $\Delta T < 150°F$ (83.3°C).
   - Thermosiphon if 190°F (87.8°C) $< T_d <$ 210°F (98.9°C) and $\Delta T <$ 83.3°C (150°F).
   - Forced liquid if $T_d >$ (210°F (98.9°C), $\Delta T >$ 150°F (83.3°C), or cylinder runs unloaded.
   - Coolant system designed for $P >$ 75 psig (517.1 kPa), $\Delta P <$ 10 psi (68.9 kPa), $\Delta T <$ 10°F (5.6°C) and fouling factor = 0.002.
5. *Pistons and piston rods*
   - Pistons attached to rod by shoulder and lock nut.
   - Rod positively locked to crosshead.
   - Bearing load of fluorocarbon wear rings <5 psi (34.5 kPa).
   - Rods shall be SAE 4140 with 50 Rockwell C hardness.
   - Rod finishing specified.
   - Rod load reversal required.
6. *Crankshafts*
   - Forged or cast in one piece.
   - For power > BHP > 150 (112 kW) use forged steel.
   - Dynamically balanced if speed >800 rpm.
7. *Crossheads*
   - For power > BHP > 150 (112 kW) use forged or cast steel.
   - Replaceable slides.
8. *Bearings*
   - For power > BHP > 150 (112 kW) use replaceable main and crank.
   - Replaceable bushings for crosshead pins.
9. *Distance pieces*
   - Drain and vent required.
   - Packing vent required.
10. *Packing*
    - Packing glands bolted on and made of steel.
    - Cooling criteria for packing gland set.

11. *Lubrication*
- For power > 112 kW (BHP > 150) use pressure system for frame.
- Auxiliary pump required
- Cooler details specified.
- Filter:
  - Full flow filter.
  - 25 μm for babbit bearings.
  - 10 μm for aluminum or micro-babbit bearings.
  - Clean $\Delta P < 5$ psi (34.5 kPa).
  - Maximum design $\Delta P > 50$ psi (344.7 kPa).
- Forced-feed mechanical lubricators for cylinder and packing with sight flow lubrication, check valve, and 30-h reservoir capacity.

12. *Guards*
- Required on all moving parts.
- Outdoor guards weatherproofed and ventilated.
- Piping and pressure codes

13. *Piping designed to ASME B31.3.*
- Charpy test for material and welds for $T < -20$°F (−28.9°C).
- Stainless lube oil piping downstream of filter; carbon piping pickled upstream.
- Threaded connections seal welded with two passes.
- Cast iron and malleable iron pipe, fittings, valves, strainers not allowed in pressurized flammable or toxic service.
- Piping 1« in and less shall be Schedule 80.
- Miscellaneous piping details specified.
- Pulsation bottles designed to ASME B31.3.
- Shell-and-tube exchangers in accordance with TEMA Class B, C, or R. For flammable or toxic gases ASME, Section VIII, Div. 1.
- Air coolers per API STD 661.

14. *Instrumentation*
- Gauges
- Bearing oil header pressure.
- Gas temperature at inlet and discharge of each stage.
- Gas pressure at inlet and discharge of each stage.
- Oil cooler inlet and outlet oil temperatures.
- Differential pressure for lube oil filter.
- Water temperatures in and out of coolers.
- Alarms or shutdowns
- Low oil pressure—shutdown.
- High discharge temperature if handling air or oxygen—alarm.
- Relief valves—as required per API STD 510.

15. *Materials*
- Cylinder—gray iron (ASTM A278) or nodular iron (ASTM A395) for MAWP up to 1000 psig (6895 kPa).
- Cast steel (ASTM A216) for MAWP up to 2500 psig (17,237 kPa).
- Forged steel for MAWP >2500 psig (17,237 kPa).
- Pressure vessels—per ASME SEC VIII.
- Welding—per ASME SEC IX.
- Pulsation bottles—1/8 in corrosion allowance.
- Miscellaneous casting requirements.

16. *Tests*
    - Hydrostatic with water to 1 1/2 MAWP for cylinders, cooling jackets, piping pressure vessels, filters, coolers, and so on.
    - Helium test at MAWP if MW of gas <12 or >0.1 mol percent H_2S in addition to hydrostatic test.
    - Test maintained for 30 min.
17. *Pulsation design*
    - Purchaser must specify design approach.
    - Criteria established for each approach.
18. *Guarantees—No negative tolerance on capacity*; that is, as a minimum, the compressor must meet specified condition of throughput and pressure.

9.11.2 API SPEC 11P-packaged high-speed separable engine-driven reciprocating gas compressors

1. *Discharge temperature*—limited to 350°F (176.7°C).
2. *Vibration and critical speeds*—no requirements.
3. *Compressor cylinders*
    - Must be rated the greater of 10% or 103.4 kPa (15 psi) above discharge pressure.
    - Replaceable liners preferred, but not required.
4. *Cylinder cooling*—as per manufacturer.
5. *Pistons and piston rods*—rods shall be SAE 4140.
6. *Crankshafts*—no requirements.
7. *Crossheads*—no requirements.
8. *Bearings*—no requirements.
9. *Distance pieces*
    - Drain and vent required.
    - Two compartment distance pieces required if crankcase and lube oil system materials not resistant to attack from compressed gas.
    - No packing vent required.
10. *Packing—no requirements*.
11. *Lubrication*
    - Pressure system not required for frame.
    - Cooler—provide if necessary.
    - Filter—full flow if pressure lubricated.
    - Forced-feed mechanical lubricators for cylinder and packing with reservoir.
12. *Guards*
    - Required on all moving parts.
    - No requirements for outdoor guards.
13. *Piping and pressure codes*
    - Piping per ASME B31.3.
    - Lube oil piping cleaned and oiled.
    - Screwed hydrocarbon piping 1«-in and less shall be Scheme 60.
    - Misc. piping details specified.
    - Pulsation bottles per ASME SEC VIII.
    - Air coolers, shell, and tube exchanger design details given (TEMA, ASME, API STD 661 not required).

14. *Instrumentation*
    - Gauges
    - Oil pressure.
    - Gas temperature at discharge only of each cylinder.
    - No gas pressure required.
    - No oil cooler temperatures required.
    - Differential pressure for lube oil filter.
    - No water temperatures required.
    - Shutdowns
    - Low oil pressure
    - Low suction pressure.
    - High suction.
    - High discharge pressure.
    - Cooler vibration.
    - High scrubber level.
    - High discharge temperature each stage.
    - Relief valves—no requirements.
15. *Materials*
    - Cylinder—manufacturer's standard.
    - Pressure vessels—per ASME SEC VIII.
    - No corrosion allowance required on bottles.
    - No miscellaneous casting requirements.
16. *Tests*
    - Tests per code requirements.
    - No helium test.
17. *Pulsation design*
    - Bottles with 5 times piston displacement, if nothing else specified.
    - No criteria established.
18. *Guarantees*
    - Capacity ±3% with $P_s > 5$ psig
    - Capacity ±6% with $P_s < 5$ psig
    - Miscellaneous items
    - Specifies scrubber design details.
    - Specifies paint.

9.12 Rerating considerations

9.12.1 Changes in capacity

To change the capacity of a reciprocating compressor the following methods may be used:

- Changing speed.
- Increase or decrease clearance of cylinders.
- Rebore unlined cylinders and change pistons.
- Reline cylinders and change pistons.
- Replace cylinders.

9.12.1.1 Changing speed

Changing compressor speed is rarely a viable option. Increasing speed will not be an option if the frame is already running at the maximum rated speed. Changing the speed of a motor-driven unit requires replacement of the driver. If the driver is a variable-speed machine, there is a change that something can be done, but the system must be carefully studied. The torque and power ratings of the driver, compressor, couplings, and gear (if any) should be checked, and a torsional analysis of the system conducted. A review of the pulsation dampers and piping design may also be required.

9.12.1.2 Increase or decrease clearance of cylinders

For small changes in capacity, it may be possible to alter the clearance of the cylinders. It will usually be very difficult to significantly increase capacity in this manner unless the cylinders were originally overclearance. That said, it might be possible to use this method in combination with other methods. Normally, capacity can be reduced quite easily by adding clearance, with spacers or clearance pockets, or by reducing piston length.

9.12.1.3 Rebore unlined cylinders, reline cylinders, and change pistons

Unless the cylinders are at maximum diameter, unlined cylinders can be rebored, sometimes by a substantial amount, to increase displacement. Similarly, liners can be replaced in lined cylinders to increase or decrease displacement. There is no guideline for the amount of displacement change that can be achieved. Consult the manufacturer. If changes are contemplated, a computer study of the combined rod loads should be made. Although a large change would not be expected, the manufacturer should review the torque-effort diagram.

In rare cases, where it is desired to increase capacity and power, the crankshaft may be a limiting factor. Crankshafts have maximum horsepower-per-throw and total horsepower (torque) ratings.

There is no way to upgrade the maximum allowable working pressure (MAWP) of a cylinder, unless it was underrated for the original application. It is worthwhile asking the manufacturer whether the cylinder's present nameplate is the actual maximum rating.

9.12.2 Valve upgrades

Before considering valve upgrades, the overall compressor system must be evaluated. For example, to cure a liquid problem, the suction vessel must be looked at.

The designer should check the vessel sizing and the damper boot sizing. The insulation and heating should also be checked to determine if condensation is a possibility. Always keep the cylinder jacket water at least 10 to 20°F above the inlet gas temperature. Most liquid slugging occurs at start-up when the compressor is cold. Make sure there are no dead legs where liquids can accumulate. All of these problems should be corrected before making a decision to upgrade the valves.

Valves do not pass liquids well. Liquid slugs have been known to cause broken pistons and can actually separate the compressor cylinder from the crankcase. The usual evidence of liquid slugging is severely damaged plates or no plates at all—only pieces. Springs can also collapse, although this is sometimes difficult to detect, since by the time the cylinder is opened the water or hydrogen liquid may have drained or evaporated.

Lube oil accumulation from overlubrication can have similar effects. This is especially prevalent with some compressors equipped with plug-type unloaders on the suction or top side of the cylinder.

There are many new valve designs available from OEM and non-OEM sources. Valve problems are often design related, and they can be solved by a change in materials or valve type. Before recommending replacement of existing compressor valves, the designer should ask the vendor to conduct a valve motion study on the existing application. One should be sure the valve design is such that neither the valve guard not the assembly bolting can fall into the cylinder, even if the valves assembly bolting breaks or comes loose.

Nonmetallic valve plate materials have been developed for relatively high temperatures. These materials are only about one-sixth the weight of steel. Also, nonmetallic materials can be contoured to reduce the drag coefficient for the flow around the plate. These designs have shown a great deal of improvement in valve life. That said, existing valves should not be indiscriminately replaced with these new designs.

It is important to determine whether the manufacturer designs the unloader system or only the valves. One should be careful of a manufacturer who supplies only valves but relies on others for the unloader mechanism. The unloader system is an integral part of most compressors and must be designed in conjunction with the valves.

The designer should also determine the manufacturer's capability in materials engineering, finite element analysis, and nondestructive examination (NDE). It is important to find out whether the manufacturer can perform mechanical testing including tensile, hardness, and impact test.

9.12.3 Modifications to suction system

When considering modifying reciprocating compressor suction systems one should be cautious because changes may alter the acoustic response. Unacceptable levels of vibration, high piping and nozzle stresses, and compressor valve problems may result from the addition of knockout vessels and coalescers, or from piping changes. Chances are often good that problems will not result, but there is always a risk that a project may become a statistic. In new equipment installations, an acoustic study is the tool used to mitigate the potential risk.

Installing a coalesce or other piece of equipment in a reciprocating compressor suction line changes the acoustic length of the line or creates two new acoustic lengths where there was previously one. An acoustic study is a design review of these lengths to determine if any acoustic resonances will occur and if they will coincide with the mechanical natural frequencies in the piping system. In some situations, it is prudent to revisit this work when making field changes.

API 618 provides guidance as to when an acoustic study is recommended for new machinery installations. An acoustic study should be considered when any of the following is true:

- Two or more compression stages
- Three or more cylinders per stage
- Final discharge pressure exceeds 1000 psig (6895 kPa)
- Driven equipment horsepower is 500 bhp (373 kW)
- Service alternatives between gases of significantly different molecular weights
- Interaction is anticipated between compressors 15 bhp and greater.

Acoustic studies are usually not performed for machines of $<$150 bhp.

The API 618 recommendations provide a good basis for deciding whether to perform an acoustic study when altering a system, but risk assessment should also play a part in decision making as these studies can be quite expensive. The risk of having harmful pulsations increases as compressor running speed decreases and as the gas acoustic velocity increases (usually as molecular weight decreases). As these parameters change, acoustic lengths get very long and fall out of the normal range of field piping lengths. Also, the intended use of the compressor plays a part in risk assessment. The cost of production losses should be weighed against the cost of performing an acoustic study as part of an alteration.

Everything being equal, a packaged high-speed reciprocating compressor used in a producing field gas application would be considered low risk because it is high-speed, high molecular weight, with low to moderate production losses. On the other hand, a large hydrogen booster compressor in a refinery would be considered high risk for acoustic problems because it is low speed, low molecular weight, with high production losses.

Example 9.1 Compressor Design (Oilfield Units)
Given:

Late in the field's life 100 MMSCFD must be compressed.

Upstream of the separator, gas pressure is 800 psig at 100 °F. The desired discharge pressure is 1000 psig. An engine-driven separable compressor is available from surplus. The engine is rated for 1600 hp. at 900 rpm. Power is proportional to speed. The compressor frame has six 7 in bore by 6.0 in stroke double-acting cylinders with a minimum clearance of 17.92%, a rod load limit of 25,000 lb., and rod diameter of 1.75 in. Assume one stage only.

Assume

$k = 1.26$,
$S = 0.88$,
$Z_s = 0.88$ and
$Z_d = 0.85$.

Determine:

- Discharge temperature
- Volumetric efficiency

- Required clearance
- Rod load
- Required power for given conditions

The lowest suction pressure at which this unit can compress 100 MMSCFD.

Solution:

Step 1. Estimate gas discharge temperature

$$T_d = T_s \left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}$$

where:

T_d = stage discharge temperature, K (°R)
T_s = stage suction temperature, K (°R)
P_d = stage discharge pressure, kPa (psia)
P_s = stage suction pressure, kPa (psia)
k = ratio of gas specific heats, $\frac{c_p}{c_v}$
c_p = specific heat at constant pressure
c_v = specific heat at constant volume

$$T_d = T_s \left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}$$
$$T_d = (560)\left(\frac{1015}{815}\right)^{\frac{126-1}{126}}$$
$$T_d = 586^{\circ}\text{R}$$
$$T_d = 126^{\circ}\text{F}$$

Step 2. Calculate the volumetric efficiency (assume 4 percent loss)

$$E_v = 100 - a - R - C\left[\left(R^{\frac{1}{k}}\right)\left(\frac{Z_s}{Z_d}\right) - 1\right]$$

where:

a = manufacturer's recommended valve losses (use 4 if unknown), percent
E_v = volumetric efficiency, percent
R = compression ratio (P_d/P_s) of the compressor stage
C = cylinder clearance, percent of piston displacement
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
k = ratio of specific heats, $\frac{c_p}{c_v}$
q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM)
PD = piston displacement, m^3/h (ACFM)

$$E_v = (100 - a) - R - C\left[\left(R^{\frac{1}{k}}\right)\frac{Z_s}{Z_d} - 1\right]$$
$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}}$$

where:

n = number of stages

$$R = \left[\frac{1015}{815}\right]^{\frac{1}{1}} = 1.245$$
$$E_v = 96 - 1.245 - 17.92\left[(1.245)^{\frac{1}{126}}\left(\frac{0.88}{0.85}\right) - 1\right]$$
$$E_v = 90.6$$

Step 3: Calculate the required clearance

Double-acting cylinder (sum of head-end and crank-end displacement)

$$\mathrm{PD} = \frac{\left[2(d_c)^2 - (d_r)^2\right](s)(R_c)}{2200}$$
$$\mathrm{PD} = \frac{\left[2(7)^2 - (1.75)^2\right](6)(900)}{2200}$$
$$q_a = E_v \times \mathrm{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(Actual ft^3/min))
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

$$\mathrm{PD} = 233$$
$$\mathrm{PD}_{tot} = (6)(233) = 1398\ \text{ACFM}$$

$$q_a = (E_v)(\mathrm{PD})$$
$$q_a = (0.906)(1398)$$
$$q_a = 1267\ \text{ACFM at suction conditions}$$

Convert to standard conditions:

$$Q_g = 0.051\frac{q_a P_s}{T_s Z_s}$$

where:

Q_g = gas flow rate, stdm^3/h (MMSCFD)

$$Q_g = \frac{(0.051)(q_a P_s)}{T_s Z_s}$$

$$Q_g = \frac{(0.051)(1267)(815)}{(560)(0.88)}$$

$$Q_g = 106.9 \text{ MMSCFD}$$

At present operating conditions, the throughput is too high. One can decrease throughput by reducing speed, increasing clearance, which will reduce volumetric efficiency, using a thicker cylinder liner to reduce cylinder volume, or lowering suction pressure.

Step 4: Calculate required rpm to give desired throughput

$$Q_g = 0.051 \frac{q_a P_s}{T_s Z_s}$$

where:

Q_g = gas flow rate, stdm^3/h (MMSCFD)

$$100 = \frac{(0.051)(q_a)(815)}{(560)(0.88)}$$

$$q_a = 1186\ cfm$$

$$q_a = E_v \times \text{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(actual ft^3/min))
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

$$\text{PD}_{\text{tot}} = \frac{q_a}{0.906} = \frac{1186}{0.906} = 1309 \text{ ACFM}$$

$$\text{PD} = \frac{\text{PD}_{\text{tot}}}{6} = \frac{1309}{6} = 218 \text{ ACFM}$$

$$\text{PD} = \frac{\left[2(d_c)^2 - (d_r)^2\right](s)(R_c)}{2200}$$

$$R_c = \frac{(PD)(2200)}{s\left[2(d_c)^2 - (d_r)^2\right]}$$

$$R_c = \frac{(218)(2200)}{6\left[2(7)^2 - (1.75)^2\right]}$$

$$R_c = 842 \text{ or} \sim 850 \text{ rpm is the suitable speed}$$

Step 5. Calculate clearance to reduce the throughput

Calculate the clearance that would be needed to reduce the throughput from 106.9 MMSCFD to 100 MMSCFD: Keep PD_{tot} constant but lower the efficiency.

$$q_a = E_v \times \mathrm{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(actual ft^3/min))
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

$$E_v = \frac{q_a}{\mathrm{PD}} = \frac{1186}{1398} = 0.848$$

Now backcalculate for the clearance that must be added to produce this volumetric efficiency.

$$E_v = 100 - a - R - C\left[\left(R^{\frac{1}{k}}\right)\left(\frac{Z_s}{Z_d}\right) - 1\right]$$

where:

a = manufacturer's recommended valve losses (use 4 if unknown), percent
E_v = volumetric efficiency, percent
R = compression ratio (P_d/P_s) of the compressor stage
C = cylinder clearance, percent of piston displacement
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
k = ratio of specific heats, $\frac{c_p}{c_v}$
q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(actual ft^3/min))
PD = piston displacement, m^3/h (ACFM)

$$84.8 = 96 - (1.245) - C\left[(1.245)^{\frac{1}{1.26}}\left(\frac{0.88}{0.85}\right) - 1\right]$$
$$C = 43.0$$

Step 6. Calculate the liner size required to reduce piston displacement

Assume E_v remains constant. This may have to be determined once a drawing of the specific cylinder and liner is available. However, it should not vary greatly. The PD required is:

$$q_a = E_v \times \mathrm{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(actual ft^3/min))
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

$$\text{PD} = \frac{1186}{0.906} = 1309 \text{ ACFM}$$

$$\frac{1309}{6} = \frac{\left[2(d_c)^2 - (1.75)^2\right](6)(900)}{2200}$$

$$d_c = 6.78 \text{ in}$$

Step 7. Calculate the rod load

$$\text{RL}_c = a_p(P_d - P_s) + a_r P_s$$

$$\text{RL}_c = \pi\left(\frac{7}{2}\right)^2 (1015 - 815) + \pi\left(\frac{1.75}{2}\right)^2 (815) = 9657 \text{ lb}$$

$$\text{RL}_t = a_p(P_d - P_s) - a_r P_d$$

where:

RL_c = rod load in compression, Newton (lb)
RL_t = rod load in tension, Newton (lb)
a_p = cross-sectional area of piston, mm^2 (in^2)
P_u = pressure in unloaded end, kPa (psia)
P_d = stage discharge pressure, kPa (psia)
P_s = stage suction pressure, kPa
a_r = cross-sectional area of rod, mm^2/(in^2)

$$\text{RL}_t = a_p(P_d - P_s) + a_r P_d$$

$$\text{RL}_t = \pi\left(\frac{7}{2}\right)^2 (1015 - 815) + \pi\left(\frac{1.75}{2}\right)^2 (1015) = 5255 \text{ lb}$$

The calculated rod loads for both the compression and tension modes are within the 11,200 N maximum rod load limit, and there is load reversal.

Step 8. Calculate the required power needed for the given condition

$$\text{BHP} = 0.0857[Z_{av}]\left[\frac{(Q_g)(T_s)}{E_a E_m}\right]\left[\frac{k}{(k-1)}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right]$$

where:

BHP = brake power, kW (hp)
Q_g = gas flow rate, stdm^3/h (MMSCFD)

T_s = stage suction temperature, K (°R)
Z_{av} = average of suction and discharge compressibility factor, $\frac{(Z_s + Z_d)}{2}$
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
E_m = mechanical efficiency of units,
- High-speed reciprocating unit use 0.93 to 0.95.
- Low-speed reciprocating unit use 0.95 to 0.98

Ea = adiabatic efficiency, 0.85 to 0.90
k = ratio of specific heats, $\frac{c_p}{c_v}$
P_s = stage suction pressure, kPa (psia)
P_d = stage discharge pressure, kPa (psia)

$$Z_{av} = \left(\frac{0.88 + 0.85}{2}\right) = 0.865$$
$$\text{BHP} = 0.0857[0.865]\left[\frac{(100)(560)}{(0.87)(0.94)}\right]\left[\frac{1.26}{1.26 - 1}\right]\left[\left(\frac{1015}{815}\right)^{\frac{1.26-1}{1.26}} - 1\right]$$
$$\text{BHP} = 1143 \text{ hp}$$

This is less than the available BHP from existing engine.

Step 9. Calculate the lowest suction pressure

If we use the minimum clearance,

$$q_a = 1267 \text{ ACFM}$$
$$Q_g = 0.051\frac{q_a P_s}{T_s Z_s}$$

where:

Q_g = gas flow rate, stdm^3/h (MMSCFD)

Assume $Z_s = 0.88$ at the new lower P_s.

$$P_s = \frac{(100)(560)(0.88)}{(0.51)(1267)} = 762.6 \text{ psia}$$
$$P_s = 747.9 \text{ psig}$$

It would be possible to recalculate this by choosing a new value for Z_s and calculating a new E_v for this condition, but the results will not change dramatically. By inspection, neither power, rod load, nor discharge temperature will limit this suction pressure.

Example 9.2 Compressor Design (SI Units)
Given:
Late in the field's life 118,000 m^3/h must be compressed.

Upstream of the separator, gas pressure is 5500 kPa(g) at 38°C. The desired discharge pressure is 6900 kPa(g). An engine-driven separable compressor is available from surplus. The engine is rated for 1190 kW at 900 rpm. Power is proportional to speed. The compressor frame has six 7-inch bore by 6.0-inch stroke double-acting cylinders with a minimum clearance of 17.92%, a rod load limit of 25,000 lb. and a rod diameter of 1.75 inch.

Assume $k=1.26$, $S=0.88$, $Z_s=0.88$ and $Z_d=0.85$.

Determine:

- Discharge temperature
- Volumetric efficiency
- Required clearance
- Rod load
- Required power for given conditions

The lowest suction pressure at which this unit can compress 100 MMSCFD.

Solution:

Step 1. Estimate gas discharge temperature

$$T_d = T_s \left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}$$

T_d = stage discharge temperature, K (°R)
T_s = stage suction temperature, K (°R)
P_d = stage discharge pressure, kPa (psia)
P_s = stage suction pressure, kPa (psia)
k = ratio of specific heats, $\frac{c_p}{c_v}$
c_p = specific heat at constant pressure
c_v = specific heat at constant volume

$$T_d = T_s \left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}$$
$$T_d = (311)\left(\frac{7003}{5603}\right)^{\frac{1.26-1}{1.26}}$$
$$T_d = 326k$$
$$T_d = 53^\circ\mathrm{C}$$

Step 2. Calculate the volumetric efficiency (assume 4 percent loss)

$$E_v = 100 - a - R - C\left[\left(R^{\frac{1}{k}}\right)\left(\frac{Z_s}{Z_d}\right) - 1\right]$$

where:

a = manufacturer's recommended valve losses (use 4 if unknown), percent
E_v = volumetric efficiency, percent

R = compression ratio (P_d/P_s) of the compressor stage
C = cylinder clearance, percent of piston displacement
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
k = ratio of specific heats, $\frac{c_p}{c_v}$
q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h (ACFM(actual ft^3/min))
PD = piston displacement, m^3/h (ACFM)

$$E_v = (100-4) - R - C\left[\left(R^{\frac{1}{k}}\right)\frac{Z_s}{Z_d} - 1\right]$$
$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}}$$

where:

n = number of stage

$$R = \left[\frac{7003}{5603}\right]^{\frac{1}{1}} = 1.250$$
$$E_v = 96 - 1.250 - 17.92\left[(1.250)\frac{1}{1.26}\left(\frac{0.88}{0.85}\right) - 1\right]$$
$$E_v = 90.5$$

Step 3: Calculate the required clearance

$$PD = \frac{\left[2(d_c)^2 - (d_r)^2\right](s)(R_c)}{353{,}650}$$
$$PD = \frac{\left[2(178)^2 - (44)^2\right](152)(15)}{353{,}650}$$
$$q_a = E_v \times PD$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h (ACFM)

$$\text{PD} = 396$$
$$\text{PD}_{tot} = (6)(396) = 2376 \text{ m}^3/\text{hr}$$

$$q_a = (E_v)(\text{PD})$$
$$q_a = (0.905)(2376)$$
$$q_a = 2150 \text{ m}^3/\text{hr at suction conditions}$$

Convert to standard conditions:

$$Q_g = 2.85\frac{(q_a P_s)}{T_s Z_s}$$

$$Q_g = 2.85\frac{(2150)(5603)}{(311)(0.88)}$$

$$Q_g = 125{,}450 \text{ stdm}^3/\text{hr}$$

At present operating conditions, the throughput is too high. One can decrease throughput by reducing speed, increasing clearance, which will reduce volumetric efficiency, using a thicker cylinder liner to reduce cylinder volume, or lowering suction pressure.

At present operating conditions, the throughput is too high. One can decrease throughput by reducing speed, increasing clearance, which will reduce volumetric efficiency, using a thicker cylinder liner to reduce cylinder volume, or lowering suction pressure.

Step 4: Calculate required rpm to give desired throughput

$$118{,}000 = \frac{(2.85)(q_a)(5603)}{(311)(0.88)}$$

$$q_a = 2022 \text{ m}^3/\text{hr}$$

$$q_a = E_v \times \text{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h

$$\text{PD}_{\text{tot}} = \frac{q_a}{0.906} = \frac{2022}{0.905} = 2234 \text{ m}^3/\text{hr}$$

$$\text{PD} = \frac{\text{PD}_{tot}}{6} = \frac{2234}{6} = 327 \text{ m}^3/\text{hr}$$

$$\text{PD} = \frac{\left[2(d_c)^2 - (d_r)^2\right] sR_c}{353{,}650}$$

$$R_c = \frac{(\text{PD})(353,650)}{s\left[2(d_c)^2 - (d_r)^2\right]}$$

$$R_c = \frac{(372)(353,650)}{152\left[2(178)^2 - (44)^2\right]}$$

$$R_c = 14.1 \text{ rps } = 845 \text{ rpm or } \sim 850 \text{ rpm is the suitable speed}$$

Step 5. Calculate clearance to reduce the throughput

Calculate the clearance that would be needed to reduce the throughput from 125,450 std. m^3/h to 118,000 std. m^3/h. Keep PD_{tot} constant but lower the efficiency.

$$q_a = E_v \times \mathrm{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h

$$E_v = \frac{q_a}{\mathrm{PD}} = \frac{2022}{2376} = 0.851$$

Now back calculate for the clearance that must be added to produce this volumetric efficiency.

$$E_v = 100 - a - R - C\left[\left(R^{\frac{1}{k}}\right)\left(\frac{Z_s}{Z_d}\right) - 1\right]$$

where:

a = manufacturer's recommended valve losses (use 4 if unknown), percent
E_v = volumetric efficiency, percent
R = compression ratio (P_d/P_s) of the compressor stage
C = cylinder clearance, percent of piston displacement
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
k = ratio of specific heats, $\frac{c_p}{c_v}$
q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h
PD = piston displacement, m^3/h

$$85.1 = 96 - (1.250) - C\left[(1.250)^{\frac{1}{1.26}}\left(\frac{0.88}{0.85}\right) - 1\right]$$
$$C = 40.9$$

Step 6. Calculate the liner size required to reduce piston displacement

Assume E_v remains constant. This may have to be determined once a drawing of the specific cylinder and liner is available. However, it should not vary greatly. The PD required is:

$$q_a = E_v \times \mathrm{PD}$$

where:

q_a = gas flow rate at suction conditions of temperature and pressure, m^3/h
E_v = volumetric efficiency, percent
PD = piston displacement, m^3/h

$$\text{PD} = \frac{2022}{0.905} = 2234 \text{ m}^3/\text{hr}$$

$$\text{PD} = \frac{\left[2(d_c)^2 - (d_r)^2\right] sR_c}{353{,}650}$$

$$\frac{2234}{6} = \frac{\left[2(d_c)^2 - (44)^2\right](152)(15)}{353{,}650}$$

$$d_c = 172.8 \text{ mm}$$

Step 7. Calculate the rod load

$$\text{RL}_c = \frac{a_p(P_d - P_s) + a_r P_s}{1000}$$

$$\text{RL}_c = \pi\left(\frac{178}{2}\right)^2 (7003 - 5603) + \pi\left(\frac{44}{2}\right)^2 (5603) = 43{,}358 \text{ Newton}$$
$$= 4421 \text{ kg force}$$

$$\text{RL}_t = \frac{a_p(P_d - P_s) - a_r P_s}{1000}$$

$$\text{RL}_t = \pi\left(\frac{178}{2}\right)^2 (7003 - 5603) - \pi\left(\frac{44}{2}\right)^2 (5603) = 26{,}319 \text{ Newton}$$
$$= 2684 \text{ kg force}$$

The calculated rod loads for both the compression and tension modes are within the 111,200 N maximum rod load limit, and there is load reversal.

Step 8. Calculate the required power needed for the given condition

$$\text{BHP} = 9.74 \times 10^{-5}[Z_{av}]\left[\frac{Q_g T_s}{E_a E_m}\right]\left[\frac{k}{(k-1)}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right]$$

where:

BHP = brake power, kW
Q_g = gas flow rate, stdm^3/h
T_s = stage suction temperature, K
Z_{av} = average of suction and discharge compressibility factor, $\frac{(Z_s + Z_d)}{2}$
Z_s = compressibility factor at suction conditions
Z_d = compressibility factor at discharge conditions
E_m = mechanical efficiency of units,

- High-speed reciprocating unit use 0.93 to 0.95.
- Low-speed reciprocating unit use 0.95 to 0.98

Ea = adiabatic efficiency, 0.85 to 0.90
k = ratio of specific heats, $\frac{c_p}{c_v}$
P_s = stage suction pressure, kPa
P_d = stage discharge pressure, kPa

$$\mathrm{BHP} = 9.74\times10^{-5}[Z_{av}]\left[\frac{Q_gT_s}{E_aE_m}\right]\left[\frac{k}{(k-1)}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}-1\right]$$
$$Z_{av} = \left(\frac{0.88+0.85}{2}\right) = 0.865$$
$$\mathrm{BHP} = 9.74\times10^{-5}[0.865]\left[\frac{(118,000)(311)}{(0.87)(0.94)}\right]\left[\frac{1.26}{1.26-1}\right]\times\left[\left(\frac{7003}{5603}\right)^{\frac{1.26-1}{1.26}}-1\right]$$
$$\mathrm{BHP} = 863\ \mathrm{kW}$$

This is less than the available BHP from existing engine.

Step 9. Calculate the lowest suction pressure

If we use the minimum clearance,

$$q_a = 2150\ \mathrm{m^3/hr}$$
$$Q_g = 2.85\frac{q_aP_s}{T_sZ_s}$$

where:

Q_g = gas flow rate, stdm^3/h

Assume $Z_s = 0.88$ at the new lower P_s.

$$Ps = \frac{(118,000)(311)(0.88)}{(2.85)(2150)} = 5270\ \mathrm{kPa}\ (\mathrm{abs})$$
$$Ps = 5167\ \mathrm{kPa}\ (g)$$

It would be possible to recalculate this by choosing a new value for Z_s and calculating a new E_v for this condition, but the results will not change dramatically. By inspection, neither power, rod load, nor discharge temperature will limit this suction pressure.

9.13 Exercises

1. List FOUR factors that influence the basic choice of a compressor cylinder design. Be brief
   A. ____
   B. ____
   C. ____
   D. ____
2. Name three factors to be considered in the choice of materials for the cylinder body
   A. ____
   B. ____
   C. ____
3. List two advantages of the valve-in-body design
   A. ____
   ____.
   B. ____
   ____.

4. List two advantages of the valve-in-head design

A. ____

____.

B. ____

____.

5. From the following characteristics listed, choose the ones that best relate to a dry liner and place the corresponding letter in the space provided. Repeat for the wet liner. Use each letter only once

Dry liner ______,______,______,______,______,

Wet liner ______,______,______,______,______,

A. Installation performed at the factory by heating the cylinder body in an oven

B. No direct contact with cylinder coolant

C. Absence of joints that seal both water and gas with a common gasket

D. Removal performed through the disassembly of studs and nuts

E. Gas load fully on the liner, thus becoming a structural consideration

F. Little stress due to gas load

G. Advantage in ease of replacement

6. Name three types of pistons used in compressors

A. ________________________.

B. ________________________.

C. ________________________.

7. Name two ways in which pulsation can be suppressed

A. ____

B. ____

8. Describe the special equipment, in addition to adequate piping, that is used to suppress pulsation

____.

____.

____.

____.

____.

9. Define *piston displacement*. ______________________________________

____.

____.

____.

10. Define *compressor ratio*. ______________________________________

____.

____.

____.

11. Define *specific heat*. ______________________________________

____.

____.

____.

12. Define *volumetric efficiency*. ______________________________________

____.

____.

____.

13. What does *"k" value* represent? ____________________________________
____.
____.
____.

14. Define *rod load.* ____
____.
____.
____.

15. Determine the capacity of a compressor cylinder. Solve the steps in the problem using an equation, or, if more practical, use the appropriate graph or chart. Round decimals to two digits

Given:

Cylinder diameter, Y_c:	12 in.
Stroke, S:	14 in.
Cylinder clearance, C_P:	12%[a]
Suction pressure, P_s:	100 psig
Suction temperature, T_s:	90°F
Discharge pressure, P_D:	300 psig
Rod diameter, Y_R:	3 in.
Speed:	300 rpm
Ratio of specific heats, k:	1.25
Location:	Texas
Allowable rod loads:	
	Tension, R_T: 50,000 pounds
	Compression, R_c: 50,000 pounds

[a]Clearance data is furnished by the manufacturer.

Determine:

(a) Piston displacement
(b) Compression ratio
(c) Volumetric efficiency
- For $k = 1.25$, $R =$, and $C_P = 12\%$.
- From from "quick look" graph, $E_v =$.

(d) Cylinder capacity
(e) BHP required
- For $k = 1.25$, $R =$.
- From "quick look" graph, BHP =.
- Correct the BHP/MMcf for suction temperature/pressure base
- Calculate the total BHP

(f) Check rod loads
(g) Determine the discharge temperature.
- For $k = 1.25$, $R = 2.74$, and $T_s = 90°F$.
- From graph, $T_D =$.

16. List five factors that affect compressor unit capacity and developed horsepower

A. ____

B. ____

C. ____

D. ____

E. ____

17. If all factors are held constant expect the percent clearance in the volumetric efficiency equation, the capacity of the cylinder is to this factor.

18. The capacity of a specific cylinder under fixed pressure and temperature conditions is a direction function of the amount of clearance in the cylinder expressed in ____.

19. List three ways in which the permanent clearance of a cylinder can be charged.

A. ____
____

B. ____
____

C. ____
____

20. List four ways in which clearance may be added

A. ____
____

B. ____
____

C. ____
____

D. ____
____

21. As more clearance is added to a compressor cylinder, the volumetric efficiency (increases/decreases). The capacity or PV area (increases/decreases)

22. As the suction pressure of a cylinder increases, the cylinder capacity will (increase/remain the same/decrease). The horsepower per million cubic feet of gas required for compression will (increase/remain the same/decrease).

23. Temperature affects cylinder capacity. As temperature increases, gas ______________________ and consequently, the cylinder capacity (increases/decreases).

24. Discharge pressure affects the capacity equation as a part of the compression ratio in the volumetric efficiency calculation. Generally, changes in discharge pressure will have little effect on cylinder capacity. Explain how changes in discharge pressure affect horsepower.

____.
____.
____.
____.
____.
____.

25. How does the ratio of specific heat affect cylinder performance?

____.
____.
____.
____

26. If a compressor operating at 300 rpm has its speed reduced to 270 rpm, the capacity will be reduced by _________ percent.

27. Explain why discharge pressure has less effect than suction on cylinder capacity.

____.
____.
____.
____.
____.
____.

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Application of compression theory and practical solutions

10

10.1 Overview

When selecting a reciprocating compressor for oil field production applications, it is necessary to determine if the proposed service requires features conforming to API STD 618, API SPEC 11P, or if the compressor is built to manufacturer standards, and modifications for a proposed oil field usage are satisfactory.

10.1.1 Process design data

The process design data required for selecting a reciprocating compressor is as follows:

Q_g = gas flow rate, stdm^3/h (MMSCFD)
P_s = stage suction pressure, kPa (psia)
P_d = stage discharge pressure, kPa (psia)
T_s = stage suction temperature, °C (°F)

Gas properties—it is best to know the composition of the gas to be compressed, but as a minimum the manufacturer will need to know the specific gravity of the gas and the amount of H_2S or CO_2 present. With this information it is possible to determine power and select the number of stages using the equations presented in Chapter 8 and in this chapter.

10.1.2 Choosing high speed vs. slow speed

When selecting a reciprocating compressor, the engineer must decide whether to use a high- or a slow-speed unit. Advantages and disadvantages of each type are discussed in Chapter 8. Other factors affecting the decision are as follows:

- Foundation considerations (dirt, gravel, shell, limestone, concrete, barge, or offshore platform).
- Transportation and/or lift equipment available to transport, install, and service the unit.
- Duration of unit service.
- Delivery and timing of start-up (integral units typically have longer manufacturing lead time).
- Existence of similar or like units that can share spare parts, maintenance training, special tools, and so on.
- Preference and experience of operational staff.

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00010-7

10.1.3 Process considerations

If a packaged compressor is desired, a flowsheet should be included showing the required items discussed in Chapter 8 and in this chapter.

In some small field applications, it may be acceptable to purchase a standard package from a manufacturer. However, in most instances it will be beneficial to specify the quality desired in fabrication, choice of materials, and coating systems. Often the project engineer will want to specify scrubber size and cooler size, along with specifying process instrumentation, and so on.

10.1.4 Evaluating alternatives

If a packaged compressor is desired, a flowsheet should be included showing the required items discussed in this chapter.

In some small field applications, it may be acceptable to purchase a standard package from a manufacturer. However, in most instances it will be beneficial to specify the quality desired in fabrication, choice of materials, and coating systems. Often the project engineer will want to specify scrubber size and cooler size, along with specifying process instrumentation, and so on.

10.2 Design considerations

10.2.1 General considerations

Solving compressor problems can be reduced to a relatively simple sequence of calculations by:

- Applying a few basic equations
- Utilizing data taken from "quick look" curves presented in this chapter
- Developing Reciprocating Compressors Performance Maps

10.2.2 Performance maps

Performance maps, a graphical representation of expected performance, can be developed for a specific reciprocating compressor with base conditions held constant. Performance maps plots suction pressure versus:

- Capacity (Fig. 10.1)
- Horsepower (Fig. 10.2)
- Gas discharge temperature
- Rod load

The map is only valid if the following are held constant:

- Gas discharge pressure
- Speed
- Constant clearance

All curves take on the same appearance regardless of staging.

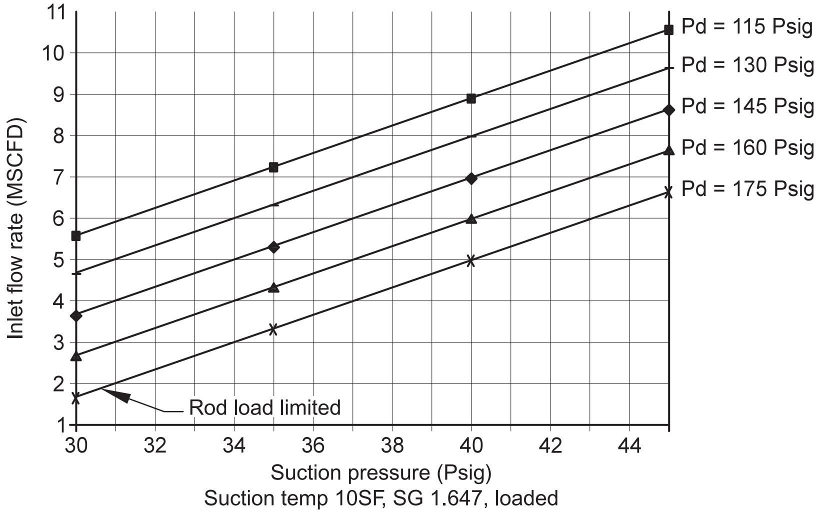


Fig. 10.1 Example reciprocating compressor performance map showing suction pressure vs. inlet flow rate.

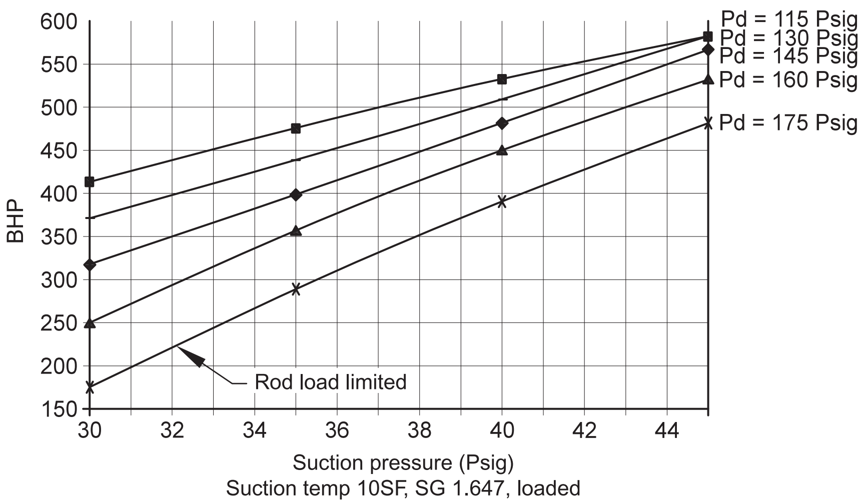


Fig. 10.2 Reciprocating compressor performance map showing suction pressure vs. brake horsepower.

The remainder of this chapter illustrates how compressor performance maps can be used to solve typical compressor problems.

Example 10.1. Determining Compressor Parameters
Given:

$FTP_i = 700$ psig
$FTP_f = 350$ psig (well will "water out")
$Q_g = 3.5$ MMSCFD
$S = 0.6$
$T = 80°F$
$k = 1.26$
Gas Sales Pressure = 1100 psig

Determine:
Determine compressor parameters necessary to develop a reciprocating compressor performance curve.
Solution:

(1) Determine the number of stages required to meet the "worst-case" conditions ($FTP_f = 350$ psig) by determining the total compressor ratio

$$R_T = \frac{P_d}{P_s} = \frac{1114.7}{364.7} = 3.06$$

Since $R_T < 5$ and $T < 110°F$, a single-stage compressor is selected.

(2) *Estimate a "rough" horsepower by using the "quick look" formula*. This allows us to select a frame in the horsepower range. Use the maximum suction pressure of 715 psia as this will result in the largest horsepower required.

(a) Calculate the compression ratio per stage for "initial" condition

$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \leq 3.5$$

Substituting

$$= \frac{1114.7}{714.7} = 1.56$$

(b) Estimate "rough" horsepower

$$\mathrm{BHP} = 22\,\mathrm{RnFQ_g}$$

where:

BHP = approximate BHP
R = ratio per stage
n = number of stages

$F =$ allowance for interstage pressure drop
$= 1.00$ for single stage
$= 1.08$ for two stage
$= 1.1\ 0$ for three stage
$Q_g =$ flow rate, MMSCFD

Substituting

$$\begin{aligned} \text{BHP} &= (22)(1.56)(1)(1)(3.5) \\ &= 120 \end{aligned}$$

As shown in Fig. 10.3, select an Aerial "JG/2"

(3) Select a "rough" cylinder diameter as a starting point

(a) Determine ACFM at suction conditions of 700 psig and 80°F

$$\begin{aligned} \text{ACFM} &= 19.6\left(\frac{Z_s T_s Q_g}{P_s}\right) \\ &= (19.6)\left(\frac{(.91)(540)(3.5)}{714.7}\right) \\ &= 47.17\,\text{ft}^3/\text{min} \end{aligned}$$

(b) Calculate piston displacement (PD)

By definition

$$\text{ACFM} = E_v\,(\text{PD})$$

Assume

$$E_v = 85\%\,\text{and solve for PD, thus}$$

$$\begin{aligned} \text{PD} &= \frac{\text{ACFM}}{E_v} \\ &= \frac{47.17}{08.5} \\ &= 55.49\,\text{ft}^3/\text{min} \end{aligned}$$

(c) Obtain data from Aerial Data Sheet No. 1 (Fig. 10.3)
Speed = 1000 rpm
Stroke = 3.5 in.
Piston Rod Diameter = 1.125 in.

(d) Assume the compressor to have two (2) identical double-acting cylinders. The piston displacement equation is as follows:

$$\begin{aligned} \text{PD} &= \left(\frac{2\left(d_c^2 - d_r^2\right)SN}{2200}\right)\begin{pmatrix}\text{Number of} \\ \text{Cylinder}\end{pmatrix} \\ &= \left(\frac{2d_c^2 - d_r^2 SN}{2200}\right)2 \end{aligned}$$

ARIEL COMPRESSORS	JGP/1 JGP/2	JGM/1 JGM/2	JG/2 JG/4	JGW/2 JGW/4	JGR/2 JGR/4
SPECIFICATIONS					
STROKE	3-	3-1/2-	3-1/2-	4-1/4-	4-1/4-
RPM	to 1800	to 1400	to 1400	to 1200	to 1200
PISTON SPEED, FPM	to 800	to 817	to 817	to 850	to 850
NUMBER OF THROWS	1 2	1 2	2 4	4 4	2 4
HORSE POWER	to 62.5 to 125	to 62.5 to 125	to 200 to 400	to 300 to 600	to 400 to 800
CRANK SHAFT CENTERLINE	9-1/4"from bottom	9-1/4"from bottom	10-1/4"from bottom	12"from bottom	12"from bottom
OVERALL WIDTH (MAXIMUM)	61" 88"	61" 88"	80"	87"	94"
OVERALL LENGTH (MAXIMUM)	33"	33"	32" 98"	38" 78"	38" 78"
WEIGHT (APPROXIMATE)	1200# 1500#	1200# 1500#	2000# 4000#	3000# 6000#	4000# 8000#
OIL PUMP CAPACITY, GPM	4	4	4 8	8 14	8 14
OIL HEAT REJECTION, BTU/HR	8.000	4.000	4.000 7.000	7.000 12.000	6.000 14.000
SUMP CAPACITY, GALLONS	2-1/2	2-1/2	4 10	6 18	8 18
INTERNAL ROD LOADS					
DOUBLE ACTING					
COMPRESSION+TENSION	10.000#	10.000#	18.000#	24.000#	38.000#
TENSION	5.000#	5.000#	8.000#	12.000#	18.000#
COMPRESSION	8.250#	8.250#	10.000#	15.000#	20.000#
SINGLE ACTING					
TENSION	8.000#	8.000#	8.000#	12.000#	16.000#
COMPONENTS					
CRANKSHAFT					
CRANKPIN DIAMETER	3"	3"	3"	4-1/8"	4-1/8"
JOURNAL DIAMETER	3-1/4"	3-1/4"	3-1/4"	4-1/8"	4-1/8"
THRUST BEARING	3-1/4"x2-1/2"	3-1/4"x2-1/2"	3-1/4"x2-1/2"	4-1/8"x2-7/8"	4-1/8"x2-7/8"
MAIN BEARING	3-1/4"x2	3-1/4"x2	3-1/4"x2	4-1/8"x2-1/2"	4-1/8"x2-1/2"
CONNECTING ROD (B TO C)	8-1/4"	8-1/4"	8-1/4"	10-1/4"	10-1/4"
CONNECTING ROD BEARINGS	3"x1-11/16"	3"x1-11/16"	3"x1-11/16"	4-1/8"x2-1/2"	4-1/8"x2-1/2"
CONNECTING ROD BOLT	1/2"(four)	1/2"(four)	1/2"(four)	6/8"(four)	6/8"(four)
CROSSHEAD SURFACE	5-3/4"x4-5/8"	5-3/4"x4-5/8"	5-3/4"x4-5/8"	7-5/8"x5-1/4"	7-5/8"x5-1/4"
FLOATING CROSSHEAD PIN	2-1/2"x4-5/32"	2-1/2"x4-5/32"	2-1/2"x4-5/32"	3-3/8"x8-25/32"	3-3/8"x8-25/32"
CROSSHEAD PIN BUSHINGS	NONE	NONE	(Two) 2-1/2"x1"	NONE	(Two) 3-3/8"x1-7/16"
CONNECTING ROD BUSHINGS	2-1/2"x2"	2-1/2"x2"	2-1/2"x2"	3-3/8"x2-3/4"	3-3/8"x2-3/4"
PISTON ROD	1-1/8"	1-1/8"	1-1/8"	1-1/2"	1-1/2"
MATERIAL					
CRANKCASE	Class 30 Grey Iron	Class 30 Grey Iron	Class 30 Grey Iron	Class 30 Grey Iron	Class 30 Grey Iron
BEARING CAPS	65-45-12 Ducting Iron	65-45-12 Ducting Iron	65-45-12 Ducting Iron	65-45-12 Ducting Iron	65-45-12 Ducting Iron
CROSSHEAD GUIDES	Integrated With Clearance	Integrated With Clearance	Class 30 Grey Iron	Class 30 Grey Iron	Class 30 Grey Iron
CRANKSHAFT	65-45-12 Ducting Iron Flash Chromed	65-45-12 Ducting Iron Flash Chromed	C-1045 Forged Steel	65-45-12 Ducting Iron Flash Chromed	65-45-12 Ducting Iron
SPACER BARS	None	None	Cold Drawn Steel	Cold Drawn Steel	Cold Drawn Steel
CONNECTING RODS	65-45-12 Ducting Iron	65-45-12 Ducting Iron	C-1045 Forged Steel	65-45-12 Ducting Iron	C-1045 Forged Steel
MAIN BEARINGS	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal
THRUST BEARINGS	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Bronze Plate	Bronze Plate
CONNECTING RODS BEARINGS	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal	Cleville 77 Tri-Metal
CROSSHEADS	Class 40 Grey Iron	Class 40 Grey Iron	65-45-12 Ducting Iron Dobble Surface	Class 40 Grey Iron	65-45-12 Ducting Iron Dobble Surface
CROSSHEAD PINS	ANSI C-8820 Surface Hardened to Plc 80	ANSI C-8820 Surface Hardened to Plc 80	ANSI C-8820 Surface Hardened to Plc 80	ANSI C-8820 Surface Hardened to Plc 80	ANSI C-8820 Surface Hardened to Plc 80
CROSSHEAD PIN BUSHINGS	None	None	SAE 990 Bronze	None	SAE 880 Bronze
PISTON RODS	SAE Series 4100 Steel	SAE Series 4100 Steel	SAE Series 4100 Steel	SAE Series 4100 Steel	SAE Series 4100 Steel
ROD PACKING RINGS	Filled Teflon Cast Iron Back Up Flange	Filled Teflon Cast Iron Back Up Flange	Filled Teflon Cast Iron Back Up Flange	Filled Teflon Cast Iron Back Up Flange	Filled Teflon Cast Iron Back Up Flange
ALL STRESSED BOLTS	Grade 8 Alloy Steel	Grade 8 Alloy Steel	Grade 8 Alloy Steel	Grade 8 Alloy Steel	Grade 8 Alloy Steel
PISTONS	Class 30 Gray Iron	Class 30 Gray Iron	Class 30 Gray Iron	Class 30 Gray Iron	Class 40 Gray Iron
PISTON RINGS	Filled Teflon	Filled Teflon	Filled Teflon	Filled Teflon	Filled Teflon
CYLINDERS	Ducting Iron	Ducting Iron	Class 40 Gray Iron Ducting Iron	Class 40 Gray Iron Ducting Iron	Class 40 Gray Iron Ducting Iron
CYLINDER HEAD GASKETS	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"
VALVE GASKETS	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"	AMG "Steel Tile"
VALVE CAP "O" RINGS	Viton	Viton	Viton	Viton	Viton

Fig. 10.3 Aerial compressor data sheet no. 1.

Rearranging and solving for cylinder diameter

$$d_c = \left[\frac{\left[\frac{(\mathrm{PD})2200}{\mathrm{SN}(2)} + d_r^2\right]}{2}\right]^{0.5}$$

Substituting

$$= \left[\frac{\left[\frac{(55.49)2200}{(3.5)(1000)(2)} + (1.125)^2\right]}{2}\right]^{0.5}$$

$$= 3.06\,\text{inches}^2$$

(e) Select a cylinder from Aerial Data Sheet No. 2 (Fig. 10.4)

Choose a 3.375-in. cylinder with 1320 psi MAWP

(4) Calculate the PD for the selected cylinders

$$\mathrm{PD} = \left(\frac{2d_c^2 - d_r^2 SN}{2200}\right)\left(\begin{matrix}\text{Number of}\\ \text{Cylinder}\end{matrix}\right)$$

$$= \left(\frac{(2)(3.375)^2 - (1.125)^2(3.5)(1000)}{2200}\right)(2)$$

$$= 68.5\,\mathrm{ft}^3/\mathrm{min}$$

(5) Calculate the gas discharge temperature

$$T_d = T_s\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{Knp}}$$

$$= 540\left(\frac{1114.7}{714.7}\right)^{\frac{1.26-1}{(1.26)(1)}}$$

$$= 592^\circ\mathrm{R}\ \text{or}\ 132^\circ\mathrm{F}$$

(6) Calculate actual volumetric efficiency (E_v)

(a) Determine compressibility factor at suction/discharge

Z_s @ 700 psig and 80°F = 0.91 (Fig. 10.5)

Z_d @ 1100 psig and 132°F = 0.88 (Fig. 10.5)

(b) Determine average normal clearance (C)

From Aerial Data Sheet No. 2 (Fig. 10.4) a 3.375-in. cylinder has an average normal clearance of 14.55%

RDP — RATED DISCHARGE PRESSURE
MAWP — MAXIMUM ALLOWABLE WORKING PRESSURE
HTP — HYDROSTATIC TEST PRESSURE

SEE PAGE 12 — NOTES 7,8,9,10,11,12,13,15,20

ARIEL MODEL JG COMPRESSOR
3-1/2" STROKE

(1) 3# JG SPECIAL
\ 2000# RDP
2000# MAWP
3300# HTP

(2) 2-3/4"JGT
1310# RDP
1440# MAWP
2160# HTP0

CYLINDER CLASS	CYLINDER BORE INCHES	PISTON AREA SQ. IN.		PISTON DISPLACEMENT CFM AT 1400 RPM			NORMAL CLEARANCE, %			SPACERS ADDED CLEARANCE % PER SPACER		M.E. VARIABLE VOLUME UNLOADERS % TOTAL CLEARANCE				
		M.E.	C.E.	M.E.	C.E.	TOTAL	M.E.	C.E.	AVG.	M.E.	C.E.	C.E.	M.E.(MIN.)	AVG.(MIN.)	M.E.(MAX.)	AVG.(MAX.)
3" JG(1) 1545# RDP	2-3/4" 2.750"	5.94	4.95	16.64	14.03	30.87	17.03	21.20	18.91	8.54	10.28	—	—	—	—	—
1700# MAWP 2550# HTP	3" 3.000"	7.07	6.07	20.05	17.23	37.27	13.87	16.90	15.28	7.18	8.37	—	—	—	—	—
3-5/9" JG 1200# RDP	3-3/8" 3.375"	8.95	7.95	25.37	22.56	47.92	12.80	16.50	14.55	13.47	15.17	16.50	13.68	15.13	62.95	41.11
1320# MAWP 1980# HTP	3-5/8" 3.625"	10.32	9.33	29.27	26.45	55.72	10.70	13.25	11.95	11.66	12.93	13.25	11.64	12.42	54.18	34.76
4-1/9" JG 1025# RDP	3-7/8" 3.675"	11.79	10.80	33.45	30.63	64.07	12.40	14.73	13.50	9.08	9.92	14.73	13.58	14.13	68.19	42.63
1125# MAWP 1690# HTP	4-1/8" 4.125"	13.38	12.37	37.90	35.08	72.98	10.19	12.05	11.08	0.01	9.68	12.05	11.24	11.63	59.42	36.65
5-1/8" JG 620# RDP	4-3/4" 4.750"	17.72	18.73	50.26	47.44	97.89	15.45	16.77	16.09	9.25	9.81	16.77	16.45	16.61	61.98	40.03
680# MAWP 1020# HTP	5-1/8" 5.125"	20.63	19.63	58.50	55.68	114.19	12.47	13.45	12.95	7.95	8.35	13.45	13.34	13.39	52.45	33.43
6-1/2" JG 455# RDP	6-1/8" 6.125"	23.46	26.47	83.08	80.74	184.30	13.65	14.22	13.93	6.89	7.13	14.22	15.06	14.65	70.18	42.68
500# MAWP 750# HTP	6-1/2" 6.500"	33.18	32.19	94.11	91.29	185.40	11.35	11.78	11.57	6.12	6.31	11.78	12.61	12.20	61.57	37.05
7-1/2" JG 300# RDP	7-1/8" 7.125"	39.87	36.88	113.07	110.26	223.33	13.28	13.21	13.25	6.33	6.49	13.21	14.81	14.02	72.73	43.34
330# MAWP 495# HTP	7-1/2" 7.500"	44.18	43.18	125.29	122.47	247.78	10.82	10.81	10.81	6.71	6.84	10.81	12.19	11.51	64.46	37.94
8-7/8" JG 205# RDP	8-1/2" 8.500"	56.75	55.75	180.93	168.11	319.04	15.28	14.87	15.08	6.65	6.77	14.87	16.90	15.89	91.99	53.77
230# MAWP 345# HTP	8-7/8" 8.875"	61.96	60.87	178.44	172.62	348.06	12.84	12.52	12.68	6.10	6.20	12.52	14.32	13.43	83.20	48.15
11" JG 125# RDP	10-1/2" 10.500"	86.56	85.80	245.57	242.75	488.32	15.80	15.37	15.59	6.10	6.17	15.37	17.46	16.42	73.00	44.36
150# MAWP 225# HTP	11" 11.000"	95.03	94.04	286.51	266.70	536.21	12.98	12.67	12.83	5.55	5.61	12.67	14.50	13.59	65.11	39.02
13-1/2" JG 90# RDP	13" 13.000"	132.73	131.74	378.43	373.61	750.04	15.18	14.89	15.08	—	—	14.99	16.64	15.92	74.66	44.94
115# MAWP 175# HTP	13-1/2" 13.500"	143.14	142.14	405.94	403.12	809.08	12.64	12.56	12.60	—	—	12.56	14.19	13.38	67.81	40.28
3 JGT(2) 1025# RDP	2-3/4" 2.750"	5.94	—	15.84	—	—	10.78	—	—	8.09	—	—	—	—	—	—
1125# MAWP 1890# HTP	3" 3.000"	7.07	—	20.05	—	—	9.64	—	—	9.89	—	—	—	—	—	—
3-5/9" JGT 1200# RDP	3-3/8" 3.375"	9.95	—	25.37	—	—	12.33	—	—	6.59	—	—	—	—	—	—
1320# MAWP 1980# HTP	3-5/8" 3.625"	10.32	—	29.27	—	—	10.16	—	—	5.71	—	—	—	—	—	—

Fig. 10.4 Aerial compressor data sheet no. 2.

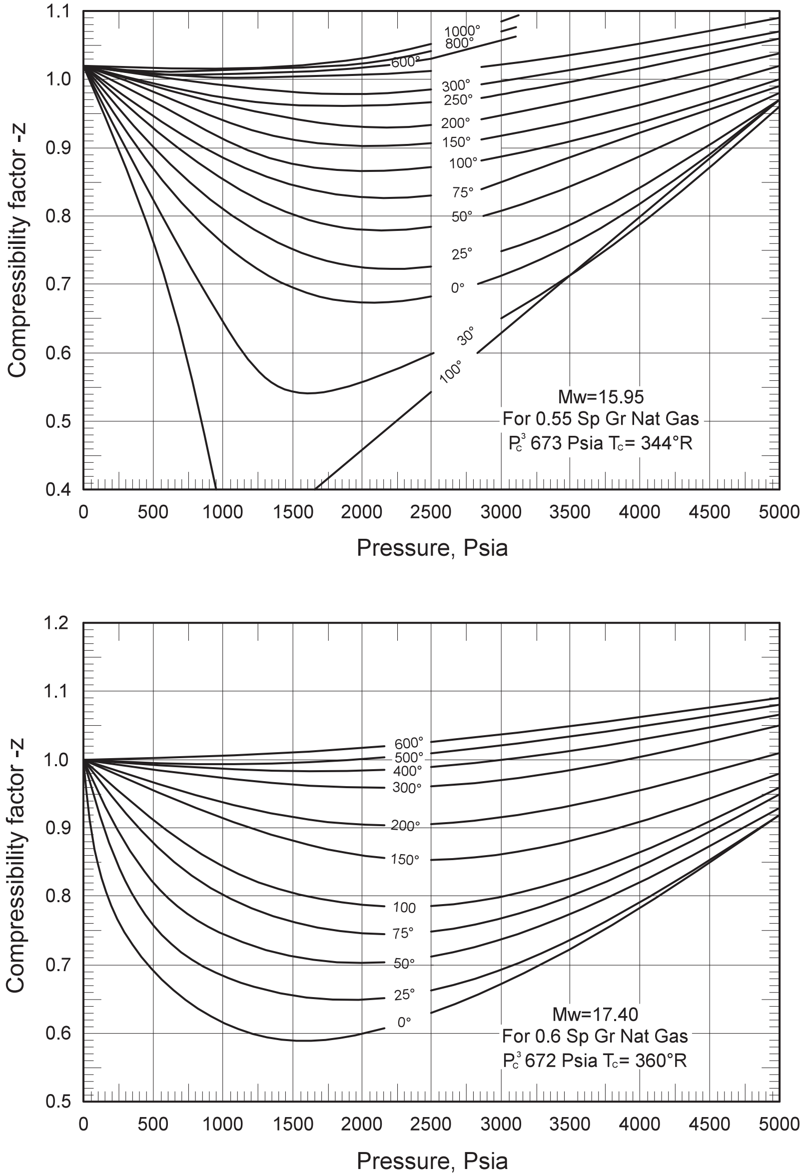
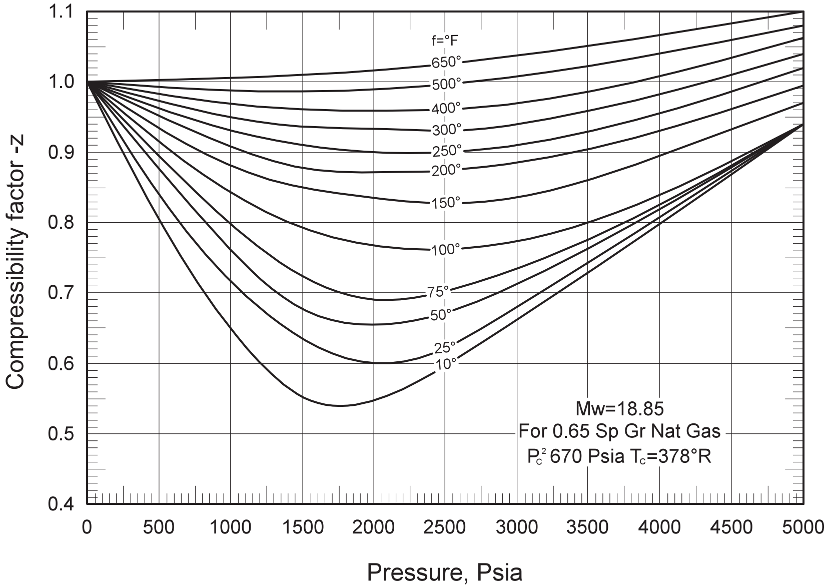


Fig. 10.5 "Quick-look" compressibility factors.

(Continued)



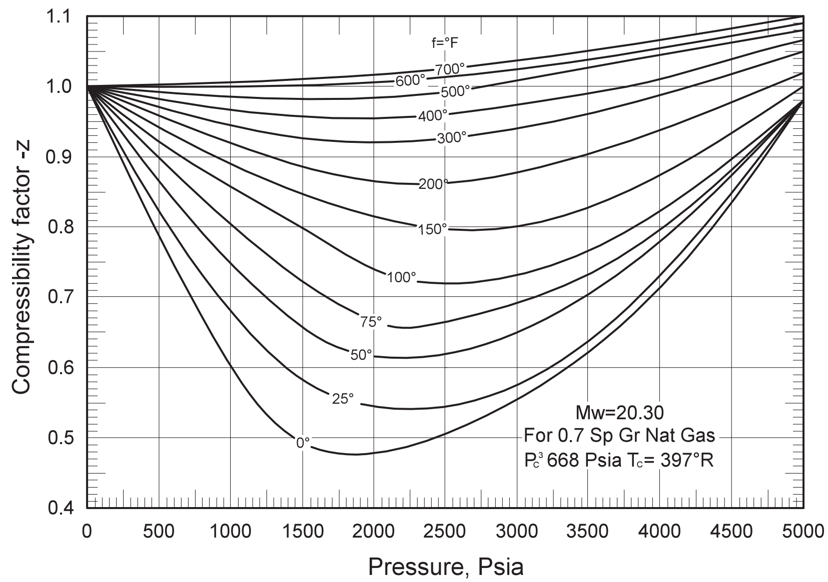
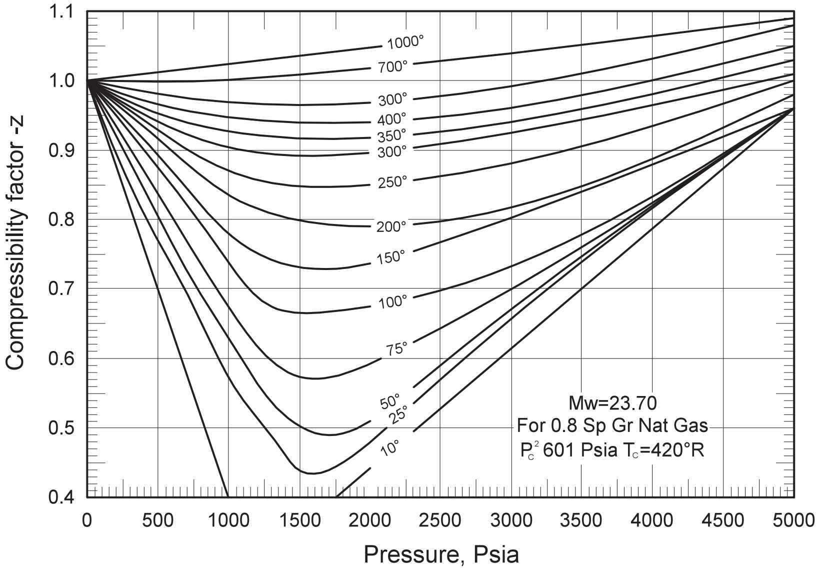


Fig. 10.5—cont'd



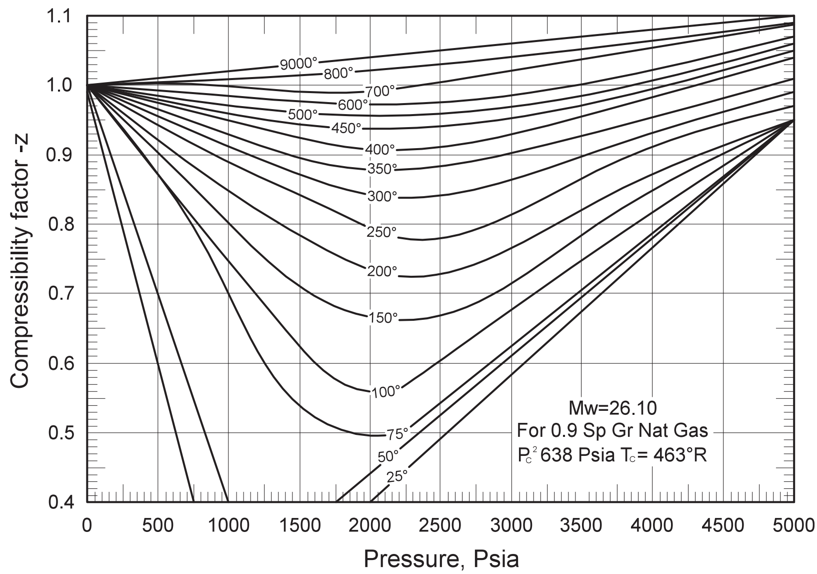


Fig. 10.5—cont'd

(c) Calculate volumetric efficiency (E_v) (assume high-speed separable reciprocating compressor)

$$
\begin{aligned}
E_v &= 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right] \\
&= 96 - 1.56 - 14.55\left[\left(\frac{0.91}{0.88}\right)(1.56)^{\frac{1}{1.26}} - 1\right] \\
&= 86\%
\end{aligned}
$$

Note: For low-speed integral compressors slippage of gas past the piston rings is 0%, for high speed separable 1% and 5% for nonlubricated compressors

(7) Calculate capacity with new cylinder

(a) Determine ACFM

$$
\begin{aligned}
\text{ACFM} &= (\text{PD})E_V \\
&= (68.5)(0.86) \\
&= 58.91\ \text{ft}^3/\text{min}
\end{aligned}
$$

(b) Determine capacity in MMSCFD

$$
\begin{aligned}
Q_g &= (0.051)\left(\frac{(ACFM)(P_2)}{T_s Z_s}\right) \\
&= (0.051)\left(\frac{(58.91)(714.7)}{(540)(0.91)}\right) \\
&= 4.36\ \text{MMSCFD}
\end{aligned}
$$

The 3.375-in. cylinder may be too large for this application. Therefore investigate the next smaller cylinder size, that is, 3.00 in.

(8) Repeat Steps 4, 6 and 7

(a) Step 4: PD = 53.3 ft^3/min

(b) Step 6: $E_v = 85.67\%$

(c) Step 7: $Q_g = 3.38$ MMSCFD

The 3-in. cylinder is probably too small for this application. Therefore a decision must be made based on additional information, such as

- Well may flow >3.5 MMSCFD at 700 psig
- Discharge pressure may become greater than the working pressure on the 3.375-in. cylinder
- Availability of the cylinders

For this problem we will choose the larger cylinder, that is, 3.375 in.

(9) Calculate rod loads in tension

(a) Determine rod load at 700 psig

$$\begin{aligned}RL_t &= P_d\left(A_p - A_r\right) - P_sA_p\\&= 1114.7\left[\left(\frac{\pi(3.357)^2}{4}\right) - \left(\frac{\pi(1.125)^2}{4}\right)\right]\\&= 714.1\left(\frac{\pi(3.357)^2}{4}\right)\\&= 2485 \text{ Ib}\end{aligned}$$

(b) Determine rod load at 350 psig

$$\begin{aligned}RL_t &= 8747 - 3131\\&= 5616\end{aligned}$$

Conclusion: From Aerial Data Sheet No. 1 (Fig. 10.1) allowable rod load in tension is 8000 lbs. maximum expected rod loads are well within the allowables.

(10) Determine accurate break horse required

$$\begin{aligned}\text{BHP} &= 0.0857\left[Z_{AV}^{\frac{1}{k}}\right][Z_s]^{\frac{K-1}{K}}\left[\frac{Q_gT_s}{E_mE_a}\right]\\&\quad\times\left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{kn_p}} - 1\right]\\&= 0.087\left[0.98^{\frac{1}{1.26}}\right][0.91]^{\frac{1.26-1}{1.26}}\left[\frac{(4.36)(540)}{(.93)(.85)}\right]\\&\quad\times\left[\frac{(1.26)(.1)}{1.26-1}\right]\left[\left(\frac{114.7}{714.7}\right)^{\frac{1.26-1}{714.7}} - 1\right]\\&= 134\,\text{BHP}\end{aligned}$$

Example 10.2. Developing a Reciprocating Compressor Performance Map

Given:

$P_d = 1100$ psig
Speed = 1000 rpm
Clearance = Minimum

Determine:

Develop a performance curve that can be used for the life of the project.

Solution:

Perform the same calculations as in Example 10.1 pressures of 350 psig (final) and at 525 psig (mid-point value)

P_S	R_C	T_D	Z_S	Z_D	VE	Q	BHP
700	1.56	132	0.91	0.88	86.5	4.36	134
525	2.06	167	0.93	0.91	80.4	3.01	135
350	3.06	220	0.95	0.94	68.7	1.71	115

(1) Plot suction pressure (P_s) versus capacity (Q_g) on linear graph (Fig. 10.6).
(2) Select a driver

Brake horsepower of the driver should be derated for

- *Friction* (use a 0.93 multiplier)
- *Accessory drives* such as a belt-driven cooler; use a multiplier of 0.97 in order to arrive at the horsepower available for compression

Pick an engine, derate it, and plot the results on the performance curve.

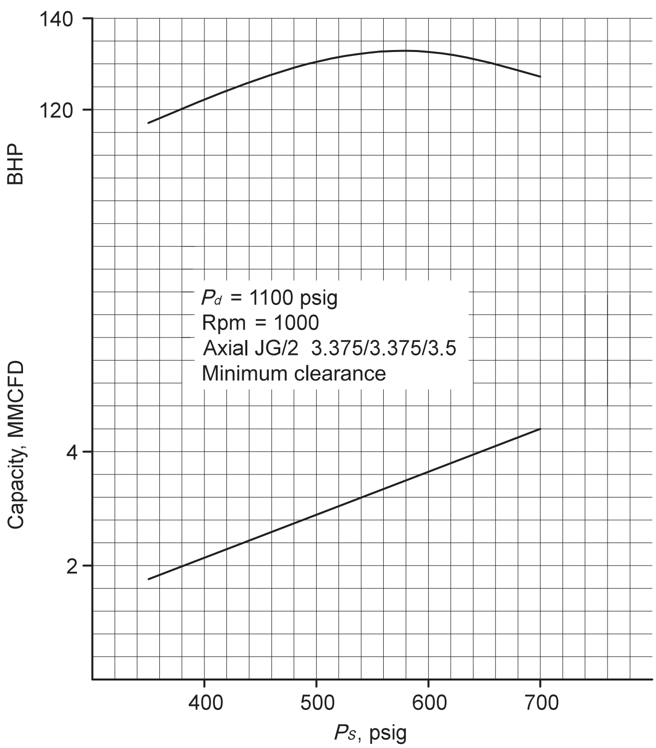


Fig. 10.6 Performance map for Example 10.2.

The caterpillar G342NA, refer to Manufacturer's Data Sheet (Fig. 10.7) has a continuous shaft rating at 1000 rpm of 160 hp.

Derating for friction and driving a cooler fan results in 144 hp. available for compression.

Example 10.3. Effects on Compressor Performance Due to a Reduction in Capacity
Given:

- Single-stage compressor performance curve (Fig. 10.8)
- Compressor is compressing 3.6 MMSCFD at 600 psig. One well, producing 0.8 MMSCFD, is shut-in

MODEL	CONTINUOUS SHAFT RATINGS RPM						LIST PRICE
	800	1000	1200	1400	1600	1800	
VRG155[1]	---	---	20	22	25	27	
HR3G[4]	---	16	21	24	28	31	
3DAG[1]	---	22	25	29	32	35	
VRG220[1] (LC)	---	---	---	32	35	36	
G-2300[7] (LC)	---	---	28	32	35	39	
G-2300[7] (HC)	---	---	32	36	40	43	
4DAG[4] (HC)	---	29	34	39	473	47	
G-3400[7] (LC)	---	---	41	45	51	56	
VRG263[1]	---	---	39	47	54	61	
G-34007 (HC)	---	---	46	52	58	63	
VRG330[1]	---	---	---	54	60	67	
6DAG[4] (HC)	---	43	51	57	65	71	
336A-4A[3]	35	45	54	62	68	---	
3304NA[2] (LC)	---	(40)	50	59	69	78	
3304NA[2] (HC)	---	(52)	61	71	82	89	
HD504A-6A[5]	49	62	74	85	95	104	
3306NA[2] (LC)	---	(61)	72	89	103	118	
3306NA[2] (HC)	---	(78)	89	107	122	136	
FB17G[1] (LC)	72	90	109	124	138	---	
FB17G[1] (HC)	75	94	113	130	148	---	
HD800A-6A[5]	82	104	123	142	---	---	
3306TA[2] (LC)	---	(103)	122	146	---	---	
F1197G[1] (LC)	101	122	149	172	---	---	
THD800A-6A[5]	100	128	152	181	---	---	
F1197G[1] (HC)	112	137	162	185	---	---	
G342NA[2] (LC)	122	160	188	---	---	---	
G342NA[2] (HC)	141	162	212	---	---	---	
F1905G[1] (LC)	168	203	233	---	---	---	
F1905G[1] (HC)	172	208	239	---	---	---	
G342TA[2] (LC)	178	227	256	---	---	---	
G379NA[2] (LC)	174	230	282	---	---	---	
F2475G[1] (LC)	227	273	309	---	---	---	
G379NA[2] (HC)	202	259	310	---	---	---	
F2475G[1] (HC)	239	283	323	---	---	---	
F2895G[1] (LC)	278	337	376	---	---	---	
8G-510[3] (LC)	270	337	---	---	---	---	
G379TA[2] (LC)	265	352	400	---	---	---	
F2695G[1] (HC)	302	369	421	---	---	---	
G396NA[2] (LC)	249	348	423	---	---	---	
8G-510[3] (HC)	306	383	---	---	---	---	
F3521G[1] (LC)	337	405	456	---	---	---	
G396NA[2] (HC)	301	387	470	---	---	---	
L3711G[1] (LC)	337	411	476	---	---	---	
L3711G[1] (HC)	353	427	486	---	---	---	
F3521G[1] (HC)	366	448	512	---	---	---	
G399NA[2] (LC)	348	465	564	---	---	---	
IR2500[4]	421	527	575	---	---	---	
G398TA[2] (LC)	410	531	603	---	---	---	
F2895GSI[1] (LC)	405	506	607	---	---	---	
G399NA[2] (HC)	409	517	620	---	---	---	
8G-625[3] (LC)	439	548	---	---	---	---	
F5106G[1] (LC)	489	588	661	---	---	---	
F3521GSI[1] (LC)	492	615	738	---	---	---	
8G-825[4] (HC)	496	622	---	---	---	---	
F5106G[1] (HC)	529	634	741	---	---	---	
L5709G[1] (LC)	554	667	750	---	---	---	
G399TA[2] (LC)	555	704	801	---	---	---	
F5790G[1] (HC)	604	738	842	---	---	---	

NOTES
1. Ratings are manufacturer's recommendation for continuous compressor duty at 100°F and 2000' elevation with 1000 btu/cf fuel gas.
2. Deduct 3% of continuous shaft BHP if driving serial cooler from front and of engine
3. [1]Waukesha [2]Caterpillar [3]White [4]Superior [5]Dorman [6]Ingeraoll-Rand [7]Minnespolis-Moline [8]Lister
4. ( ) Verity speed and BHP with manufacturer.

Fig. 10.7 Natural gas engine ratings.

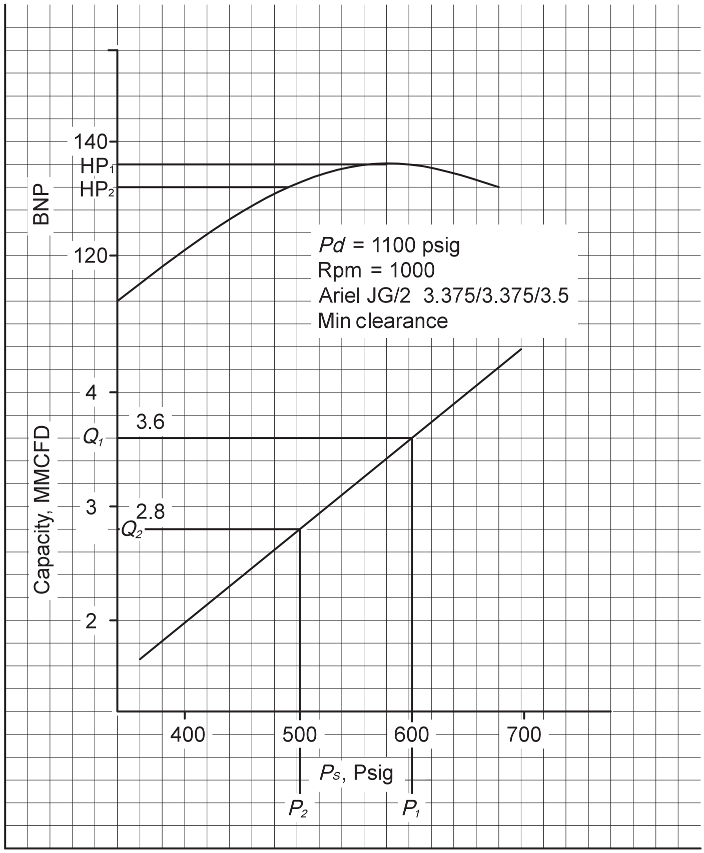


Fig. 10.8 Performance map for Example 10.3.

Determine:

What will be the effect on the compressor?

Solution:

(1) Plot the initial conditions of P1 and Q?

(2) Decrease the capacity to the final condition, Q2, and plot P2 by drawing a horizontal line from point Q2 to the curve.

(3) At the intersection read P2. Suction pressure will drop to 500 psig. In addition, the horsepower required will decrease from 138 hp. to 133 hp. An increase in vacuum should also be realized on the engine panel.

Example 10.4. Effect on Compressor Discharge

Given:

Data in Example 10.3.

Determine:

How will the discharge temperature be affected in Example 10.3?

Solution:

(1) The compression ratio will change since the discharge pressure is constant and the suction pressure decreases

(a) Initial compression ratio:

$$R_1 = \frac{P_d}{P_{s1}} = \frac{1115}{615} = 1.81$$

(b) Final compression ratio:

$$R_2 = \frac{P_d}{P_{s2}} = \frac{1115}{515} = 2.17$$

(2) Determine the change in gas discharge temperature assuming $k = 1.26$ and $T_s = 100°F$

(a) Initial gas discharge temperature:

$$Td_1 = T_{s1}\left(R_1^{\frac{k-1}{k}}\right) = 560(1.81)^{\frac{1.26-1}{1.26}} = 600^\circ \text{ or } 173^\circ F$$

(b) Final gas discharge temperature:

$$Td_2 = T_{s2}\left(R_2^{\frac{k-1}{k}}\right) = 560(2.17)^{\frac{1.26-1}{1.26}} = 657^\circ R \text{ or } 197^\circ F$$

∴Gas discharge temperature will increase by (197–173) 24°F.

Example 10.5. Effect on Compressor Load

Given:

Data in Example 10.3.

Determine:

What is the rod load in Example 10.3? Assume piston rod diameter is 1–1/8″ (1.125).

Solution:

(1) Calculate the piston area

$$A_p = \frac{\pi D^2}{4}$$
$$= \frac{\pi(3.375)^2}{4} = 8.95\ \text{in}^2$$

(2) Calculate the piston rod area

$$A_r = \frac{\pi d^2}{4}$$
$$= \frac{\pi(1.125)^2}{4} = 0.99\ \text{in}^2$$

(3) Calculate the rod load in compression since it will be the larger on a double-acting cylinder

(a) Initial conditions:

$$R_{\text{LC}} = P_d\left(A_p\right) - P_{s1}\left(A_p - A_r\right)$$
$$= (1100)(8.95) - (600)(8.95 - 0.99)$$
$$= 5068\ \text{Ibs}$$

(b) Final conditions:

$$R_{\text{LC}} = P_d\left(A_p\right) - P_{s2}\left(A_p - A_r\right)$$
$$= 1100(8.95) - 500(8.95 - 0.99)$$
$$= 5865\ \text{Ibs}$$

(4) The rod load increased (5865–5069) by 769 Ibs; however, it is still well under the rated rod load of 10,000 Ibs.

Example 10.6. Effect of Adding Clearance

Given:

Fig. 10.9

Determine:

Determine what is the effect of adding clearance as shown in Fig. 10.9, when operating at 600 psig.

Solution:

(1) Refer to Fig. 10.9. The capacity of the compressor will drop approximately 0.8 MMCFD

(2) The suction pressure will remain at 600 PSIG

(3) Note that adding clearance to a single-stage compressor, while maintaining constant suction pressure, will not effect

- Gas discharge temperature
- Rod load

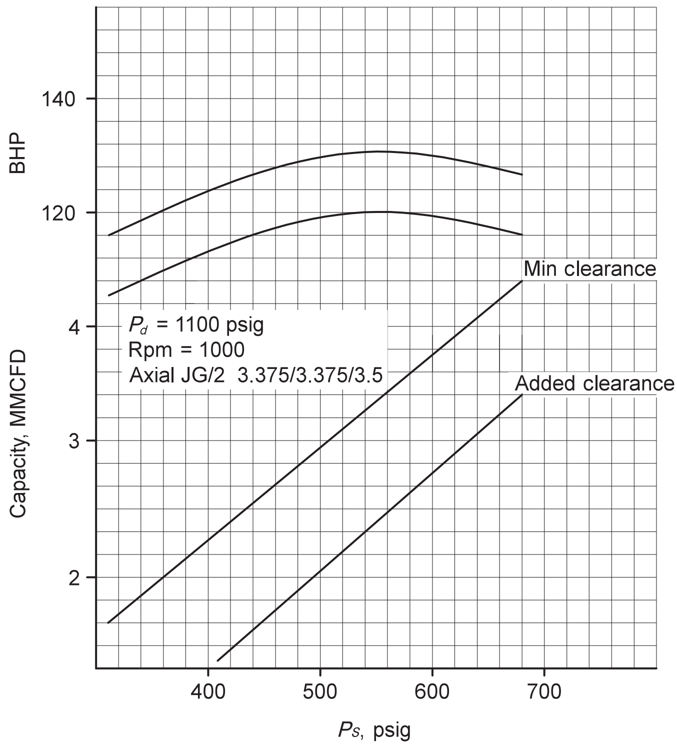


Fig. 10.9 Performance map for Example 10.6.

(4) Horsepower will decrease proportionally to capacity and one should see a slight increase in vacuum on the engine panel

Example 10.7. Two-Stage Performance Map
Given:

Attached two-stage performance curve (refer to Fig. 10.10). Note that the performance curve resembles a performance curve for a single-stage compressor.
Determine:

If the suction pressure drops 10 PSIG how much does capacity change? How much does horsepower change?
Solution:

(a) Capacity changes 0.45 MMCFD for each 10-PSIG pressure drop
(b) Horsepower changes 30 hp. for each 10-PSIG pressure drop

Example 10.8. Determining the Range of a Compressor
Given:

Compressor performance curve in Example 10.7 (Fig. 10.10). Compressor CP-650A is limited to 25,000 rod load.

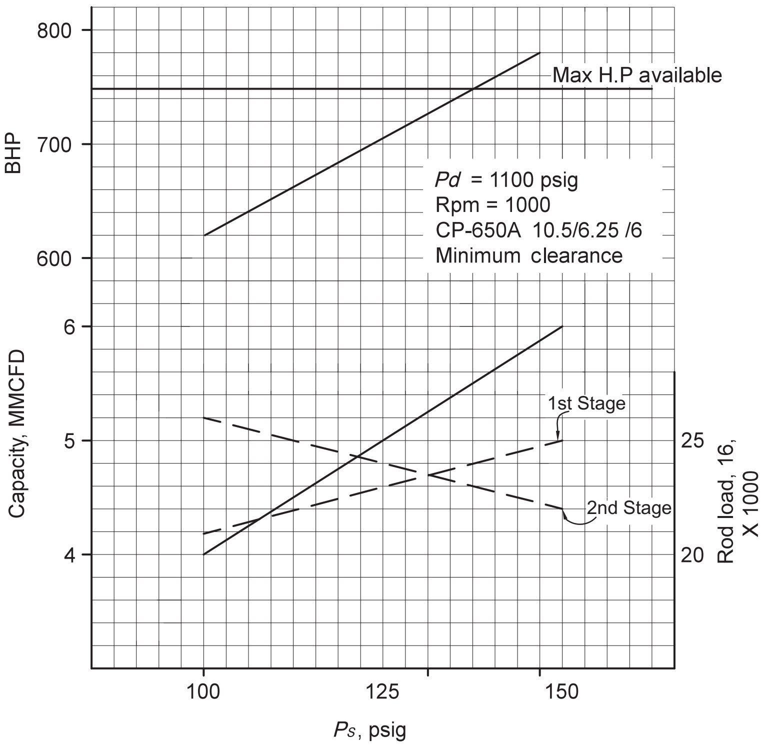


Fig. 10.10 Performance map for Example 10.7.

Determine:

(a) What is the total range of the compressor?
(b) What is the total range if a margin of 1000 is applied which limits the rod load to 24,000 Ibs?

Solution:

(a) At 25000 Ibs the range is 102 PSIG to 147 PSIG
(b) At 24,000 Ibs the range is from 118 PSIG to 138 PSIG

10.3 Adding clearance to multistage compressors

The reasons for adding clearance are as follows:

- Reduce control capacity.
- Reduce gas discharge temperature.
- Reduce rod loads.
- Optimize horsepower.

The procedure for adding clearance:

Step 1. Determine which cylinder clearance needs to be added to from the operating range by examining:

- Compression ratios
- Gas discharge temperature
- Rod loads

Step 2. Add clearance in small increments while checking all parameters after each addition. Try to equalize compression ratios.

Table 10.1 presents the initial conditions of a two-stage compressor. Notice the difference in compressor ratios of the first stage 2.979) and the ratio of the second stage (3.339). Since the second-stage ratio is higher than the first stage, clearance is added to that stage as additional clearance will reduce ratio.

Table 10.2 presents the effects of adding 5% clearance to the second stage. Note the ratios are nearly equal with the first stage increasing (3.136) and the second stage decreasing (3.168). Also note discharge temperature of the second stage decreases from 297°F to 289°F, rod load in tension of the second stage decreases from 20,473 to 19,920, and the volumetric efficiency decreases from 67% to 61%.

Table 10.3 presents the effects of adding more clearance (8%) to the second stage. The compressor ratios become unequal—first-stage ratio increases (3.228) while second-stage ratio decreases (3.077). Therefore too much clearance was added.

Table 10.1 Computer simulation—initial conditions

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	15.2
T_s (°F)	100	130
P_s (psig)	100	319
T_d (°F)	242	297
P_d (psig)	327	1100
R (Comp ratio)	2.97956	3.33987
VE (%)	65.3245	67.4403
Q_g (MMCFD)	4.07887	4.07729
BHP	286.841	336.137
RL_C	20,036.40	25,026.80
RL_T	18,601.30	20,472.80
BHP (total)	–	622.978

Table 10.2 Computer simulation—effects of adding 5% clearance to the second stage

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	20.2
T_s (°F)	100	130
P_s (psig)	100	337
T_d (°F)	249	289*
P_d (psig)	345	1100
R (Comp ratio)	3.13648	3.16897*
VE (%)	63.1335	61.7225
Q_g (MMCFD)	3.94206	3.9415
BHP	291.398	309.57
RL_C	21,596.20	24,530.70
RL_T	20,104.50	19,920.2*
BHP (total)	–	600.968

Table 10.3 Computer simulation—effects of adding 8% clearance to the second stage

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	23.2
T_s (°F)	100	130
P_s (psig)	100	348
T_d (°F)	253	284
P_d (psig)	356	1100
R (Comp ratio)	3.22813	3.07702
VE (%)	61.8591	58.6543
Q_g (MMCFD)	3.86248	3.86247
BHP	293.35	294.067
RL_C	22,507.10	24,241.10
RL_T	20,982.40	19,597.50
BHP (total)	–	588.317

Table 10.4 Computer simulation—effects of increasing suction pressure

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_g	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	15.2
T_s (°F)	100	130
P_s (psig)	125	373
T_d (°F)	234	274
P_d (psig)	381	1100
R (Comp ratio)	2.83437	2.87323
VE (%)	67.2064	72.5016
Q_g (MMCFD)	5.12932	5.1291
BHP	342.964	366.075
RL_C	22,645.90	23,532.90
RL_T	20,961.80	18,808.50
BHP (total)	–	709.039

Table 10.4 presents the effects of increasing the suction pressure from 100 psig to 125 psig. Note the ratios of both stages decrease and are nearly equal (first stage =2.834 and second stage = 2.873). Flow rate increases from 4.0 MM to 5.1 MM, total BHP increases from 623 to 709, discharge temperature decreases from 297°F to 274°F, volumetric efficiency increases from 67% to 72%, and rod load decreases from 20,473 to 18,808.

Table 10.5 presents the effects of adding 5% clearance to the second stage with the suction pressure remaining at 125 psig. Adding the 5% clearance makes the ratios unequal, reduces volumetric efficiency from 72% to 67%, reduces flow rate from 5.1 MM to 4.9 MM, and reduces the rod load from 18, 808 to 18,294. Adding clearance to the second stage does not provide any benefit.

Table 10.6 presents the effects of increasing suction pressure to 150 psig. Since the compressor ratio of the first stage is higher (2.728) than the second stage (2.525), clearance is added to the second stage.

Table 10.7 presents the effects of adding 5% clearance to the first stage. Note the compressor ratios are nearly equal, volumetric efficiency is reduced, flow rate reduced from 6.1 MM to 5.8 MM, and rod loads increase.

Table 10.5 Effects of adding 5% clearance to the second stage with suction pressure remaining at 125 psig

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	20.2
T_s (°F)	100	130
P_s (psig)	125	390
T_d (°F)	240	267
P_d (psig)	398	1100
R (Comp ratio)	2.95423	2.75435
VE (%)	65.506	67.6067
Q_g (MMCFD)	4.99954	4.99953
BHP	384.545	341.585
RL_C	24,097.00	23,071.40
RL_T	22,360.20	18,294.40
BHP (total)	–	690.13

Table 10.6 Computer simulation—effects of increasing suction pressure to 150 psig

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	20.3	15.2
T_s (°F)	100	130
P_s (psig)	150	427
T_d (°F)	229	254
P_d (psig)	435	1100
R (Comp ratio)	2.72891	2.52508
VE (%)	68.5725	76.4196
Q_g (MMCFD)	6.19233	6.19235
BHP	397.78	384.578
RL_C	25,193.30	22,058.80
RL_T	23,262.40	17,166.20
BHP (total)	–	782.358

Table 10.7 Effects of adding 5% clearance to first stage with suction pressure at 150 psig

Base conditions		
No. of stages	2	
P_b (psia)	14.7	
T_b (°F)	60	
k	1.26	
S_q	0.65	
Parameter	**First stage**	**Second stage**
PD (CFM)	590.413	202.145
CL Vol (%)	25.3	15.2
T_s (°F)	100	130
P_s (psig)	150	408
T_d (°F)	223	261
P_d (psig)	416	1100
R (Comp ratio)	2.61591	2.63622
VE (%)	64.4459	75.1543
Q_g (MMCFD)	5.81969	5.81969
BHP	357.036	379.319
RL_C	22,580.60	22,571.70
RL_T	21,708.20	17,737.60
BHP (total)	–	736.355

10.4 Effects of speed

Speed is proportional to

- Capacity
- Horsepower

Changing the speed of a compressor effects capacity control only. Speed does not effect:

- Compression ratio
- Gas discharge temperature
- Rod loads

Example 10.9. Effects of Speed Reduction

Given:

Data in Example 10.8.

Determine:

Develop the compressor performance at a speed of 800 rpm.

Solution:

(1) Calculate piston displacement at a speed of 800 rpm

$$PD = \left(\frac{\left(2d_r^2 - d_r^2\right)}{2200}\right) Sn \begin{pmatrix} \text{Number of} \\ \text{Cylinder} \end{pmatrix}$$

$$= \left(\frac{\left((2)(3.375)^2 - (1.125)^2\right)(3.5)(800)}{2200}\right)(2)$$

$$= 54.78 \text{ ft}^3/\text{min}$$

(2) Calculate capacity

(a) Determine ACFM

$$\begin{aligned} \text{ACFM} &= (PD)E_v \\ &= (54.78)(0.86) \\ &= 47.11 \text{ ft}^3/\text{min} \end{aligned}$$

(b) Determine capacity in MMSCFD

$$\begin{aligned} Q_{350} &= (0.051)\left(\frac{(\text{ACFM})(P_s)}{T_s Z_s}\right) \\ &= (0.051)\left(\frac{(54.78)(365)}{(540)(0.96)}\right) \\ &= 1.3 \text{ MMSCFD} \end{aligned}$$

$$\begin{aligned} Q_{525} &= (0.051)\left(\frac{(54.78)(540)}{(540)(0.93)}\right) \\ &= 2.33 \text{ MMSCFD} \end{aligned}$$

$$\begin{aligned} Q_{700} &= (0.051)\left(\frac{(54.78)(715)}{(540)(0.91)}\right) \\ &= 3.39 \text{ MMSCFD} \end{aligned}$$

(3) Calculate brake horsepower

$$\text{BHP} = 0.0857\left[Z_{AV}^{\frac{1}{k}}\right][Z_s]^{\frac{k-1}{k}}\left[\frac{Q_g T_s}{E_m E_a}\right] \times \left[\frac{k n_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{\ln_p}} - 1\right]$$

Substitute

$$\text{BHP}_{350} = 87$$

$$\text{BHP}_{525} = 106$$

$$\text{BHP}_{700} = 102$$

(4) Plot the compression's performance

See Fig. 10.11.

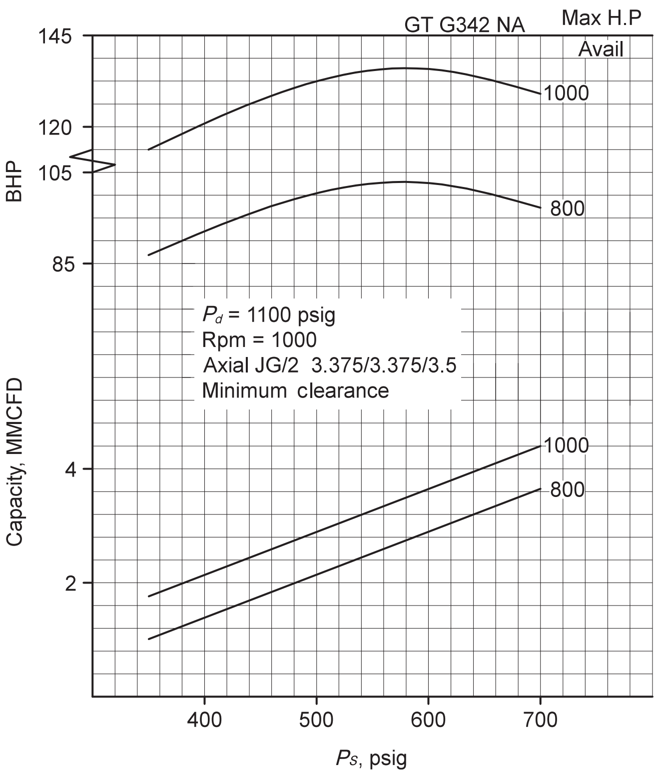


Fig. 10.11 Performance map for Example 10.9 showing the effects of reducing speed from 1000 rpm to 800 rpm.

10.5 Multistage compressors

Developing a performance map for a multistage compressor is more complex than that of a single-stage compressor. If properly handled in a logical, sequential order, the performance map can be developed. Performance map takes on the same general appearance as that of a single-stage compressor.

Example 10.10. Developing a Performance Map for a Multistage Compressor
Given:

Number of stages	**2**
Cylinders	10.5/6.25 DA
Stroke	6
RPM	1000
Piston rod diameter	2″
P_s	100–150 PSIG
P_d	1100PSIG
T_s	120°F
T_{S2}	130°F
S_g	0.65
K	1.26
Clearance	Minimum

Determine:

Develop a performance curve on a Chicago Pneumatic 6FE650A with series 75 cylinders.

Solution:

(1) Calculate the piston displacement with the last stage first

$$\begin{aligned}
\mathrm{PD} &= \left(\frac{\left(2d_c^2 - d_r^2\right)}{2200}\right)\begin{pmatrix}\text{Number of}\\ \text{Cylinder}\end{pmatrix}\\
&= \left(\frac{\left(2(6.25)^2 - (2)^2\right)(6)(1000)}{2200}\right)(1)\\
&= 202.4\ \mathrm{ft^3/min}
\end{aligned}$$

(2) Calculate the "ideal" compression ratio. Assume a midrange suction pressure, $P_s = 125$ psig

$$\begin{aligned}
R &= \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}}\\
&= \left(\frac{1115}{140}\right)^{\frac{1}{2}}\\
&= 2.82
\end{aligned}$$

(3) Calculate the gas discharge temperature, T_d

$$\begin{aligned}
T_{d2} &= T_s R^{\frac{k-1}{kn_p}}\\
&= (590)(2.82)^{\frac{1.26-1}{(1.26)(1)}}\\
&= 730^\circ R \ or\ 270^\circ F
\end{aligned}$$

(4) Calculate the volumetric efficiency

(a) Determine suction pressure of second stage

$$\begin{aligned}
P_{s2} &= P_{s1}R\\
&= (140)(2.82)\\
&= 394\ \text{psia}
\end{aligned}$$

(b) Determine compressibility factors at suction and discharge pressure (refer to Fig. 10.5)

$$Z_s @ 394\ \text{psia and}\ 130^\circ\mathrm{F} = 0.95$$

$$Z_d @ 1115\ \text{psia and}\ 270^\circ\mathrm{F} = 0.96$$

(c) Determine % average clearance

From Vendor Data Sheet (Fig. 10.12), read average clearance for 6.25-in. cylinder = 15.17%

(d) Calculate volumetric efficiency (Assume High-Speed Separable Reciprocating Compressors)

$$E_v = 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right] - L$$

$$= 96 - (2.28) - 15.17\left[\left(\frac{.95}{.96}\right)(2.82)^{\frac{1}{1.26}} - 1\right] - 1$$

$$= 71.3\%$$

Note: For low-speed integral compressors the slippage of gas past the piston rings is 0%, for high speed separable 1%, propane service 4%, and nonlubricated 5%.

(5) Calculate the capacity

(a) Determine ACFM

$$\text{ACFM} = (\text{PD})E_v$$

$$= (202.4)(.713)$$

$$= 144.31\ \text{ft}^3/\text{min}$$

(b) Determine capacity in MMSCFD

$$Q_g = (0.051)\left(\frac{(\text{ACFM})(P_s)}{T_s Z_s}\right)$$

$$= (0.051)\left(\frac{(144.31)(394)}{(590)(0.95)}\right)$$

$$= 5.17\ \text{MMSCFD}$$

(6) Calculate the piston displacement of the first stage

$$\text{PD} = \left(\frac{(2_c^2 - d_r^2)SN}{2200}\right)\left(\begin{matrix}\text{Number of}\\ \text{Cylinder}\end{matrix}\right)$$

$$= \left(\frac{\left((2)(10.5)^2 - (2)^2\right)(6)(1000)}{2200}\right)(1)$$

$$= 591\ \text{ft}^3/\text{min}$$

Series 75 cylinders FE 650 6.0" stroke 1000 RPM													
		Valves/end				Openings		Clearance %			MWP		V.V.VL. Pocket
Bore	Basic pattern	Size	Quantity in. & dis.	Area inlet	Area disch.	Inlet	Disch.	NE	CE	AVG.	API (PSI)	PD CFM	total vol. (vol/in.)
3.50	1-027634	65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	30.26	47.30	37.12	3600	55.91	89.1 in.^{3}
3.75		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	25.70	37.80	30.75	3600	65.8	(7.06 in.^{3}/in)
4.00		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	22.00	31.00	25.86	3600	76.36	
4.25		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	17.80	25.70	21.26	3600	87.61	
4.50	1-026577*	65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	26.31	30.19	28.04	1500*	99.55	187.7 in.^{3}
4.75		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	22.35	24.90	23.50	1500	112.16	(11.04 in.^{3}/in)
5.00		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	19.04	20.65	19.78	1500	125.45	
5.25		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	16.24	17.23	16.70	1500	139.43	
5.50	1-025919**	65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	22.76	24.62	23.62	1500**	154.09	187.7 in.^{3}
5.75		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	19.75	21.02	20.34	1500	169.43	(11.04 in.^{3}/in)
6.00		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	17.16	17.99	17.55	1500	185.45	
6.25		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	14.93	15.43	15.17	1500	202.16	
7.00	1-026140	65CED	1 + 1	5.21	5.21	4 - 900#	4 - 900#	26.16	23.46	24.87	1500	256.36	270 in.^{3}
7.50		65CED	1 + 1	5.21	5.21	4 - 900#	4 - 900#	21.30	18.60	20.00	1500	295.91	(15.88 in.^{3}/in)
8.50	1-025965	65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	27.09	25.56	26.35	1250	383.18	270 in.^{3}
9.00		65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	22.78	21.20	22.01	1250	430.91	(15.88 in.^{3}/in)
9.50		65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	19.21	17.63	10.44	1250	481.36	
10.00	1-026363	65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	24.76	23.59	24.18	1000	534.55	654.2 in.^{3}
10.50		65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	20.87	19.65	20.27	1000	590.45	(38.48 in.^{3}/in)
11.00		65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	17.59	16.37	16.99	1000	649.09	

*D.I. - 1-029227, 2000 MWP. **D.I. - 1-029226, 2000 MWP.

Remarks:

Fig. 10.12 Vendor data sheet.

(7) Calculate interstage pressure loss

$$\begin{aligned}\Delta P &= (0.02)(P_{s2})\\ &= (0.02)(394\ \text{psia})\\ &= 7.88\ \text{psia}\end{aligned}$$

(8) Calculate first-stage discharge pressure (P_{d1})

$$\begin{aligned}P_{d1} &= P_{s2} + \Delta P\\ &= 394 + 7.88\\ &= 401.88\ \text{psia}\end{aligned}$$

(9) Calculate compression ratio of first stage

$$\begin{aligned}R_1 &= \frac{401.88}{(125+14.7)}\\ &= 2.88\end{aligned}$$

(10) Calculate gas discharge temperature of first stage

$$\begin{aligned}T_{d1} &= T_{s1}\left(\frac{P_{d1}}{P_{s1}}\right)^{\frac{k-1}{kn_p}}\\ &= 580\left(\frac{401.88}{139.7}\right)^{\frac{1.26-1}{(1.26)(1)}}\\ &= 721°\text{R or } 261°\text{F}\end{aligned}$$

(11) Calculate the volumetric efficiency

(a) Determine compressibility factors at suction and discharge pressure (refer to Fig. 10.5)

$$Z_{s1}\ @\ 125\ \text{psig and } 120°\text{F} = 0.975$$

$$Z_{d1}\ @\ 386\ \text{psig and } 261°\text{F} = 0.97$$

(b) Determine C, % average clearance

From Fig. 10.12, read $C = 20.27\%$

(c) Calculate volumetric efficiency

$$\begin{aligned}E_v &= 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right]\\ &= 96 - (2.87) - (20.27)\left[\left(\frac{0.975}{0.97}\right)(2.87)^{\frac{1}{1.26}} - 1\right]\\ &= 63.47\%\end{aligned}$$

(12) Calculate the capacity
(a) Determine ACFM

$$\begin{aligned}\text{ACFM} &= (\text{PD})E_v\\ &= (591)(0.6347)\\ &= 375.11\ \text{ft}^3/\text{min}\end{aligned}$$

(b) Determine capacity in MMSCFD

$$\begin{aligned}Q_g &= (0.051)\left(\frac{(\text{ACFM})(P_s)}{T_s Z_s}\right)\\ &= (0.051)\left(\frac{(375.11)(140)}{(580)(0.975)}\right)\\ &= 4.72\ \text{MMSCFD}\end{aligned}$$

Discussion:

- 10.5″ cylinder's capacity is less than the 6.5″ cylinder at these conditions.
- Q_1 must equal Q_2.
- An iteration process must take place so that $Q_1 = Q_2$.
- Since Q_2 is greater than Q_1, the second-stage cylinder is large with respect to the first-stage cylinder at $P_s = 125$ psig.
- In order to achieve $Q_1 = Q_2$, P_{S2} must be adjusted downward and Step 3 through 11 must be repeated.

(13) Iteration for $Q_1 = Q_2$
(a) Trial #1: $P_{S2} = 380$ psia

1. $$\begin{aligned}T_{d2} &= 590\left(\frac{1115}{380}\right)^{\frac{1.26-1}{1.26(1)}}\\ &= 736°\text{R or } 275°\text{F}\end{aligned}$$

2. $$\begin{aligned}E_v &= 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right]\\ &= 96 - 2.93 - 15.17\left[\left(\frac{0.94}{0.96}\right)(2.93)^{\frac{1}{1.26}} - 1\right]\\ &= 70.45\%\end{aligned}$$

3. $$\begin{aligned}Q_2 &= (0.051)\left(\frac{(PD)(E_v)P_s}{T_s Z_s}\right)\\ &= (0.051)\left(\frac{(202.4)(.7045)(380)}{(590)(0.94)}\right)\\ &= 4.98\,MMSCFD\end{aligned}$$

4. If $\Delta P = 7.88$ psia, the $P_{d1} = 387.88$

5. $T_{d1} = 580\left(\dfrac{387.88}{140}\right)^{\frac{1.26-1}{1.26}}$

$= 715^{\circ}\text{R or } 255^{\circ}\text{F}$

6. $E_v = 96 - 2.76 - 20.27\left[\dfrac{(0.975)}{0.98}(2.76)^{\overline{11.26}} - 1\right]$

$= 65.61\%$

7. $Q_1 = (0.051)\left(\dfrac{(591)(0.6561)(140)}{(0.975)(580)}\right)$

$= 4.88$ MMSCFD

Since $Q_2 > Q_1$ must iterate again

(b) Trial #1: $P_{S2} = 375$ psia

1. $T_{d2} = 590\left(\dfrac{1115}{375}\right)^{\frac{1.26-1}{1.26(1)}}$

$= 738^{\circ}\text{R or } 278^{\circ}\text{F}$

2. $E_v = 96 - 2.97 - 15.17\left[\left(\dfrac{(0.94)}{(0.96)}\right)(2.97)^{\overline{11.26}} - 1\right]$

$= 69.99\%$

3. $Q_2 = (0.051)\left(\dfrac{(202.4)(0.6997)(375)}{(590)(0.94)}\right)$

$= 4.89$ MMSCFD

4. If $\Delta P = 7.88$ psia, the $P_{d1} = 382.88$ psia

5. $T_{d1} = 580\left(\dfrac{382.88}{140}\right)^{\frac{1.26-1}{1.26}}$

$= 713^{\circ}\text{R or } 253^{\circ}\text{F}$

6. $E_v = 96 - 2.73 - 20.27\left[\dfrac{(0.975)}{(0.98)}(2.73)^{\frac{1}{1.26}} - 1\right]$

$= 66.06\%$

7. $Q_1 = (0.051)\left(\dfrac{(591)(0.6606)(140)}{(0.975)(580)}\right)$

$= 4.88$ MMSCFD

$Q_1 \approx Q_2 = 4.9$ MMSCFD

(14) Calculate the break horsepower for each cylinder

1. $$\text{BHP}_1 = 0.0857\left[Z_{av}^{\frac{1}{k}}\right][Z_s]^{\frac{1-k}{kn_p}}\left[\frac{Q_g T_s}{E_m E_a}\right] \times \left[\frac{kn_p}{k-1}\right]\left[\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{kn_p}} - 1\right]$$

$$= (0.0857)[0.975]^{\frac{1}{1.26}}[0.94]^{\frac{(1.26-1)}{1.26}}\left[\frac{(4.9)(580)}{(.93)(.85)}\right]$$

$$\times \left[\frac{(1.26)(1)}{(1.26-1)}\right]\left[\left(\frac{382.9}{140}\right)^{\frac{(1.26-1)}{(1.26)(1)}} - 1\right] = 299$$

2. $$\text{BHP}_1 = (0.0857)(0.95)^{\frac{1}{1.26}}(0.975)^{\frac{1.26-1}{1.26}}\left(\frac{(4.9)(590)}{(0.93)(0.85)}\right)$$
$$\times\left[\left[\frac{(1.26)(1)}{(1.26-1)}\right]\left(\frac{1115}{375}\right)^{\frac{(1.26-1)}{(1.26)(1)}}-1\right] = 325$$

3. $$\begin{aligned}\text{Total BHP} &= (\text{BHP})_1 + (\text{BHP})_2\\ &= 299 + 325\\ &= 624\end{aligned}$$

(15) Repeat Steps 2 through 14 for $P_s = 100$ psig

(a) For $P_s = 100$ psig

$$R = \left(\frac{1115}{115}\right)^{\frac{1}{2}} = 3.12$$
$T_{d2} = 746°\text{R}$ or $286°\text{F}$
$E_{v2} = 67.5\%$
$Q_2 = 4.44$ MMSCFD
$\Delta P = 6$ psig
$P_{d1} = 364$ psia
$T_{d1} = 735°\text{R}$ or $276°\text{F}$
$E_{v1} = 59.59\%$
$Q_1 = 3.6$ MMSCFD

$Q_1 > Q_2$ $\therefore$ Decrease P_{S2} to 340 psia and repeat the calculations

(b) Trial #1. Assume $P_{S2} = 340$ psia
$$R = \left(\frac{1115}{340}\right) = 3.28$$
$T_{d2} = 754°\text{R}$ or $294°\text{F}$
$E_{v2} = 65.67\%$
$Q_2 = 4.11$ MMSCFD
$\Delta P = 6.4$ psig, $P_{d1} = 346$ psia
$$R = \left(\frac{346}{115}\right) = 3.02$$
$T_{d1} = 728°\text{R}$ or $268°\text{F}$
$E_{v1} = 61.5\%$
$Q_1 = 3.74$ MMSCFD

$Q_1 > Q_2$ $\therefore$ Reduce P_{S2} to 340 psia

(c) Trial #2. Assume $P_{s2} = 327$ psia
$$R = \left(\frac{1115}{327}\right) = 3.41$$
$T_{d2} = 760°\text{R}$ or $300°\text{F}$
$E_{v2} = 64.19\%$
$Q_2 = 3.86$ MMSCFD
$\Delta P = 6.4$ psig, $P_{d1} = 333$ psia
$$R = \left(\frac{333}{115}\right) = 2.90$$

$T_{d1} = 722°R$ or $262°F$
$E_{v1} = 63.28\%$
$Q_1 = 3.84$ MMSCFD
$Q_1 = Q_2$ thus $Q = 3.85$ MMSCFD

(d) Calculate BHP

$$\text{BHP}_1 = 246$$

$$\text{BHP}_2 = 282$$

$$\begin{aligned}\text{Total BHP} &= 246 + 282\\ &= 528\end{aligned}$$

(16) Repeat Steps 2 through 14

(a) For $P_s = 150$ psia

$R = \left(\frac{1115}{165}\right)^{\frac{1}{2}} = 2.60$
$T_{d2} = 718°R$ or $258°F$
$E_{v2} = 74\%$
$Q_2 = 5.89$ MMSCFD
$\Delta P = 6.8$ psig; $P_{d1} = 435$ psia
$R = \left(\frac{435}{165}\right) = 2.64$
$T_{d1} = 718°R$ or $258°F$
$E_v = 67\%$
$Q_1 = 5.87$ MMSCFD
$\therefore Q_1 = Q_2 = 5.88$ MMSCFD

(b) Calculate BHP

$$\text{BHP}_1 = 347$$

$$\text{BHP}_2 = 344$$

$$\begin{aligned}\text{Total BHP} &= 347 + 344\\ &= 691\end{aligned}$$

(17) Calculate rod loads in compression (RL_C) for each suction pressure on the first stage

(a) Calculate area of piston and area of rod

$$\begin{aligned}A_p &= \frac{\pi d_c^2}{4}\\ &= \frac{\pi(10.5)^2}{4}\\ &= 86.6 \text{ in}^2\end{aligned}$$

$$\begin{aligned}A_r &= \frac{(\pi d_r)^2}{4}\\ &= \frac{\pi(2)^2}{4}\\ &= 3.14 \text{ in}^2\end{aligned}$$

(b) Calculate rod load for each suction pressure

$$\mathrm{RL_c} = P_d A_p - P_s\left(A_p - A_r\right)$$

$$\text{For}\,\mathrm{RL_{c100}} = (318)(86.6) - 100(86.6 - 3.14)$$
$$= 19{,}193\ \text{lbs}$$

$$\text{For}\,\mathrm{RL_{c125}} = 367(86.6) - 125(86.6 - 3.14)$$
$$= 21{,}350\ \text{lbs}$$

$$\text{For}\,\mathrm{RL_{c150}} = 420(86.6) - 150(86.6 - 3.14)$$
$$= 23{,}853\ \text{lbs}$$

(18) Calculate the rod loads on the second stage at each pressure point
(a) Calculate area of second-stage piston (A_p)

$$A_p = \frac{\pi d_c^2}{4}$$
$$= \frac{\pi (6.25)^2}{4}$$
$$= 30.7\ \text{in}^2$$

(b) Calculate the rod loads

$$\text{For}\,\mathrm{RL_{c100}} = 1100(30.7) - 312(30.7 - 3.14)$$
$$= 25{,}171\ \text{lbs}$$

$$\text{For}\,\mathrm{RL_{c125}} = 1100(30.7) - 360(30.7 - 3.14)$$
$$= 23{,}848\ \text{lbs}$$

$$\text{For}\,\mathrm{RL_{c150}} = 1100(30.7) - 413(30.7 - 3.14)$$
$$= 22{,}385\ \text{lbs}$$

(19) Plot the performance curve (refer to Fig. 10.13)

Example 10.11. Determining Clearance Required to Reduce Capacity

Given:

Data in Example 10.10.

Determine:

Determine the amount of clearance that needs to be added to the head end of the second-stage cylinder to reduce the capacity at $P_s = 125$ psig to $Q = 4.72$ MMSCFD.

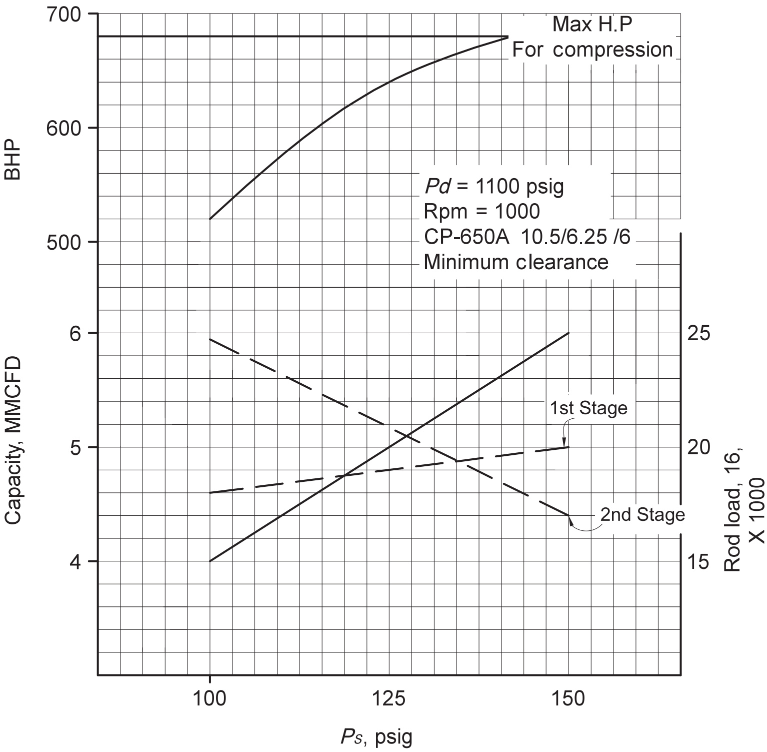


Fig. 10.13 Multistage performance map for Example 10.10.

Solution:

(1) Set $Q_2 = 4.72$ MMSCFD and solve for ACFM

$$Q_2 = (0.051)\left(\frac{(\mathrm{ACFM})P_s}{T_s Z_s}\right)$$

Rearranging

$$4.72 = (0.051)\left(\frac{(\mathrm{ACFM})(394)}{(590)(0.95)}\right)_{\mathrm{T}}$$

Solving for ACFM

$$\mathrm{ACFM} = 132.3\ \mathrm{ft}^3/\mathrm{min}$$

(2) Calculate volumetric efficiency (E_v)

$$\mathrm{ACFM} = (\mathrm{PD})E_v$$

Rearranging and solving for E_v

$$E_v = \frac{\text{ACFM}}{\text{PD}}$$
$$= \frac{132.2}{202.4}$$
$$= 65.22\%$$

(3) Calculate average cylinder clearance needed to satisfy the decreased volumetric efficiency (E_v)

$$E_v = 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right]$$

Rearranging and solving for C

$$65.22 = 96 - 2.82 - C\left[\left(\frac{0.95}{0.96}\right)2.82^{\frac{1}{1.26}} - 1\right]$$
$$= 20.11\%$$

(4) Determine clearance to be added to the second-stage cylinder

$$= 20.11 - 15.17$$

$$= 4.94\%(\text{Average Clearance})$$

(5) Determine the clearance to be added to the head end of the cylinder

$$\binom{\text{AVG}}{C} = \frac{\text{HE} + \text{CE}}{2}$$
$$20.11 = \frac{\text{HE} + 15.43}{2}$$
$$\text{HE} = 24.79\%$$

Normal clearance on the head end is 14.93%. Therefore the amount of clearance that needs to be added to the HE is:

$$= 24.79 = 14.93$$

$$= 9.86\%$$

Example 10.12. Designing a Compressor to Meet Gas Sales Line Pressure

Given:

A bulk separator is designed to handle 5000 BOPD. A well test yielded the following information:

$S_g = 0.65$
$k = 1.26$
$\text{API} = 36$
$\text{GOR} = 1000\text{SCF/BBL}$
$P_b = 14.7$ psia
$T_b = 520°\text{R}$
$T_{s1} = 120°\text{F}$
$T_{s2} = 130°\text{F}$
$P_o = 125$ psig
$Q_Q = 5$ MMSCFD

Vendor data sheet (Fig. 10.12) for CP FE 650A.

Determine:

Design the compressor to meet the gas sales line pressure of 1100 psig.

Solution:

(1) Determine the number of stages that are required to meet the given conditions

$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}} \leq 3.5$$
$$= \left(\frac{1114.7}{139.7}\right)^{\frac{1}{2}}$$
$$= 2.82$$

∴ Use two stages

(2) Calculate a "rough" horsepower by using the "quick look" formula. This allows us to select a frame in the horsepower range.

$$\text{BHP} = 22RNFQ_g$$
$$= (22)(2.82)(2)(1.08)(5)$$
$$= 670$$

(3) Calculate "rough" cylinder diameters. Begin with the last (second) stage.

(a) Determine second-stage suction pressure

$$P_{s2} = (P_{s1})(R)$$
$$= (139.7)(2.82)$$
$$= 394 \text{ psia}$$

(b) Convert MMSCFD to ACFM at 394 psia and130°F

$$\begin{aligned}\text{ACFM} &= 19.6\left(\frac{Z_s T_s Q_g}{P_s}\right)\\ &= 19.6\left(\frac{(0.95)(590)(5)}{394}\right)\\ &= 139.6\ \text{f}t^3/\text{min}\end{aligned}$$

(c) Calculate piston displacement (PD)

By defining

$$\text{ACFM} = E_v(\text{PD})$$

Assuming $E_v = 60\%$ for the first cut. Rearrange and solve for PD

$$\begin{aligned}\text{PD} &= \frac{\text{ACFM}}{E_v}\\ &= \frac{139.6}{0.6}\\ &= 232.7\ \text{ft}^3/\text{min}\end{aligned}$$

(d) Obtain data from Manufacturer's Data Sheet (Fig. 10.14)

Assume the compressor to have one (1) double-acting cylinder on the second stage and a CP FE650A frame. Refer to Fig. 10.14.

Speed = 1000 rpm
Stroke = 6 in.
Piston rod diameter = 2 in.

Note: Frame identification; FE type (balanced opposed); 6 stroke (inches); 50 frame size; A-model change indicated.

(e) Use the second-stage piston displacement equation to solve for d_c

$$\text{PD} = \left(\frac{(2d_c^2 - d_r^2)SN}{2200}\right)\left(\begin{array}{l}\text{Number of}\\ \text{Cylinders}\end{array}\right)$$

Rearranging, substitute and solve for d_c

$$\begin{aligned}d_c &= \left[\frac{\dfrac{[(PD)2200]}{SN(2)} + d_r^2}{2}\right]^{0.5}\\ &= \left[\frac{\left(\dfrac{(232.7)2200}{(6)(1000)(2)} + 2^2\right)}{2}\right]\\ &= 6.68\ \text{in}\end{aligned}$$

Series 75 cylinders FE 650 6.0" stroke 1000 RPM													
		Valves/end				Openings		Clearance %			MWP		V.V.VL. Pocket
Bore	Basic pattern	Size	Quantity in. & dis.	Area inlet	Area disch.	Inlet	Disch.	NE	CE	AVG.	API (PSI)	PD CFM	total vol. (vol/in.)
3.50	1-027634	65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	30.26	47.30	37.12	3600	55.91	89.1 in.^{3}
3.75		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	25.70	37.80	30.75	3600	65.8	(7.06 in.^{3}/in)
4.00		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	22.00	31.00	25.86	3600	76.36	
4.25		65CED	1 + 1	1.24	1.24	2 1/2 - 5000#	2 1/2 - 5000#	17.80	25.70	21.26	3600	87.61	
4.50	1-026577*	65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	26.31	30.19	28.04	1500*	99.55	187.7 in.^{3}
4.75		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	22.35	24.90	23.50	1500	112.16	(11.04 in.^{3}/in)
5.00		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	19.04	20.65	19.78	1500	125.45	
5.25		65CED	2 + 2	2.48	2.48	3 - 900#	3 - 900#	16.24	17.23	16.70	1500	139.43	
5.50	1-025919**	65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	22.76	24.62	23.62	1500**	154.09	187.7 in.^{3}
5.75		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	19.75	21.02	20.34	1500	169.43	(11.04 in.^{3}/in)
6.00		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	17.16	17.99	17.55	1500	185.45	
6.25		65CED	2 + 2	3.48	3.48	4 - 900#	4 - 900#	14.93	15.43	15.17	1500	202.16	
7.00	1-026140	65CED	1 + 1	5.21	5.21	4 - 900#	4 - 900#	26.16	23.46	24.87	1500	256.36	270 in.^{3}
7.50		65CED	1 + 1	5.21	5.21	4 - 900#	4 - 900#	21.30	18.60	20.00	1500	295.91	(15.88 in.^{3}/in)
8.50	1-025965	65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	27.09	25.56	26.35	1250	383.18	270 in.^{3}
9.00		65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	22.78	21.20	22.01	1250	430.91	(15.88 in.^{3}/in)
9.50		65CED	1 + 1	7.78	7.78	6 - 900#	6 - 900#	19.21	17.63	10.44	1250	481.36	
10.00	1-026363	65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	24.76	23.59	24.18	1000	534.55	654.2 in.^{3}
10.50		65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	20.87	19.65	20.27	1000	590.45	(38.48 in.^{3}/in)
11.00		65CED	2 + 2	10.42	10.42	6 - 900#	6 - 900#	17.59	16.37	16.99	1000	649.09	
*D.I. - 1-029227, 2000 MWP.				**D.I. - 1-029226, 2000 MWP.									
Remarks:													

Fig. 10.14 Vendor data sheet—Chicago Pacific (CP FE650A).

(f) Select a cylinder from Manufacturer Data Sheet (Fig. 10.14)

Choose a 6.25″ Series 75 cylinder

(4) Calculate the PD for the selected cylinder

$$\begin{aligned}\mathrm{PD} &= \left(\frac{(2_c^2 - d_r^2)SN}{2200}\right)\begin{pmatrix}\text{Number of}\\ \text{Cylinders}\end{pmatrix}\\ &= \left[\frac{\left(2(6.25)^2-(2)^2\right)(6)(1000)}{2200}\right](1)\\ &= 202.4\ \mathrm{ft}^3/\mathrm{min}\end{aligned}$$

(5) Calculate the gas discharge temperature

$$\begin{aligned}T_d &= T_s\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{kn_p}}\\ &= 590\left(\frac{1114.7}{139.7}\right)^{\frac{1.26-1}{(1.26)(1)}}\\ &= 730^\circ\mathrm{R}\ \text{or}\ 270^\circ\mathrm{F}\end{aligned}$$

(6) Calculate actual volumetric efficiency (E_v)
(a) Determine compressibility factors at suction and discharge conditions

$$Z_s @ 394\ \text{psia and} 130^\circ\mathrm{F} = 0.95$$

(b) Determine average normal clearance (C)

From Manufacturer Data Sheet (Fig. 10.14) a 6.25-in. cylinder has an average normal clearance of 15.17%

(c) Calculate volumetric efficiency (E_v)

$$\begin{aligned}E_v &= 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right]\\ &= 96 - 2.82 - 15.17\left[\frac{0.95}{0.95}(2.82)^{\frac{1}{1.26}} - 1\right]\\ &= 70.99\%\end{aligned}$$

(7) Calculate capacity with new cylinder
(a) Determine ACFM

$$\begin{aligned}\mathrm{ACFM} &= (\mathrm{PD})E_v\\ &= (202.4)(0.7098)\\ &= 143.68\ \mathrm{ft}^3/\mathrm{min}\end{aligned}$$

(b) Determine capacity in MMSCFD

$$
\begin{aligned}
Q_g &= (0.051)\left(\frac{(\mathrm{ACFM})P_s}{T_s Z_s}\right)\\
&= (0.051)\left(\frac{(143.68)(394)}{(590)(0.95)}\right)\\
&= 5.14\mathrm{MMSCFD}
\end{aligned}
$$

(8) Calculate the first-stage cylinder size
(a) Determine interstage pressure loss (assume 2%)

$$
\begin{aligned}
\Delta P &= (0.02)(P_d)\\
&= (0.02)(379.3)\\
&= 7.6\ \text{psig}
\end{aligned}
$$

(b) Determine first-stage discharge pressure

$$
\begin{aligned}
P_{d1} &= (379.4) + (7.6)\\
&= 387\ \text{psig or}\ 401.7\ \text{psia}
\end{aligned}
$$

(c) Convert MMSCFD to ACFM at 139.7 psia and 120°F

$$
\begin{aligned}
\mathrm{ACFM} &= 19.6\left(\frac{Z_s T_s Q_g}{P_s}\right)\\
&= 19.6\left(\frac{(0.98)(580)(5.14)}{139.7}\right)\\
&= 410.56\ \mathrm{ft}_3/\mathrm{min}
\end{aligned}
$$

(d) Calculate piston displacement (PD)

By definition ACFM $= E_v$ (PD). Assume $E_v = 70\%$ for first cut. Rearrange and solve for PD

$$
\begin{aligned}
\mathrm{PD} &= \frac{\mathrm{ACFM}}{E_v}\\
&= \frac{410.56}{E_v}\\
&= 586.5\ \mathrm{ft}^3/\mathrm{min}
\end{aligned}
$$

(e) Determine diameter of the piston (d_c)

Assume compressor has one double-acting cylinder as the first stage.
Solving for d_c

$$d_c = \left[\frac{\left(\frac{(\mathrm{PD})2200}{\mathrm{SN}(2)} + d_r^2\right)}{2}\right]^{0.5}$$

$$= \left[\frac{\left(\frac{(586.5)(2200)}{(6)(1000)(2)} + (2)^2\right)}{2}\right]^{0.5}$$

$$= 10.46 \text{ inches}$$

(f) Refer to Manufacturer's Data Sheet (Fig. 10.14)

Select a 10.5-in. cylinder with an average minimum clearance = 20.27%

(9) Calculate the piston displacement for the selected cylinder

$$\mathrm{PD} = \left(\frac{\left(2d_c^2 - d_r^c\right)\mathrm{SN}}{2200}\right)\begin{pmatrix}\text{Number of}\\ \text{Cylinders}\end{pmatrix}$$

$$= \left[\frac{\left((2)(10.5)^2 - (2)^2\right)(6)(1000)}{2200}\right](1)$$

$$= 591 \ \mathrm{ft}^3/\mathrm{min}$$

(10) Calculate the gas discharge temperature

$$T_d = T_s\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}}$$

$$= 580\left(\frac{401.7}{139.7}\right)^{\frac{1.26-1}{1.26}}$$

$$= 721^{\circ}\mathrm{R} \text{ or } 61^{\circ}\mathrm{F}$$

(11) Calculate volumetric efficiency (E_v) for selected cylinder

(a) Determine compressibility factors at suction and discharge condition.

$$Z_s @ 394 \text{ psia and } 130^{\circ}\mathrm{F} = 0.95$$

$$Z_d @ 1114.7 \text{ psia and } 270^{\circ}\mathrm{F} = 0.95$$

(b) Determine average normal clearance (C)

From Manufacturer's Data Sheet (Fig. 10.14) a 10.5-in. cylinder has an average normal clearance of 20.27%

(c) Calculate volumetric efficiency (E_v)

$$E_v = 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right]$$
$$= 96 - 2.87 - 20.27\left[\left(\frac{0.975}{0.97}\right)(2.87)^{\frac{1}{1.26}} - 1\right]$$
$$= 63.47\%$$

(12) Calculate capacity of selected cylinder
(a) Determine ACFM

$$\text{ACFM} = (\text{PD})E_v$$
$$= (591)(0.6347)$$
$$= 375.1\ \text{ft}^3/\text{min}$$

(b) Determine capacity in MMSCFD

$$Q_g = (0.051)\left(\frac{(\text{ACFM})P_s}{T_s Z_s}\right)$$
$$= (0.051)\left(\frac{(143.68)(394)}{(590)(0.95)}\right)$$
$$= 4.72\ \text{MMSCFD}$$

(c) *Note*: The capacity of 10.5-in. cylinder is less than desirable. The following options are available:
1. Pick the next larger size cylinder
2. Decrease the P_d of the first-stage cylinder
3. Live with 4.72 MMSCFD at $P_s = 125$ psig

Select Option #3.
Therefore set $Q_2 = 4.72$ MMSCFD and solve for ACFM

(13) Set $Q_2 = 4.72$ MMSCFD and solve for ACFM

$$Q_2 = (0.051)\left(\frac{(\text{ACFM})P_s}{T_s Z_s}\right)$$

Rearranging

$$4.72 = (0.051)\left(\frac{(\text{ACFM})(394)}{(590)(0.095)}\right)$$

Solving for ACFM

$$\text{ACFM} = 132.3\ \text{ft}^3/\text{min}$$

(14) Calculate volumetric efficiency (E_v)

$$\mathrm{ACFM} = \frac{\mathrm{ACFM}}{\mathrm{PD}}$$

Rearranging and solving for E_v

$$\begin{aligned} E_v &= \frac{\mathrm{ACFM}}{\mathrm{PD}} \\ &= \frac{132.2}{202.4} \\ &= 65.22\% \end{aligned}$$

(15) Calculate average cylinder clearance needed to satisfy the decreased volumetric efficiency

$$E_v = 96 - R - C\left[\left(\frac{Z_s}{Z_d}\right)(R)^{\frac{1}{k}} - 1\right] - L$$

Rearranging and solving for C

$$65.22 = 96 - 2.83 - C\left[\left(\frac{0.975}{0.97}\right)(2.83)^{\frac{1}{1.26}} - 1\right] - 1$$

$$C = 18.18\%$$

Therefore clearance to be added to the second-stage cylinder is

$$\begin{aligned} &= 18.18 - 15.17 \\ &= 3.01\% \end{aligned}$$

(16) Calculate rod loads in both tension and compression for first and second stages

(a) First-stage calculations

(1) Calculate area of piston and rod

$$A_p = \frac{\pi(10.5)^2}{4} = 86.6 \text{ in}^2$$

$$A_r = \frac{\pi(2)^2}{4} = 3.14 \text{ in}^2$$

(2) Calculate rod load in compression

$$\begin{aligned} \mathrm{RL}_c &= P_d A_p - P_s\left(A_p - A_r\right) \\ &= (386)(86.6) - (125)(86.36 - 3.14) \end{aligned}$$

$$= 22{,}995 \text{ lbs}$$

(3) Calculate rod load in tension

$$\begin{aligned}\mathrm{RL}_t &= P_d\left(A_p - A_r\right) - P_s\left(A_p\right)\\ &= 386(86.6 - 3.14) - 125(86.6)\\ &= 21{,}390 \text{ lbs}\end{aligned}$$

(b) Second-stage calculations

1. Calculate area of piston and rod

$$A_p = \frac{\pi(6.25)^2}{4} = 30.7 \text{ in}^2$$
$$A_r = \frac{\pi(2)^2}{4} = 3.14 \text{ in}^2$$

2. Calculate rod load in compression

$$\begin{aligned}\mathrm{RL}_c &= P_d\left(A_p\right) - P_s\left(A_p - A_r\right)\\ &= 1100(30.7) - 379(30.7 - 3.14)\\ &= 23{,}324 \text{ lbs}\end{aligned}$$

3. Calculate rod load in tension

$$\begin{aligned}\mathrm{RL}_t &= P_d\left(A_p - A_r\right) - P_s\left(A_p\right)\\ &= 1100(30.7 - 3.14) - 379(30.7)\\ &= 18{,}681 \text{ lbs}\end{aligned}$$

Allowable rod load for the FE 650A frame is 25,000 Ibs. Therefore all rod loads are satisfactory.

(17) Calculate the break horsepower

(a)

$$\mathrm{BHP}_1 = 0.0857[0.975]^{\frac{1}{1.26}}[0.97]^{\frac{(1.26-1)}{1.26}}\left[\frac{(4.72)(580)}{(.93)(.85)}\right] \times \left[\frac{(1.26)(1)}{(1.26-1)}\right]\left[\left(\frac{402}{140}\right)^{\frac{(1.26-1)}{(1.26)(1)}} - 1\right]$$
$$= 298$$

(b)

$$\mathrm{BHP}_1 = 0.0857(0.95)^{\frac{1}{1.26}}(0.95)^{\frac{1.26-1}{1.26}}\left(\frac{(4.72)(590)}{(.93)(.85)}\right) \times \left[\frac{(1.26)(1)}{(1.26-1)}\right]\left[\left(\frac{1114.7}{394}\right)^{\frac{(1.26-1)}{(1.26)(1)}} - 1\right]$$
$$= 294$$

(c)

$$\begin{aligned}\text{Total BHP} &= (\mathrm{BHP})_1 + (\mathrm{BHP})_2\\ &= 298 + 294\\ &= 592\end{aligned}$$

Example 10.13. Developing a Performance Map for a Multistage Compressor

Given:

Data in Example 10.12.

Determine:

Develop a performance curve for a suction pressure range between 100 and 150 psig.

Solution:

(1) Calculate the "ideal" compression ratio with $P_s = 100$ psig

$$R = \left(\frac{P_d}{P_s}\right)^{\frac{1}{n}}$$
$$= \left(\frac{1114.7}{139.7}\right)^{\frac{1}{2}}$$
$$= 3.12$$

(2) Calculate Q_z for $P_s = 358$ psia

(a) Calculate the second-stage suction pressure

$$P_{s2} = R(P_{s1})$$
$$= (3.12)(114.7)$$
$$= 358 \text{ psia}$$

(b) Calculate the gas discharge temperature

$$T_d = (T_{sR})^{\frac{1.26-1}{1.26}}$$
$$= 590(3.12)^{\frac{1.26-1}{1.26}}$$
$$= 746^{\circ}\text{R or } 286^{\circ}\text{F}$$

(c) Calculate the volumetric efficiency

$$E_v = 96 - (3.12) - 18.18\left[\left(\frac{0.95}{0.96}\right)(3.12)^{\frac{1}{1.26}} - 1\right]$$
$$= 63.29\%$$

(d) Calculate the capacity, Q_2, in MMSCFD with $P_s = 358$ psia

$$Q_2 = (0.051)\left(\frac{(\text{PD})(E_v)P_s}{T_s Z_s}\right)$$
$$= (0.051)\left(\frac{(202.4)(.6329)(358)}{(590)(0.95)}\right)$$
$$= 4.17 \text{ MMSCFD}$$

(3) Calculate Q_1 with $P_s = 100$ psig

(a) Calculate the gas discharge temperature

$$T_d = 580(3.17)^{\frac{1.26-1}{1.26}}$$
$$= 735^{\circ}\text{R or } 276^{\circ}\text{F}$$

(b) Calculate the volumetric efficiency

$$E_v = 96 - (3.17) - 20.27\left[\left(\frac{0.99}{0.98}\right)(3.17)^{\frac{1}{1.26}} - 1\right]$$
$$= 58.75$$

(c) Calculate Q_1 in MMSCFD

$$Q_2 = (0.051)\left(\frac{(\mathrm{PD})(E_v)P_s}{T_s Z_s}\right)$$
$$= (0.051)\left(\frac{(591)(0.5875)(114.7)}{(580)(0.99)}\right)$$
$$= 3.5 \text{ MMSCFD}$$

(4) $Q_2 > Q_1$. Since clearance needs to remain constant, an iteration must begin in order to achieve $Q_1 = Q_2$ by adjusting P_{s2}

(a) Trial #1:

1. Repeat Step 2 with $P_{s2} = 340$ psia

$$Td = 754°\text{R or } 294°\text{F}$$

$$E_v = 61.16\%$$

$$Q_2 = 3.825 \text{ MMSCFD}$$

2. Repeat Step 3 with $P_{s2} = 100$ psig

$$Td = 728^\circ\text{R or } 268^\circ\text{F}$$

$$E_v = 60.98\%$$

$$Q_2 = 3.67 \text{ MMSCFD}$$

3. Since $Q_2 > Q_1$, therefore adjust P_{s2} down again

(b) Trial #2:

1. Repeat Step 2 with $P_{s2} = 333$ psia

$$Td = 757^\circ\text{R or } 297^\circ\text{F}$$

$$E_v = 60.72\%$$

$$Q_2 = 3.72 \text{ MMSCFD}$$

2. Repeat Step 3 with $P_{s1} = 100$ psig

$$Td = 725^\circ\text{R or } 265^\circ\text{F}$$

$$E_v = 61.88\%$$

$$Q_2 = 3.72 \text{ MMSCFD}$$

3. $Q_2 = Q_1$

(5) Calculate BMP

$$\begin{aligned}\text{BHP}_1 &= 242\\ \text{BHP}_2 &= 271\\ \text{Total BHP} &= 242 + 271\\ &= 513\end{aligned}$$

(6) Calculate the "ideal" compression ratio with $P_s = 150$ psig

$$\begin{aligned}R &= \left(\frac{1114.7}{164.7}\right)^{\frac{1}{2}}\\ &= 2.60\end{aligned}$$

(7) Calculate Q_2 for $P_s = 428$

(a) Calculate the second-stage suction pressure

$$\begin{aligned}P_{s2} &= RP_{s1}\\ &= (2.6)(164.7)\\ &= 428 \text{ psia}\end{aligned}$$

(b) Calculate gas discharge temperature

$$\begin{aligned}T_d &= T_s R^{\frac{k-1}{k}}\\ &= (590)(2.6)^{\frac{1.26-1}{1.26}}\\ &= 718^\circ\text{R or } 258^\circ\text{F}\end{aligned}$$

(c) Calculate E_v
(d) Calculate Q_2

(8) Calculate Q_f with $P_s = 164.7$psia

(a) $T_d = 708°\text{R}$ or $248°\text{F}$
(b) $E_v = 67.18\%$
(c) $Q_1 = 5.86$ MMSCFD

(9) Since $Q_1 > Q_2$; adjust P_{s2} up

(a) Solve for Q_2: Repeat Steps 7 and 8 with $P_{s2} = 440$psia

$T_d = 714°\text{R}$ or $254°\text{F}$
$E_v = 70.94\%$
$Q_1 = 5.74$MMSCFD

(b) Solve for Q-,

$T_d = 772°R$ or $252°F$
$E_v = 66.12\%$
$Q_1 = 5.76MMSCFD$

(10) Calculate BHP
(11) Engine selection

Pick an engine (refer to Fig. 10.15) and plot the results on the performance curve.

Model	Continuous shaft ratings RPM						List price
	800	1000	1200	1400	1600	1800	
VRG155[1]	---	---	20	22	25	27	
HR3G[4]	---	16	21	24	28	31	
3DAG[1]	---	22	25	29	32	35	
VRG220[1] (LC)	---	---	---	32	35	36	
G-2300[7] (LC)	---	---	28	32	35	39	
G-2300[7] (HC)	---	---	32	36	40	43	
4DAG[4] (HC)	---	29	34	39	473	47	
G-3400[7] (LC)	---	---	41	45	51	56	
VRG263[1]	---	---	39	47	54	61	
G-34007 (HC)	---	---	46	52	58	63	
VRG330[1]	---	---	---	54	60	67	
6DAG[4] (HC)	---	43	51	57	65	71	
336A-4A[3]	35	45	54	62	68	---	
3304NA[2] (LC)	---	(40)	50	59	69	78	
3304NA[2] (HC)	---	(52)	61	71	82	89	
HD504A-6A[5]	49	62	74	85	95	104	
3306NA[2] (LC)	---	(61)	72	89	103	118	
3306NA[2] (HC)	---	(78)	89	107	122	136	
FB17G[1] (LC)	72	90	109	124	138	---	
FB17G[1] (HC)	75	94	113	130	148	---	
HD800A-6A[5]	82	104	123	142	---	---	
3306TA[2] (LC)	---	(103)	122	146	---	---	
F1197G[1] (LC)	101	122	149	172	---	---	
THD800A-6A[5]	100	128	152	181	---	---	
F1197G[1] (HC)	112	137	162	185	---	---	
G342NA[2] (LC)	122	160	188	---	---	---	
G342NA[2] (HC)	141	162	212	---	---	---	
F1905G[1] (LC)	168	203	233	---	---	---	
F1905G[1] (HC)	172	208	239	---	---	---	
G342TA[2] (LC)	178	227	256	---	---	---	
G379NA[2] (LC)	174	230	282	---	---	---	
F2475G[1] (LC)	227	273	309	---	---	---	
G379NA[2] (HC)	202	259	310	---	---	---	
F2475G[1] (HC)	239	283	323	---	---	---	
F2895G[1] (LC)	278	337	376	---	---	---	
8G-510[3] (LC)	270	337	---	---	---	---	
G379TA[2] (LC)	265	352	400	---	---	---	
F2695G[1] (HC)	302	369	421	---	---	---	
G396NA[2] (LC)	249	348	423	---	---	---	
8G-510[3] (HC)	306	383	---	---	---	---	
F3521G[1] (LC)	337	405	456	---	---	---	
G396NA[2] (HC)	301	387	470	---	---	---	
L3711G[1] (LC)	337	411	476	---	---	---	
L3711G[1] (HC)	353	427	486	---	---	---	
F3521G[1] (HC)	366	448	512	---	---	---	
G399NA[2] (LC)	348	465	564	---	---	---	
IR2500[4]	421	527	575	---	---	---	
G398TA[2] (LC)	410	531	603	---	---	---	
F2895GSI[1] (LC)	405	506	607	---	---	---	
G399NA[2] (HC)	409	517	620	---	---	---	
8G-625[3] (LC)	439	548	---	---	---	---	
F5106G[1] (LC)	489	588	661	---	---	---	
F3521GSI[1] (LC)	492	615	738	---	---	---	
8G-825[4] (HC)	496	622	---	---	---	---	
F5106G[1] (HC)	529	634	741	---	---	---	
L5709G[1] (LC)	554	667	750	---	---	---	
G399TA[2] (LC)	555	704	801	---	---	---	
F5790G[1] (HC)	604	738	842	---	---	---	

NOTES
1. Ratings are manufacturer's recommendation for continuous compressor duty at 100°F and 2000′ elevation with 1000 btu/cf fuel gas.
2. Deduct 3% of continuous shaft BHP if driving serial cooler from front and of engine
3. [1]Waukesha [2]Caterpillar [3]White [4]Superior [5]Dorman [6]Ingersoll-Rand [7]Minneapolis-Moline [8]Lister
4. (   ) Verify speed and BHP with manufacturer

Fig. 10.15 Natural gas engine ratings.

Select a Waukesha F5790G (HC)

- Continuous shaft rating of 738 HP @ 1000 rpm
- Deration for friction and driving a cooler fan results in 665 HP available for compression

$$= (738)(0.93)(0.97)$$
$$= 665\ \text{HP}$$

(12) Fig. 10.16 is the performance curve for the multistage compressor

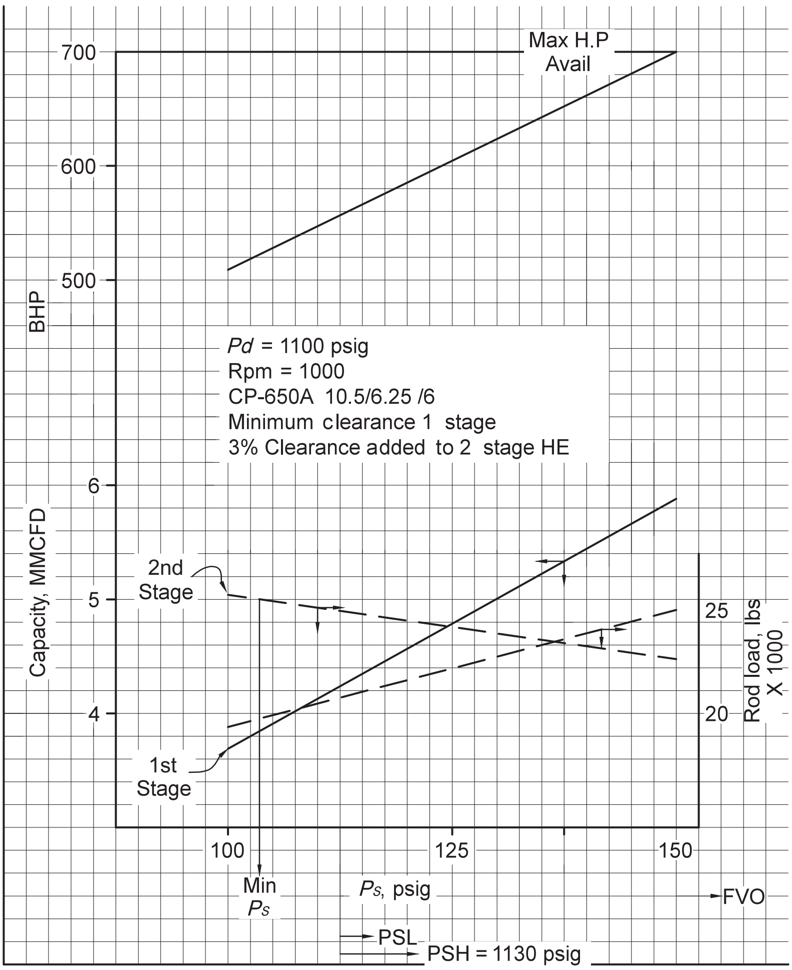


Fig. 10.16 Performance map for Example 10.13.

10.6 Determination of safety device set points

Safety device set points should be determined after evaluating the compressor performance map. The following parameters should be given the highest priority:

- Gas discharge temperature
- Rod load
- Break horsepower

Example 10.14. Determination of Safety Device Set Points.

Given:

Single-stage performance curve (Fig. 10.15) for a CP-6FE065 compressor with two 15-in. DA cylinders.

Maximum rod load is 28,000 Ibs.

Determine:

Determine the safety device set point with a 1000 Ib. safety, that is, a maximum allowable rod load of 27,000 Ibs.

Solution (refer to Fig. 10.17*)*:

(1) Determine the "suction side" PSL set point

The intersection of the maximum allowable rod load line and the actual rod load line establishes the PSL set point of 58 psi.

If the suction pressure decreases to 58 psi, the compressor will be protected by the low-pressure shutdown sensor.

(2) Determine the set point of the recycle valve

The recycle valve should be set between 5 and 8 psig above the PSL set point. Thus the recycle valve should open at 65 psig.

This will maintain suction pressure above the low-pressure set point, thus the compressor will not shut down unless there is a malfunction of the recycle valve.

(3) Determine the set point of the PSH on the cylinder discharge

(a) Using the recycle valve open (RVO) set point, determine the maximum discharge pressure that the cylinder could see without exceeding the maximum allowable rod load

(b) Calculate the area of piston and area of rod

$$A_p = \frac{\pi d^2}{4} = \frac{\pi (15)^2}{4} = 176.7\ \text{in}^2$$

$$A_r = \frac{\pi d^2}{4} = \frac{\pi (2.25)^2}{4} = 3.98\ \text{in}^2$$

(c) Determine rod load in compression

Set up and solve:

$$RL_c = P_d A_p - P_s (A_p - A_r)$$

$$27,000 = P_d (176.7) + 65(176.7 - 3.98)$$

$$P_d = 216\ \text{psig}$$

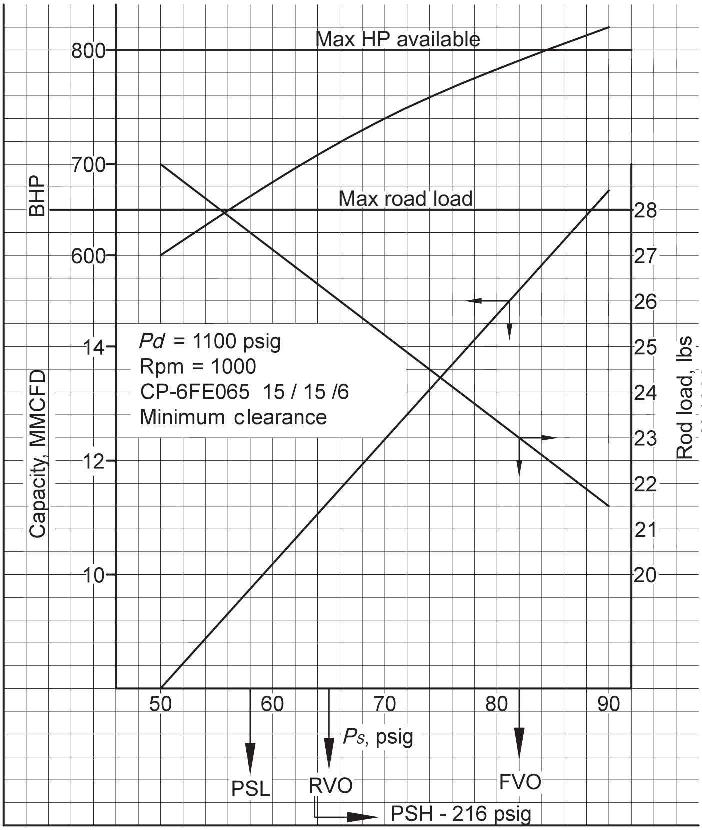


Fig. 10.17 Performance map for Example 10.14.

The PSH on the discharge should be set at 216 psig

(4) Determine the set point of the flare valve

The intersection of the actual BHP curve with the maximum BHP available yields the point where the flare valve should open, that is, 82 psig. Opening the flare valve at 82 psig prevents overloading the driver.

(5) Determine the set point of the PSH on the suction

The PSH on the suction should be set about 3–5 psig above the flare valve set point. Thus a value of 85 psig should be selected. This PSH protects the engine should the flare valve malfunction.

Example 10.15. Determination of Safety Device Set Points for a Two-Stage Compressor

Given:

Two-stage compressor performance curve (Fig. 10.18) and set maximum allowable rod load at 24,500 Ibs.

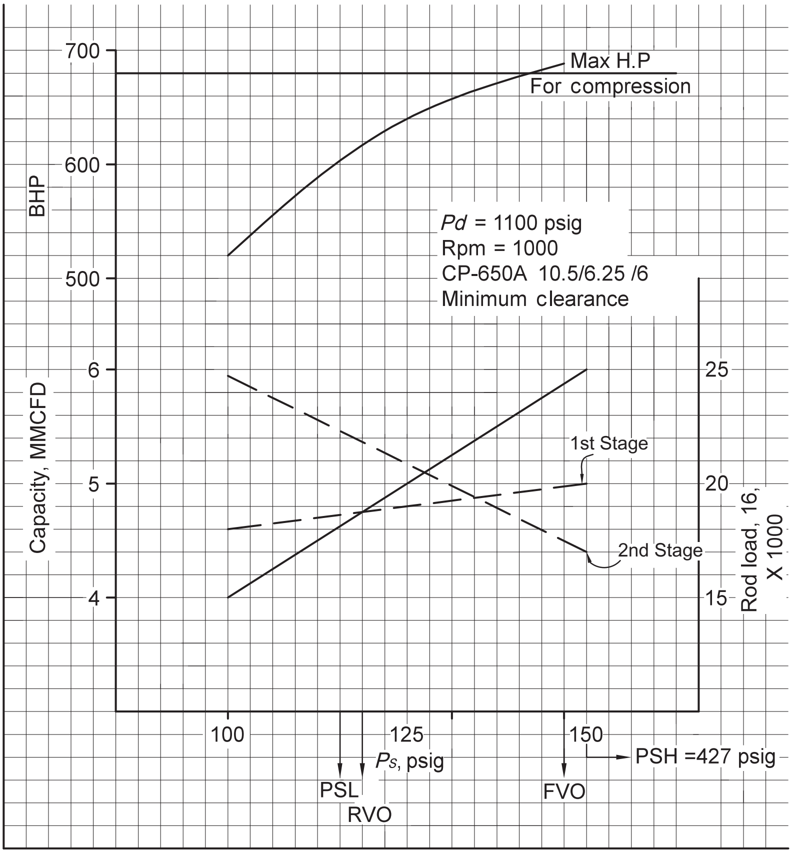


Fig. 10.18 Performance map for Example 10.15.

Determine:

Determine the safety device set points.

Solution:

(1) Determine the "suction side" PSL set point

The maximum allowable load line intersects the second-stage actual rod load line at $P_s = 115$ psig. Thus the PSL set point is $P_s = 115$ psig.

(2) Determine the set point of the recycle valve

The recycle valve is normally set between 5 and 8 psig above the PSL set point. Therefore set the recycle valve to open at 120 psig. This will maintain suction pressure above the low-pressure shutdown set point.

(3) Determine the set point of the flare valve

The intersection of the actual BHP curve with the maximum BHP available yields the point where the flare valve should open. From Fig. 10.18, read 147 psig. Therefore set the flare valve to open at 147 psig so as to prevent the engine from overloading.

(4) Determine the set point of the PSH on the first-stage suction

The PSH on the first-stage discharge should be set to (1) protect the first-stage cylinder and (2) protect the engine from overloading should the flare valve malfunction. Therefore determine the discharge pressure, P_s, of the first stage at a suction pressure (P_s) of 150 psig.

$$\mathrm{RL_c} = P_d A_p - P_s\left(A_p - A_r\right)$$

$$24{,}500 = P_d(86.6) - 150(86.6 - 3.14)$$

$$P_d = 427 \text{ psig}$$

$$\therefore \text{Set PSH}_1 = 427 \text{ psig}$$

(5) Determine the set point of PSH_2 on the second-stage discharge

(a) Set up and solve the rod load equation for P_d of the first stage using $P_s = \mathrm{RVO}$

$$\mathrm{RL_c} = P_d A_p - P_s\left(A_p - A_r\right)$$

$$21{,}000 = P_d(86.6) - 120(86.6 - 3.14)$$

$$P_d = 358 \text{ psig}$$

(b) Set up and solve the rod load equation for

$$P_{d1} = P_S 2$$

$$\mathrm{RL_c} = P_d A_p - P_s\left(A_p - A_r\right)$$

$$24{,}500 = P_d(30.7) - 358(30.7 - 3.14)$$

$$P_d = 1120 \text{ psig}$$

Therefore set $\text{PSH}_2 = 1120\,\text{psig}$.

10.7 Determination of capacity from *P-V* diagram

A gas compressor's throughput (capacity) can be calculated with the aid of a *P-V* indicator diagram. The following procedure is useful when cylinder clearances are unavailable. The procedure follows the same steps as previously outlined in determination of capacity. Procedure is as follows:

(1) Calculate head-end and crank-end swept volume in ft^3

$$\mathrm{VOL_{HE}}\left(\mathrm{ft}^3\right) = \left(\frac{\pi D^2(\mathrm{in})}{4}\right)\left(\frac{\mathrm{Stroke(in)}}{1728\dfrac{\mathrm{in}^3}{\mathrm{ft}^3}}\right)$$

(2) Calculate displaced volume

$$\mathrm{VOLdis}\left(\mathrm{ft}^3\right) = \mathrm{VOL}_{\mathrm{HE}}\left(\mathrm{ft}^3\right)\mathrm{VE}_{\mathrm{HE}}(\mathrm{From}\, P - V\,\mathrm{Diagram})$$

$$\mathrm{VOLdis}\left(\mathrm{ft}^3\right) = \mathrm{VOL}_{\mathrm{CE}}\left(\mathrm{ft}^3\right)\mathrm{VE}_{\mathrm{CE}}(\mathrm{From}\, P - V\,\mathrm{Diagram})$$

(3) Calculate ACFM

$$\mathrm{ACFM} = \mathrm{VOL}_{\mathrm{dis.HE}}\left(\mathrm{ft}^3\right) + \mathrm{VOL}_{\mathrm{dis.CH}}\left(\mathrm{ft}^3\right)(\mathrm{RPM})$$

(4) Convert ACFM to MMCFD

$$\mathrm{MMCFD} = (\mathrm{CFM})\left(\frac{1440\ \mathrm{Min}}{\mathrm{Day}}\right)\left(\frac{\mathrm{MMCF}}{10^6\mathrm{CF}}\right)\left(\frac{P_s}{P_a}\right)\left(\frac{T_a}{T_s}\right)\left(\frac{1}{Z_s}\right)$$

Example 10.16. Determination of Capacity From a *P*-*V* Indicator Diagram

Given:

Cylinder Bore	14″
Cylinder Stroke	14″
Piston Rod Diameter	3″
VE_{HE}	78%
VE_{CE}	86%
Speed	330 RPM
P_s	25 PSIG
T_s	100º
Z_s	1.25

Determine:

Determine capacity.

Solution:

(1) Swept volume

$$\mathrm{VOL}_{\mathrm{HE}} = \left(\frac{\pi D^2}{4}\right)\left(\frac{\mathrm{Stroke}}{1728}\right)$$

$$= \left(\frac{\pi(14\ \mathrm{in})^2}{4}\right)\left(\frac{14\ \mathrm{in}}{1728\dfrac{\mathrm{in}^3}{\mathrm{ft}^3}}\right) = 1.247\ \mathrm{ft}^3$$

$$\mathrm{VOL}_{\mathrm{CE}} = \frac{\pi}{4}\left[(D\,\mathrm{in})^2 - (d\,\mathrm{in})^2\right]\left(\frac{\mathrm{Stroke}}{1728}\right)$$

$$= \frac{\pi}{4}\left[\left(14^2 - 3^2\right)\right]\left(\frac{14}{1728}\right) = 1.19\ \mathrm{ft}^3$$

(2) Calculate displaced volume

$$VOL_{dis.HE} = (1.27)(0.78) = 0.973$$

$$VOL_{dis.CE} = (1.190)(0.86) = 1.023$$

$$\text{Total} = 0.973 + 1.023 = (1.996)(330) = 659 \text{ ft}^3/\text{min}$$

(3) Convert to MMCFD

$$\begin{aligned} &= (659\text{CFM})\left(\frac{1440\,\text{Min.}}{\text{Day}}\right)\left(\frac{\text{MMCF}}{10^6\text{CF}}\right)\left(\frac{39.7}{14.7}\right)\left(\frac{520}{560}\right)\left(\frac{1}{1.25}\right) \\ &= 1.9\text{MMCFD} \end{aligned}$$

10.8 Compressor suction and discharge line sizing

Suction, discharge, and interstage piping should be sized to minimize

- Pulsation
- Vibration
- Noise

Allowable gas velocities

- Suction = 2000 ft./min
- Discharge = 3000 ft./min

Pipe inside diameter can be calculated using the following equation

$$V_g = \frac{60ZQT}{d_i^2} \tag{10.1}$$

Rearranging and solve for ID

$$d_i = \left(\frac{60ZQT}{V_g P}\right)^{0.5} \tag{10.2}$$

where:

V = Gas velocity, ft./s
d_i = Pipe ID, inches
Q = Gas flow rate, MMSCFD
Z = Gas compressibility factor, psia
T = Operating temperature, °R

Example 10.17. Sizing Suction and Discharge Piping

Given:

Q = 4.75 MMSCFD
Z_1 = 0.95
Z_2 = 0.98

$T = 580\ ^{\circ}\mathrm{R}$
$V_s = 33.33\ \mathrm{ft./s}$
$V_d = 50\ \mathrm{ft./s}$
$P_1 = 139.7\ \mathrm{psia}$
$P_2 = 404\ \mathrm{psia}$
$R = 2.88$

Determine:
Size the compressor first-stage suction and discharge pipe sizes.

Solution

(1) Calculate the ID of the first-stage suction piping

$$\begin{aligned} d_i &= \left(\frac{60ZQT}{V_g P}\right)^{0.5} \\ &= \left(\frac{(60)(0.95)(4.75)(580)}{(33.33)(139.7)}\right)^{0.5} \\ &= 5.8 \text{ inches} \end{aligned}$$

Choose a 6-in. standard weight pipe, MAWP = 1206 psig for first-stage suction.

(2) Calculate the first-stage discharge pipe ID

(a) Determine discharge temperature

$$\begin{aligned} T_d &= T_s(R)^{\frac{k-1}{k}} \\ &= 580(2.88)^{\frac{1.26-1}{1.26}} \\ &= 721^{\circ}\mathrm{R} \end{aligned}$$

(b) Calculate first-stage discharge ID

$$\begin{aligned} d_i &= \left(\frac{(60)(0.98)(4.75)(721)}{(50)(404)}\right)^{0.5} \\ &= 3.16 \text{ inches} \end{aligned}$$

Choose a 4″ standard weight pipe, MAWP = 1439 psig for first-stage discharge.

(3) Follow the same procedure to size the remainder of the piping

10.9 Pulsation bottles

The purpose of pulsation bottles is to reduce pulsations due to gas flow. They are located on both the suction and discharge sides of each cylinder.

The GPSA Engineering Data Book method uses a multiplier to determine bottle volume. Bottle volume is determined from the following equation:

$$V_B = \left(V_{\mathrm{Swept}}\right)(C_m) \tag{10.3}$$

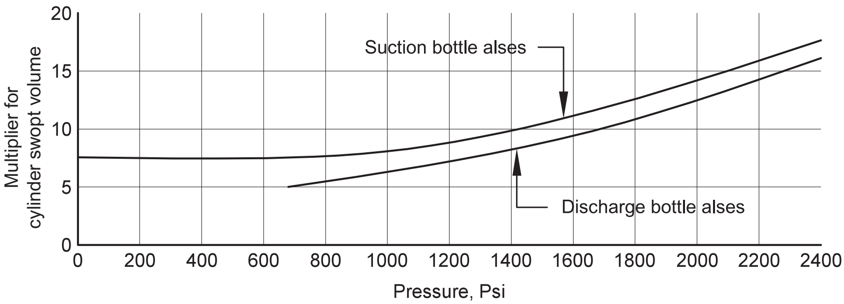


Fig. 10.19 GPSA bottle sizing chart.

where:

V_B = Volume of pulsation bottle, in^3
V_{Swept} = Cylinder swept volume, in^3

$$=\frac{\pi D_c^2}{4}(\text{Stroke})$$

D_c = Cylinder diameter, inches
C_m = Multiplier
= Refer to Fig. 10.19.

Diameter of bottle is determined from the following equation:

$$D_B = 0.86 V_B^{\frac{1}{3}} \tag{10.4}$$

where:

D_B = Minimum diameter of volume bottle, inches

Length of the pulsation bottle is determined from following equation:

$$L = 2D_B \tag{10.5}$$

where:

L = Overall length of the bottle from end-cap to end-cap, inches

Example 10.18. Sizing Suction and Discharge Pulsation Bottles

Given:

Compressor cylinder diameter = 10.5 in.
Suction pressure (P_s) = 125 psia
Stroke = 6 in.
Temperature = 300°F

Corrosion allowance = 1/8 in.
Discharge pressure (P_d) = 428 psia

Determine:

Design the suction and discharge pulsation bottle for the first stage of a two-stage compressor.

Solution:

(1) Calculate compressor cylinder swept volume (V_{Swept})

$$V_{\text{Swept}} = \frac{\pi D_c^2}{4}(\text{Stroke}) = \frac{\pi (10.5)^2}{4}(6) = 519.5\ \text{in}^3$$

(2) Determine multiplier (C_m) from bottle sizing chart (Fig. 10.19) for suction bottle. For suction pressure of $P_s = 125$ psia read $C_m = 7.5$

(3) Calculate suction pulsation bottle volume

$$V_B = (V_{\text{Swept}})(C_m) = (519.5)(7.5) = 3896.3\ \text{in}^2$$

(4) Calculate bottle diameter

$$D_B = 0.86(V)^{\frac{1}{3}} = (0.86)(3,896.3)^{\frac{1}{3}} = 13.2\ \text{inchs}$$

Select 14-in. pipe. This allows us to use a standard size welding cap.

(5) Calculate suction bottle wall thickness

Refer to worksheet (Fig. 10.20).

Design parameters

$P = 210$ psi (285 psi derated for temperature of 300°F)
$T = 300$°F
CA = 0.125 in.
WT = 0.375 (Fig. 10.20) (from worksheet)
ID = 14–2 (0.375)
= 13.25 in.

(6) Calculate bottle length

(a) Determine volume of pipe

- Suction pulsation bottle volume = 3896.3
- Less volume of caps (685 in^3 each) = − 1370.0.
- Volume of pipe = 2526.3 in^3.

Note: Volume of welding caps is presented in Table 10.8.

Design conditions							
Design pressure,	P = 210 psi						
Design temperature	= 300 F				Subject 1st Stage Suction Bottle		
Diameter,	D = 14 in						
Material SA S16-70	S = 17500 psi				Required wall thickness for internal pressure		
Radiographic exam.	E = 1.00						
Corrosion allowance	= 0.125 in				Computed	Checked	Number

Values of SE	E = 1.00		E = 0.85		E = 0.80		
	SE	2SE	SE	2SE	SE	2SE	
	17500	35000	14875	29750	14000	28000	
	15000	30000	12750	25500	12000	24000	
	13800	27600	11730	23460	11040	22080	

Flange Rating 150 lb.
Max. Allow. Pressure@100F 275 psi.
Max. Allow. Pressure@300F 210 psi.
Hydrostatic Test Pressure 315 psi.

Required wall thickness, *t*

		SHELL	CONE	HEAD Hemist	HEAD 2:1 E11	HEAD F & D
Formulas	I.S.	$t = \frac{PR_i}{SE - 0.6P}$	$t = \frac{PD_i}{2\cos\alpha(SE - 0.6P)}$	$t = \frac{PR_i}{2SE - 0.2P}$	$t = \frac{PD_i}{2SE - 0.2P}$	$t = \frac{PR_i}{SE - 0.6P}$
	O.S.	$t = \frac{PR_o}{SE + 0.4P}$	$t = \frac{PD_o}{2\cos\alpha(SE + 0.4P)}$	$t = \frac{PR_o}{2SE + 0.8P}$	$t = \frac{PD_o}{2SE + 1.8P}$	$t = \frac{0.885PL}{SE + 0.8P}$
For Design Pressure Corroded Condition E=						
For Design Pressure Corroded Condition E= 1.00						
For MAWP Corroded Condition E= 1.00		$\frac{210 \times 7}{17500 + 0.4(210)} =$ $0.083 + CorrAllow \frac{.125}{0.2086}$ $Use\ STD\ 0.375$		$\frac{210 \times 14}{35000 + 1.8(215)} = 0.083 + CorrAllow \frac{0.125}{0.208}$ $Use\ STD\ W\ T$		
For MAWP New & Cold Condition E= 1.00						

Maximum allowable working pressure, *P*

		SHELL	CONE	HEAD Hemist	HEAD 2:1 E11	HEAD F & D
Formulas	I.S.	$P = \frac{SE_t}{R_i - 0.6t}$	$P = \frac{2SE_t\cos\alpha}{D_i + 1.2t\cos\alpha}$	$P = \frac{2SE_t}{R_i + 0.2t}$	$P = \frac{2SE_t}{D_i + 0.2t}$	$P = \frac{SE_t}{0.885L + 0.1}$
	O.S.	$P = \frac{SE_t}{R_o - 0.4t}$	$P = \frac{2SE_t\cos\alpha}{D_i - 0.8t\cos\alpha}$	$P = \frac{2SE_t}{R_o - 0.8t}$	$P = \frac{2SE_t}{D_i - 1.8t}$	$P = \frac{SE_t}{0.885L - 0.1}$
Corroded Condition E= 1.00		$\frac{17500 \times .250}{7 - 0.4(.250)} = 634 psi$	Max Allow W.P. limited by 150# ANSI Flanges	$\frac{35000 \times .250}{14 - 1.8(.250)} = 646 psi$		
New & Cold Condition E=						
New & Cold Condition E= 1.00						

Fig. 10.20 Pressure vessel wall thickness worksheet for suction bottle.

Table 10.8 Welding caps

Pipe size	Standard weight		Extra strong		Double extra strong	
	Volume, cu. in.	**Length, in.**	**Volume, cu. in.**	**Length, in.**	**Volume, Cu. in.**	**Length, in.**
4″	24.2	2/1/2002	20	2/1/2002	15	3
6″	77.3	3/1/2002	65.7	3/1/2002	48	4
8″	148.5	4/11/2016	122.3	4/11/2016	120	5
10″	295.6	5/3/2004	264.4	8/3/2004		
12″	517	6/7/2008	475	6/7/2008		
14″	684.6	7/13/2016	640	7/13/2016		
16″	967.6	9	911	9		
18″	1432.6	10/1/2016	1363	10/1/2006		
20″	2026.4	11/1/2004	1938	11/1/2004		
24″	3451	13–7/16	3313	13–7/16		

(b) Determine bottle length (L):

$$V = \left(\frac{\pi D^2}{4}\right)L$$

$$2526.3 = \left(\frac{(\pi)(14)^2}{4}\right)$$

$$L = 16.4 \text{ inches}$$

(7) Determine multiplier (C_m) from bottle sizing chart (Fig. 10.19) for discharge bottle. For discharge pressure of $P_d = 428$ psia, read $C_m = 5.0$

(8) Calculate discharge pulsation bottle volume

$$\begin{aligned} V_B &= (V_{\text{Swept}})(C_m) \\ &= (519.5)(5) \\ &= 2597.5 \end{aligned}$$

(9) Calculate bottle diameter

$$\begin{aligned} D_b &= 0.86(V)\frac{1}{3} \\ &= (0.86)(2526.3)^{\frac{1}{3}} \\ &= 11.68 \text{ inch} \end{aligned}$$

Select 12 in., WT = 0.375, pipe with a MAWP = 888 psig. Again, this allows us to use a standard wt welding cap.

(10) Discharge bottle wall thickness

Refer to worksheet (Fig. 10.19).

Design parameters

$P = 675$ psi (740 psi derated for temperature of 350°F)
$T = 350$°F
CA = 0.125 in.
WT = 0.375 (Fig. 10.21) (from worksheet)
ID = 12.750–2(0.375)
= 12 in.

(11) Calculate bottle length

(a) Determine volume of pipe

- Suction pulsation bottle volume = 2526.3
- Less volume of caps (517 in^3 each) = −1034.0.
- Volume of pipe = 1492.3.

Note: Volume of welding caps is presented in Table 10.8.

(b) Determine bottle length (L)

$$V = \left(\frac{\pi D^2}{4}\right)L$$

Rearranging and solving for L

$$L = \left(\frac{(V)4}{\pi D^2}\right)$$
$$= \frac{(1492.3)(4)}{\pi (12)^2}$$
$$= 13.2 \text{ inches}$$

(12) Discharge pulsation bottle design summary

1 2.75″ OD × 13.5 SS
6″ - 600# ANSI RF WN Inlet Flange.
4″ - 300# ANSI RF WN Outlet Flange.
Design pressure = 675 psi.
Design temperature = 350°F.

10.10 Exercises

(1) Given the data below, develop the performance curves for a reciprocating compressor. Using linear graph paper, flow rate, BHP, and Rod Load versus Suction Pressure (use suction pressures of 90, 115, and 140 psig, respectively).

Discharge pressure = 315 psia
Suction temperature = 90°F

Design conditions		
Design pressure,	P = 675 psi	
Design temperature	= 350 F	Subject 1[st] Stage Suction Bottle
Diameter,	D = 12.75 in	
Material SA S16-70	S = 17500 psi	Required wall thickness for internal pressure
Radiographic exam.	E = 1.00	
Corrosion allowance	= 0.125 in	Computed / Checked / Number

Values of SE	E = 1.00		E = 0.85		E = 0.80		
	SE	2SE	SE	2SE	SE	2SE	
	17500	35000	14875	29750	14000	28000	Flange Rating 300 lb.
	15000	30000	12750	25500	12000	24000	Max. Allow. Pressure@100F 720 psi.
	13800	27600	11730	23460	11040	22080	Max. Allow. Pressure@300F 675 psi.
							Hydrostatic Test Pressure 1013 psi.

Required wall thickness, *t*

		SHELL	CONE	HEAD Hemist	HEAD 2:1 E11	HEAD F & D
Formulas	I.S.	$t=\frac{PR_i}{SE-0.6P}$	$t=\frac{PD_i}{2\cos\alpha(SE-0.6P)}$	$t=\frac{PR_i}{2SE-0.2P}$	$t=\frac{PD_i}{2SE-0.2P}$	$t=\frac{PR_i}{SE-0.6P}$
	O.S.	$t=\frac{PR_O}{SE+0.4P}$	$t=\frac{PD_O}{2\cos\alpha(SE+0.4P)}$	$t=\frac{PR_O}{2SE+0.8P}$	$t=\frac{PD_O}{2SE+1.8P}$	$t=\frac{0.885PL}{SE+0.8P}$
For Design Pressure Corroded Condition E=						
For Design Pressure Corroded Condition E= 1.00		$\frac{675\times 6}{17500-0.4(623)}=0.238+CorrAllow=\frac{0.125}{333}$ Use STD 35 WT		$\frac{675\times 12.75}{25000+1.8(675)}=0.338+CorrAllow\frac{0.125}{363}$ Use STD W.T.		
For MAWP Corroded Condition E= 1.00						
For MAWP New & Cold Condition E= 1.00						

Maximum allowable working pressure, *P*

		SHELL	CONE	HEAD Hemist	HEAD 2:1 E11	HEAD F & D
Formulas	I.S.	$P=\frac{SE_t}{R_i-0.6t}$	$P=\frac{2SE_t\cos\alpha}{D_i+1.2t\cos\alpha}$	$P=\frac{2SE_t}{R_i+0.2t}$	$P=\frac{2SE_t}{D_i+0.2t}$	$P=\frac{SE_t}{0.885L+0.1}$
	O.S.	$P=\frac{SE_t}{R_o-0.4t}$	$P=\frac{2SE_t\cos\alpha}{D_i-0.8t\cos\alpha}$	$P=\frac{2SE_t}{R_o-0.8t}$	$P=\frac{2SE_t}{D_i-1.8t}$	$P=\frac{SE_t}{0.885L-0.1}$
Corroded Condition E= 1.00		$\frac{17500\times 250}{6.315-0.4(250)}=697psi$	Max Allow W.P. limited by 300# ANSI Flanges	$\frac{35000\times 250}{1275-1.8(250)}=711psi$		
New & Cold Condition E=						
New & Cold Condition E= 1.00						

Fig. 10.21 Pressure vessel wall thickness for discharge bottle.

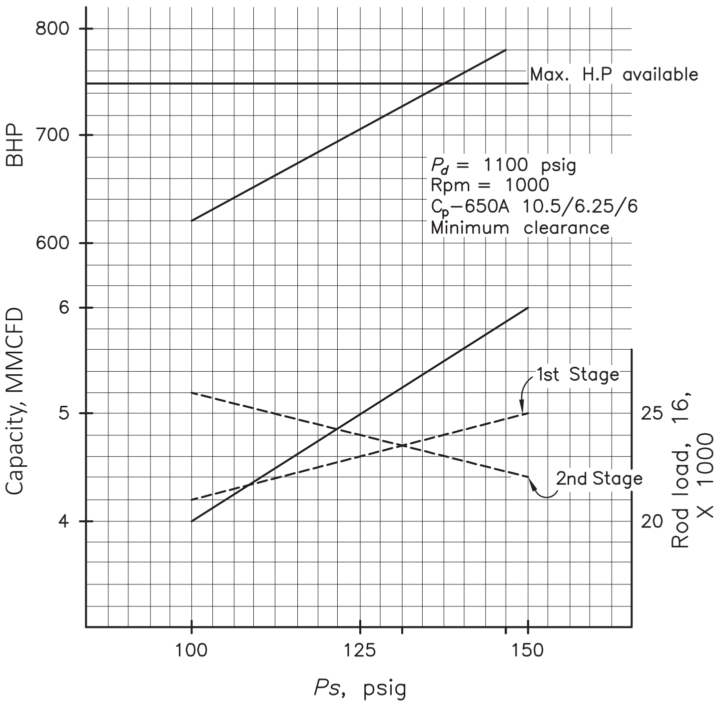


Fig. 10.22 Performance map for Exercise 2.

Cylinder diameter = 12″ double acting
Rod diameter = 3″
Stroke = 14″
Average clearance = 12%
Ratio of specific heats = 1.25
Gas SPECIFIC gravity = 0.6
Speed = 300 rpm
Rod load in compression = 50,000 lbs.
Rod load in tension = 50,000 lbs.
Mechanical efficiency = 94%
Adiabatic efficiency = 87%

Assume High-Speed Separable Reciprocating compressor is used.

(2) Speed is proportional to capacity and horsepower. Changing the speed of a compressor effects capacity control only. Speed does not affect compression ratios, discharge temperature, and rod loads. Given the attached two-stage reciprocating compressor performance curve (Fig. 10.22), develop the compressor's performance if the speed is reduced to 800 rpm.

Reference

Gas Processors Suppliers Association (GPSA), 2004. Gas Processors Suppliers Association (GPSA) Engineering Data Book, 12th ed. FPS Gas Processors Association, Tulsa, OK.

Rotary compressors

11

11.1 Engineering principles

11.1.1 Background

Rotary compressors are a subset of positive displacement machines. Rotary compressors have a number of features in common even though there are differences in construction. Unlike reciprocating compressors, rotary compressors do not use valves to move the gas through the machine. When compared to a reciprocating compressors rotary compressors:

- Are lighter in weight
- Experience less vibration
- Do not require heavy foundations

Rotary compressors physical design varies widely. Both single- and multiple-rotor construction are used. The design of the rotor is the main item that distinguishes the different types of rotary compressors.

11.1.2 Arrangements

Most often, rotary compressors are arranged as a single-rotor unit with a driver. Sometimes the compressors are also used in series arrangements, with or without an intercooler. Series arrangements use either tandem drives or multiple pinion gears to allow connection to a common driver.

Most rotary compressors used in the process industry use electric motor drivers. Compressors used in portable service use internal combustion engines. Many rotary compressors require the high speed that can be obtained from a direct-connected motor. The major exception is the helical lobe type as the units operate above motor speed and require a speed-increasing gear. That said, engines are used extensively as drivers for rotaries located in field-gathering facilities. Steam turbines, to a lesser degree, probably are used most often in large downstream petrochemical plants.

11.1.3 Application

The niche services for rotary compressors include Freon and ammonia refrigeration, plant air, some wet services, and services with vacuum suction conditions. Other types of compressors cover a majority of the range of uses. Efficiencies or rotary compressors are better than centrifugal and axial but lower than reciprocating. Speed control is the most obvious way to control capacity for rotary compressors, but they are usually direct driven by motors and run at constant speed. Capacity is controlled by bypassing

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00011-9

or blow off. Certain types of screw compressors have capacity control utilizing a side valve which reduces volumetric efficiency.

They are called blowers when they are used to move large volumes of air/gas at low compression ratios. A blower is a low-ratio compressor.

The remainder of the chapter covers some of the more common rotary compressors.

11.2 Screw compressors

Initially, screw compressors functioned between the centrifugal and reciprocating compressors. Their application areas have expanded. Today, the larger screw-type machines range up to 40,000 cfm and definitely cross into the centrifugal area. The smaller oil-flooded-type machines are used for automotive air-conditioning service, thus completely overlapping the reciprocating compressor in volume. The dry type generally ranges up to ~50 cfm.

Compression is achieved by the intermeshing of the male and female rotors. One rotor (the male) has helical lobes machined into the shaft which mesh with the second rotor (the female) with flutes or valleys machined into the shaft. Power is applied to the male rotor, and as a lobe of the male rotor starts to move out of mesh with the female rotor a void is created and gas is taken in at the inlet port. As the rotor continues to turn, the intermesh space is increased and gas continues to flow into the compressor until the entire interlobe space is filled. Further rotation uncovers the discharge port and the compressed gas starts to flow out of the compressor.

Fig. 11.1 shows the terminology used in a screw compressor. There are generally four lobes and six flutes that are machined to fit into each other, thus there are four compression cycles per revolution. The rotors are machined such that the volume between the lobe and the flute reduces along the axial length of the rotor. The gas is compressed by the rotary motion of the two rotors as the gas travels axially along the outside of the rotors from the suction end to the discharge.

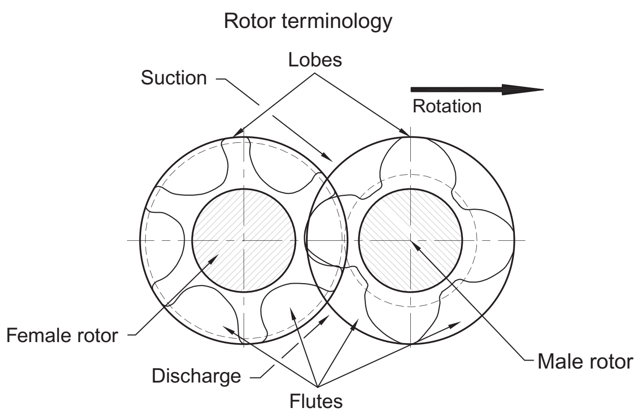


Fig. 11.1 Screw compressor rotor terminology.

11.2.1 Dry screw (oil-free screw) vs oil-flooded screw

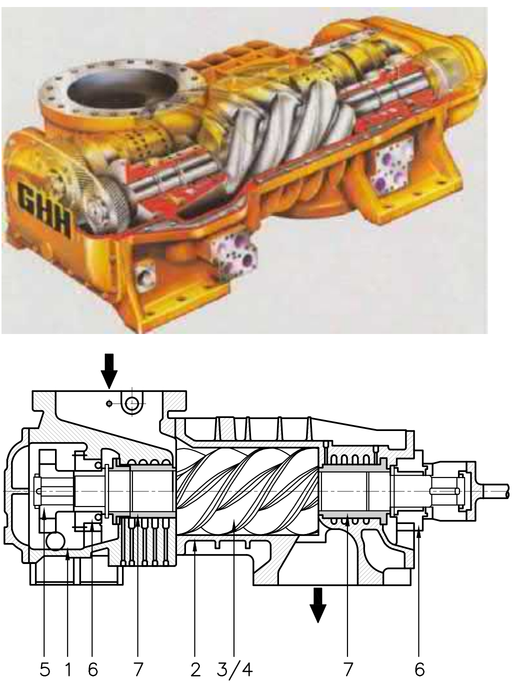
There are two types of screw compressors:

- Dry screw (oil-free screw (OFS)) type (Fig. 11.2)
- Oil-flooded screw type (Fig. 11.3)

In general, screw compressors used in the process industry are normally the dry screw type while oil-flooded screw compressors are commonly used in air or refrigeration service.

In the oil-flooded screw the male rotor is driven by the power source and the female rotor is driven by the male rotor. Oil is injected with the suction gas to lubricate, seal, and cool the rotors. The oil is removed from the discharge gas in a complex oil separation system.

In the dry screw compressor (refer to Fig. 11.4), the male rotor is driven by the power source and the female rotor is driven by a set of timing gears at the nondriven



1. Inlet casing
2. Rotor casing
3. Male rotor
4. Female rotor
5. Synchronizing gears
6. Bearings
7. Sealing

Fig. 11.2 Sectional of a dry screw compressor.

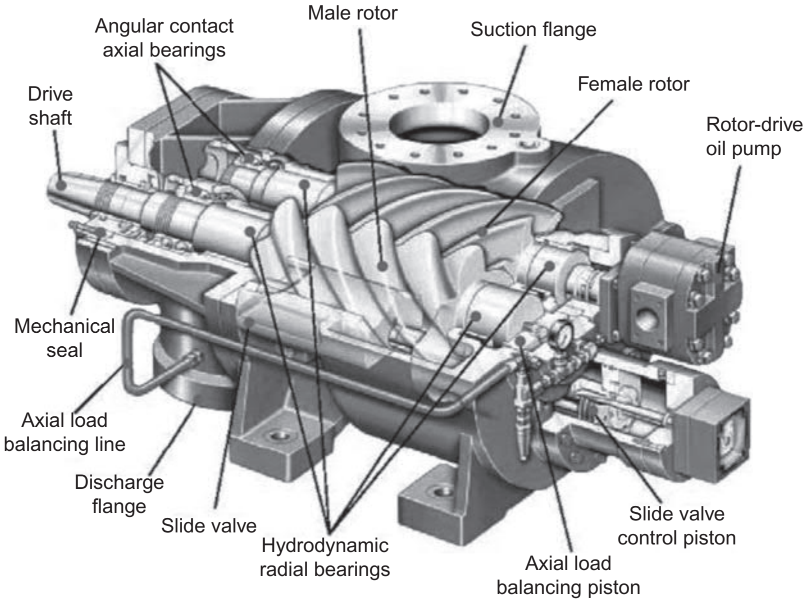
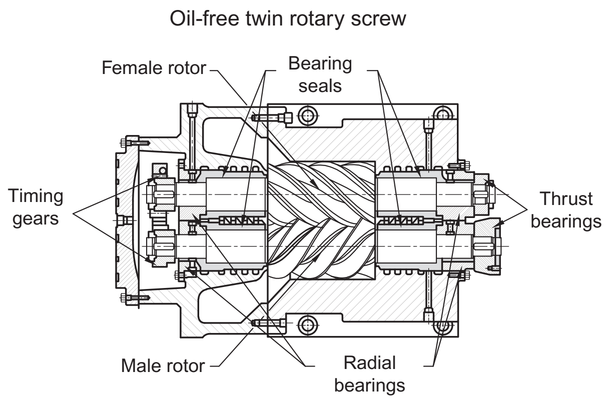


Fig. 11.3 Sectional of an oil-flooded-type screw compressor. Courtesy of Ariel Corporation.

Fig. 11.4 Sectional of a twin dry screw (oil-free screw) type compressor showing male and female rotors, bearings, and timing gears.



end of the rotors. As the name implies, there is no oil in the process gas and the rotors do not have contact with each other.

Oil-flooded screw compressors have a higher volumetric efficiency than dry screws because the oil acts as a seal. Dry screws must compensate for leakage by operating at higher speeds. High ratios are possible due to the cooling effect of the oil, and the oil acts to muffle the noise of compression.

Oil-flooded screw compressors have the disadvantage of oil contamination in the gas and have a complex oil separation and recovery system. Hydrocarbons can be absorbed in the oil and dilute the lubrication properties of the oil. This reduces the efficiency by having gas absorbed in the compression being flashed in the separator and having to be recompressed. Expensive synthetic lube oils are usually required.

The oil-flooded screw is the type of rotary compressor commonly used in Freon and ammonia refrigeration and small plant air services. Fig. 11.5 shows the components of an oil-flooded screw-type compressor. Fig. 11.6 is a schematic showing the oil system of an oil-flooded screw-type compressor. Oil is injected at the inlet of oil-flooded screw-type compressors. They are similar in construction to dry screw compressors, but they are smaller and have more open internal clearances. They can have more open internal clearances because the injected oil help seal the clearances. They have an integral slide valve that controls capacity. They are packaged with driver, baseplate, lube oil system, and controls. This type of rotary compressor is commonly used in Freon and ammonia refrigeration and small plant air services.

11.2.2 Dry screw (oil-free screw) type vs a reciprocating compressors

Both are positive displacement machines, but the compression principle of the screw compressor is rotational rather than reciprocating. There are no large reciprocating weights with dry screw compressors that have to be balanced and as a result no large foundation requirements are necessary. Screw compressors have no valves or high wear components so availability and reliability are higher with the screw compressors. Higher compression ratios are possible with dry screw compressors.

Since the gas in screw compressors is continuously compressed there are no low-frequency gas pressure pulsations in the flow. However, screw compressors are subject to high-frequency gas pulsations that are multiples of the rotor pocket passing frequencies. These frequencies create considerable noise which can be damaging and highly objectionable to the working environment. Large silencers and noise enclosures are required.

11.2.3 Dry screw (oil-free screw) type vs centrifugal compressors

Since the centrifugal compressor is a dynamic machine, discharge pressure and molecular weight of the process gas are important factors. The screw compressor is not subject to surge as is the centrifugal machine. Screw compressors also have the ability to handle gasses with changing molecular weights. Screw compressors operate with lower tip speeds which allow them to more easily process gas containing entrained liquid droplets or small particles.

11.2.4 Application ranges and selection

Table 11.1 summarizes the application ranges and selection of rotary compressors.

Rotary screw compressors are available in oil-flooded, drop-lubricated, “dry” (oil-free), and liquid design. When the oil-flooded or liquid-ring version is applied, the

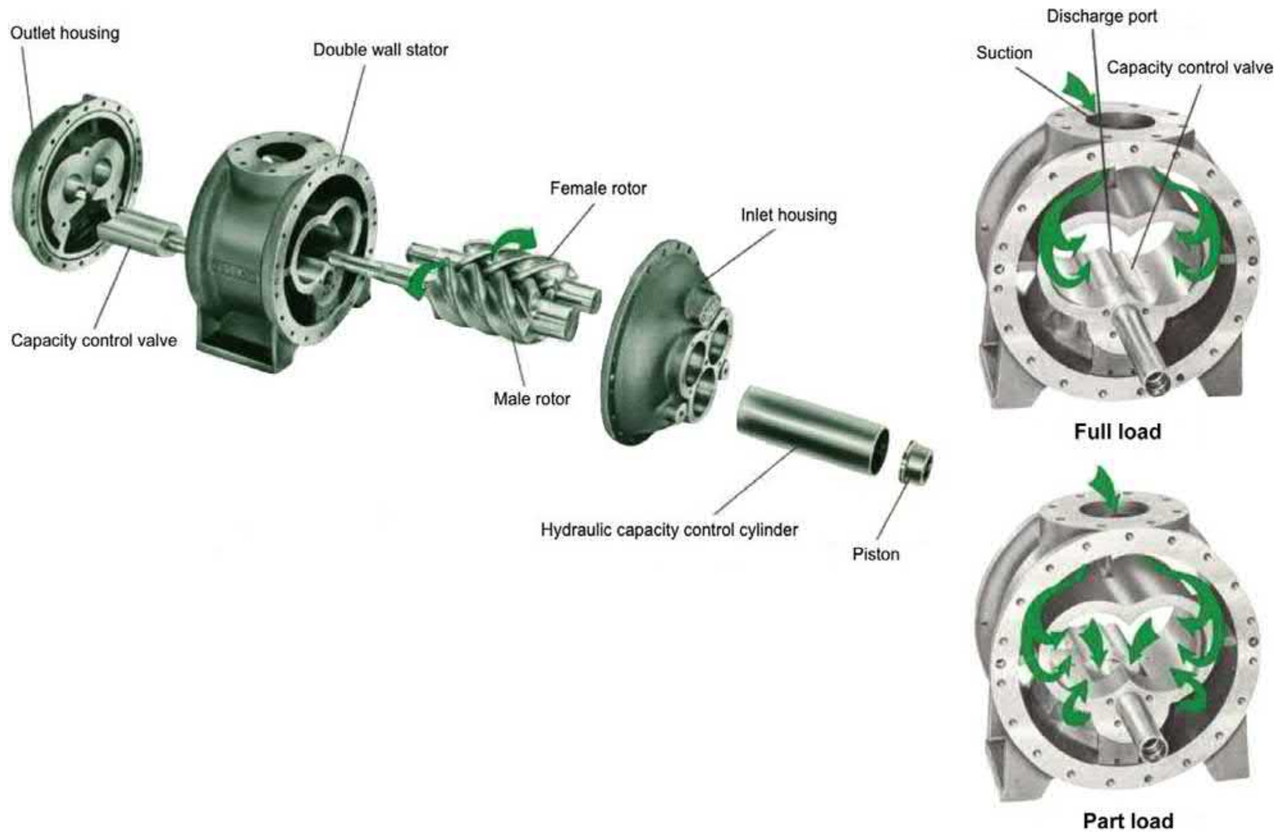


Fig. 11.5 Components of an oil-flooded screw-type compressor.

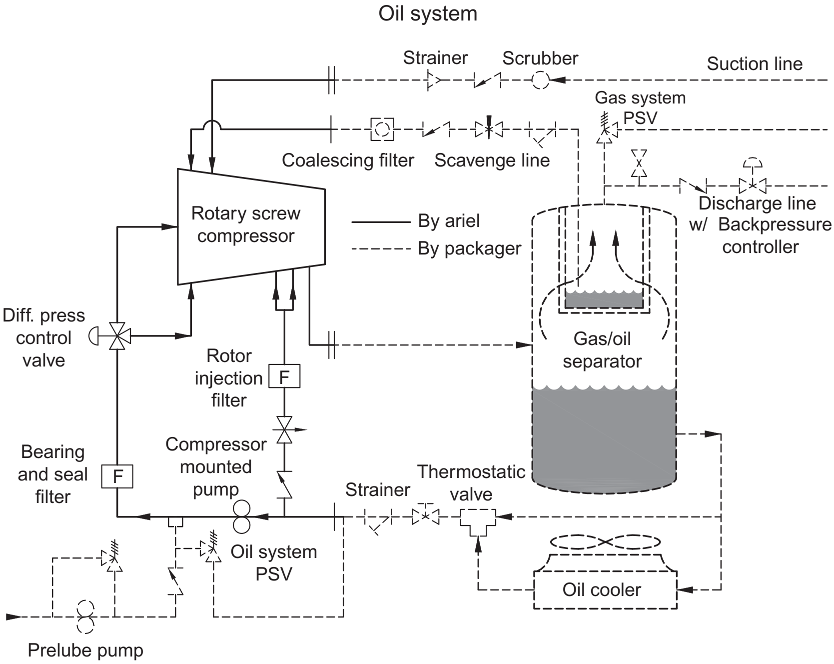


Fig. 11.6 Schematic of a typical oil system used in an oil-flooded screw-type compressor.

Table 11.1 Application ranges and selection of rotary compressors

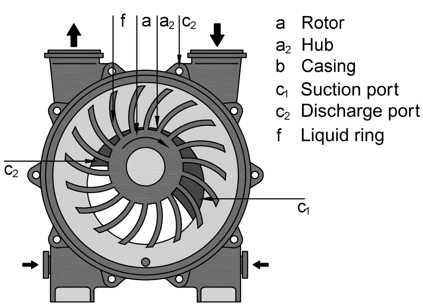
ICFM	Typical	300–2000
	High	30,000 (for low-pressure and vacuum service)
Discharge pressure (psig)	Typical	40–150
	High	550 usually attained in typical; second of two casings in tandem.
Discharge temperature (°F)	Typical	200–300
	High	450 (for some designs)
Pressure ratio (P_2/P_1)	Typical	2–3
	High	4 (20 is attainable with oil-flooded and liquid-ring machines)
Differential pressure (P_2/P_1), psi	Typical	10–75
Speed (RPM)	High	170
	Typical	300–3600
	High	20,000
BHP	Typical	50–2000
	High	6000

discharge temperature is substantially less than that indicated by adiabatic calculations. For example, the actual discharge temperature might be 200°F (93°C) or less versus a calculated value of 350°F (177°C). Screw compressors offer good efficiencies at low pressure ratios (somewhat lower than that of a reciprocating). Rotary compressors require inlet and discharge silences at higher power levels to achieve tolerable noise level. They are very good for skidding and semiportable installations due to small size and freedom from vibration. They are often two staged by connecting two casings in tandem arrangement that slows sidestream (in or out) between casings. They can have step-less capacity control down to 10%–15% of rated with hydraulically operated side vane. Efficiency loss at turndown is greater than that of reciprocating.

11.3 Liquid-ring rotary compressors

The Liquid-ring vacuum pump, shown in Fig. 11.7, is a unique type of rotary compressor in that it performs its compression by use of a liquid ring acting as a piston. The single rotor is located eccentrically inside a cylinder or stator. Extending from the rotor is a series of vanes oriented in a purely radial or radial with forward curved tips manner. Gas inlet and outlet passages are located on the rotor. A liquid partially fills the rotor and cylinder and orients itself in a ring-like manner as the rotor turns. The ring moves in an oscillatory manner. The center of the ring communicates with the inlet and outlet ports and forms a gas pocket. As the rotor turns and the pocket is moving away from the rotor, the gas enters through the inlet and fills the pocket. As the rotor turns, it carries the gas pocket with it. Additional turning takes the liquid ring from the maximum clearance area toward the minimum side. The ring seals off the inlet port and traps the pocket of gas. As the liquid ring is taken into the minimum clearance area, the pocket is compressed. When the ring uncovers the discharge port, compressed pocket of gas is discharged.

The efficiency of the liquid-ring compressor is ~50%, which is not good when compared to other rotary compressors. The capacity range is from 2 to 16,000 cfm. They are used in some wet services and in services with light to moderate vacuum.



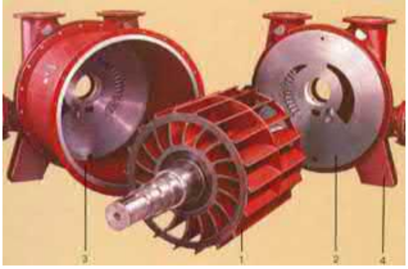


Fig. 11.7 Sectional of a liquid-ring vacuum pump rotary compressor.

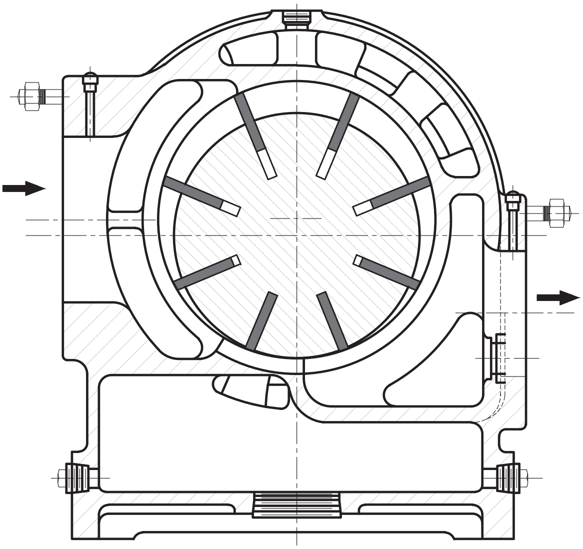


Fig. 11.8 Sectional of a sliding vane rotary compressor. Courtesy of GE Energy.

Typical services include vapor recovery, flare gas recovery, and the first-stage vacuum in crude units. They have a water effluent stream that may need to be treated.

11.4 Sliding vane compressors

The sliding vane compressor consists of a single rotor mounted eccentrically in a cylinder slightly larger than the rotor (refer to Fig. 11.8). The rotor has a series of radial slots that hold a set of vanes. The vanes are free to move in and out within the slots as the rotor revolves. Gas is trapped between a pair of vanes as the vanes cross the inlet port. Gas is moved and compressed circumferentially as the vane pair moves toward the discharge port. The compressor must have an external source of lubrication for the vanes.

The sliding vane compressor is an inexpensive, light-duty unit that is sometimes used in upstream producing fields and downstream petrochemical plants where heavier duty units are not justified. They are commonly used as a vacuum pump as well as a compressor, with a range between 50 and 6000 cfm. A single-stage compressor with atmospheric inlet pressure is limited to a 50 psi (345 kPa) discharge pressure. In booster service, they are used up to 400 psi (2758 kPa).

11.5 Straight-lobe compressors

The straight-lobe compressors are very similar to that of a helical-lobe compressor and use two intermeshing spiral lobes to compress gas between the lobes and the rotor chamber of the casing. The compression cycle begins as the open part of the

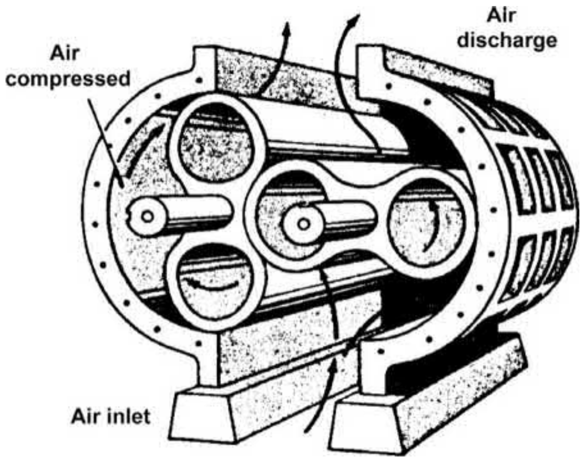


Fig. 11.9 Cutaway of a straight-lobe compressor.

straight-lobe form of the rotors passes over the inlet port and traps a quantity of gas. Gas is moved axially along the rotor to the discharge port where the gas is discharged into the discharge nozzle of the casing. Volume of the gas is decreased as it moves toward the outlet, with the port location controlling the pressure ratio. Fig. 11.9 shows a cutaway of a straight-lobe compressor.

The volume range of straight-lobe compressors is 5 to 30,000 cfm. Pressure ranges are somewhat limited, with the maximum for a single-stage rating at 15 psi (103 kPa). In two-stage units the discharge pressure is increased to 20 psi (138 kPa).

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Overview of commonly used drivers

12

12.1 General considerations

This chapter reviews the fundamentals and characteristics of commonly used drivers and power transmission devices such as gears and belts. It provides an overview of the various categories of drivers and discusses the choice of these drivers.

The information contained in this chapter will enable one to select the appropriate category of driver for an intended service.

12.2 Driver categories

12.2.1 Introduction

Drivers must be capable of developing the required horsepower and torque to drive various mechanical equipment such as pumps, compressors, fans, and generators. They must be operated at constant speed or over a range of speeds. The energy source can be electrical, gas, or liquid fuel or pressurized fluids such as steam.

This section covers the most common drivers used:

- Electric motors
- Internal combustion engines
- Combustion turbines
- Steam turbines
- Expansion turbines
- Gears and power transmission devices

12.2.2 Electric motors

Electrical drivers fall into three major types:

- AC induction motors
- AC synchronous motors
- DC motors (used less commonly)

All three types are constructed to several different industry standards, which may be modified by company-specific specifications. These various standards and specifications (NEMA, API, IEEE) are outlined in Section 12.2.3.

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00012-0

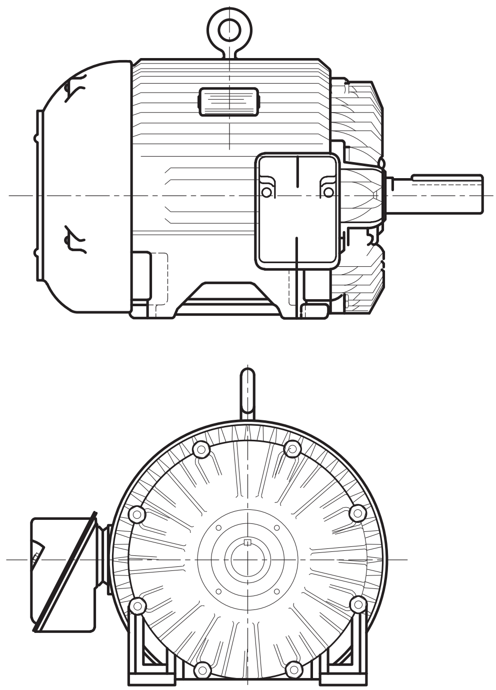


Fig. 12.1 Typical horizontal totally enclosed fan cooled (TEFC) small induction motor.
Courtesy of Siemens Energy and Automation, Inc.

12.2.2.1 Induction motors

The ruggedness and simplicity of the squirrel-cage induction motor usually makes it the first choice for a driver. Fig. 12.1 shows a typical small induction motor. These motors fill most general-purpose, constant-speed drive requirements. Induction motor types are as follows.

12.2.2.1.1 Fractional horsepower motors (universal motors)

These are single-phase motors, available in 50/60 Hz, with fixed nominal speeds of 3600, 1800, 1200, and 900 rpm. They are available in 115 and 230 V in 60 Hz and 110 and 220 V in 50 Hz.

NEMA MG 1-1.02 covers ½ and ¾ motors. Smaller motors are available, built to vendor specifications.

12.2.2.1.2 Single-phase motors

Single-phase motors are typically 1 to 10 hp, fixed or multispeed, ranging from 900 to 3600 rpm in 115 and 230 V, and 60 Hz. They are available in the same horsepower rating at 50 Hz with fixed speeds ranging from 3000 rpm down to 750 rpm.

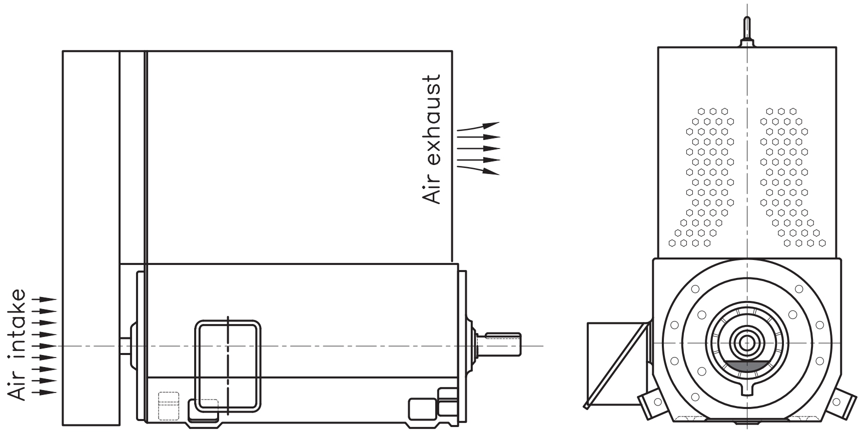


Fig. 12.2 Typical horizontal totally enclosed air-to-air cooled large motor. Courtesy of Siemens Energy and Automation, Inc.

12.2.2.1.3 Three-phase motors ranging from 1 to 250 hp

The most commonly used induction motor type is the three-phase motor ranging from 1 to 250 hp. They have speeds ranging from 900 to 3600 rpm in 60 Hz and from 750 to 3000 rpm in 50 Hz applications. Chemical Industry Severe Duty Motors are built to IEEE Standard 841. These motors are rated for 600 V and below.

12.2.2.1.4 Three-phase motors larger than 250 hp

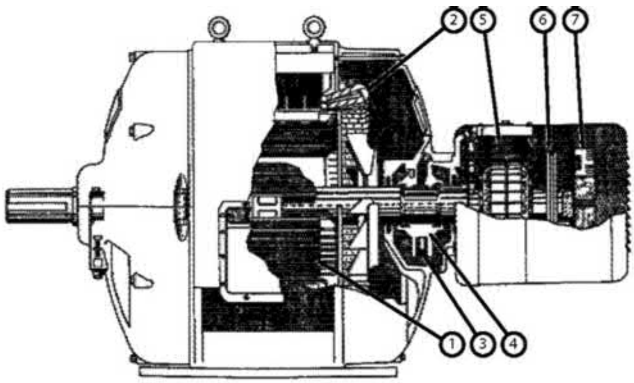
These larger three-phase motors are typically custom built in speeds of 3600, 1800, and 1200 rpm. Voltage ratings are typically 460, 2300, and 4000 V. Fig. 12.2 illustrates a typical, large horizontal motor. These large motors may be built to API 541 "Squirrel C age Induction Motors."

12.2.2.2 AC synchronous motors

Synchronous motors are generally selected to drive lower speed (300 to 1200 rpm) loads. Occasionally, a higher speed load is driven with a gear box between the motor and the load. Fig. 12.3 shows a typical small synchronous motor. The relative merits and selection criteria of synchronous and induction motors are discussed in Chapter 13.

12.2.2.3 DC motors

Occasionally, DC electric motors have been used in large sizes to allow adjustable speed. This is an expensive high-maintenance alternative to an AC motor with an adjustable frequency drive. DC motors are not covered in this text, except for a few minor references.



1. Field Coils
2. Stator Coils
3. Oil Ring, Front and Rear
4. Bearing, Front and Rear
5. Shunt Field Coils
6. Field Discharge Resistor
7. Synchronization Control Components

Fig. 12.3 Typical small synchronous motor.
Courtesy of Electrical Machinery-Dresser Rand.

12.2.3 Industry standards for motors and generators

Fig. 12.4 illustrates the relationship among the various industry standards for alternating (AC) induction motors and generators, AC synchronous motors and generators, and direct current (DC) motors. The "medium" induction motors (see Fig. 12.4 for a definition) are commonly called "NEMA frame" motors due to their standardized dimensions according to numerical frame sizes. Refer to Chapter 13 for figures that provide typical horsepower ratings and speeds available in NEMA frame sizes and for a list of NEMA frame sizes and dimensions. Large induction machines can either be purchased to "general purpose" or "special purpose" specifications, depending on the application.

12.2.4 Mechanical drivers

Mechanical drivers fall into four major types:

- Internal combustion engines (I/C)
- Combustion gas turbines (CGTs)
- Steam turbines
- Expansion turbines

12.2.4.1 Internal combustion engines (I/C)

These engines are used primary for driving pumps, generators, and reciprocating compressors. They are also sometimes used for driving large fans and blowers. To improve reliability, they are selected on a lower rpm range of 300 to 1800. For high-speed applications, use of a gear box is usually preferred.

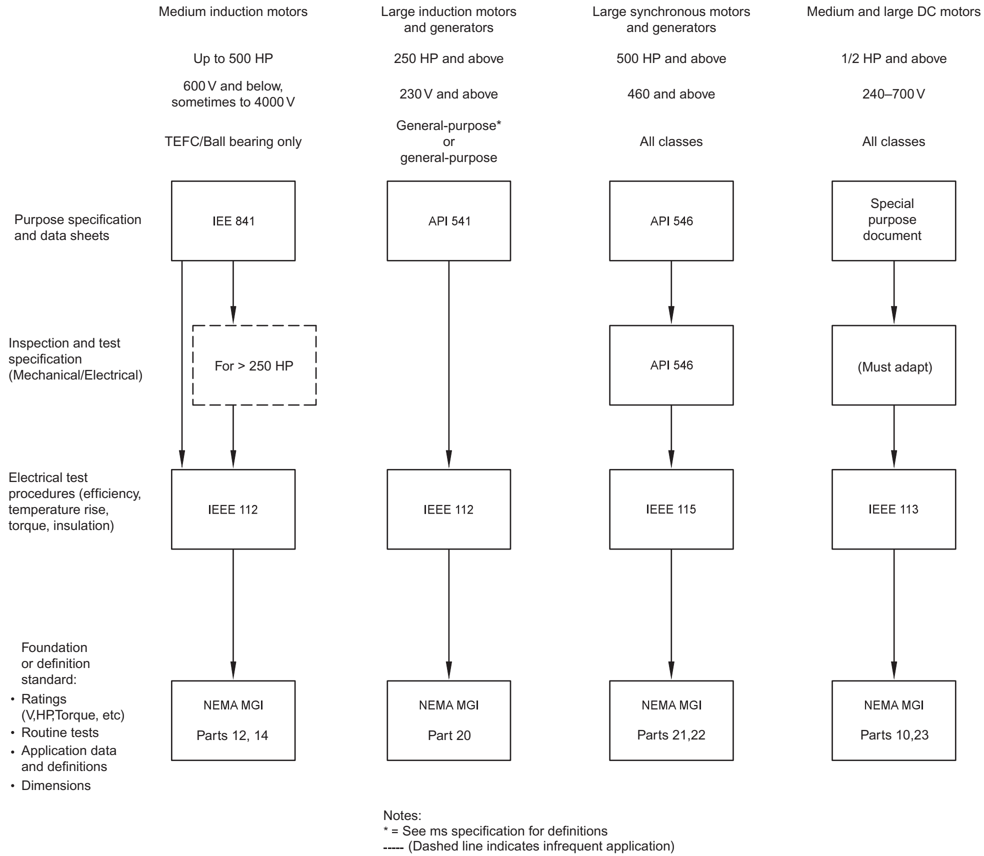


Fig. 12.4 Relationship of industry standards for motors and generators.

I/C engines are typically used when:

- Gaseous or liquid fuel is available at competitive cost
- The driven equipment is low-speed reciprocating
- Power range is low to moderate
- The efficiency of a gas turbine must be improved, and heat recovery is not possible
- Applications are remote, portable, and/or offshore

Engines are categorized by

- Ignition type (gas or diesel)
- Fuel burned
- Cylinder configuration

Most offshore platforms use I/C engines for driving standby generators and fire pumps. Occasionally, I/C engines on platforms drive main generators and gas compressors.

Several common engine categories are summarized in Fig. 12.5. Fig. 12.6 shows a typical I/C engine.

12.2.4.2 Combustion gas turbines (CGTs)

Combustion Gas Turbines are normally selected for driving large pumps, compressors, and generators. They are selected in horsepower ranges of 1000 to 270,000. Their efficiency can be improved by using waste heat recovery units. Turbines in the smaller horsepower of the earlier range have been used for driving crude oil pumps and vapor recovery compressors where electric power is not available. They may also

Diesel engine	An engine in which the fuel is ignited entirely by the heat of compression of the air supplied for combustion
Gas engine	An engine that operates on a combustible gas fuel (usually natural gas) and in which the gas is ignited by an electric spark
Oil diesel engine	This diesel engine operates on fuel-oil
Gas-diesel engine	This engine operates on a combustible gas as primary fuel; the ignition of the gas is aided by pilot-oil fuel injected after compression is practically completed. The gas fuel may be compressed in the engine cylinder with the air or compressed separately and injected into the combustion chamber near the end of the compression stroke
Dual-fuel diesel engine	This engine may be operated as an oil-diesel or a gas-diesel engine and is equipped with controls and parts to permit operation as one or the other
In-line engine	In this engine the centerlines of the cylinders are in a vertical plane containing the centerline of the crankshaft. The combustion, or working ends of the cylinder are at the top
Vee engine	In this engine, there are two banks of cylinders, each bank at opposite sides and at the same angle to the vertical plane containing the crankshaft centerline
Horizontally opposed engine	This engine has a horizontal cylinder, or bank of cylinders on each side of the crankshaft. If the pistons of two opposing cylinders are connected to the same crank pin, it is a special form of V-type engine having an included angle of 180 degrees

Fig. 12.5 Common internal combustion (I/C) types.

Fig. 12.6 Six-cylinder, four-stroke cycle turbocharged diesel engine.
Courtesy of Caterpillar, Inc.

be used where additional power from a utility company may be expensive to get to the site.

On offshore platforms, gas turbines generally drive the main electrical generators in the 1000 to 45,000 hp range. Waste heat recovery units are used whenever heat can be used for process fluids. Steam is typically not generated economically on platforms.

CGTs used in the upstream and downstream petrochemical industry are divided into four basic types of construction:

- Industrial gas turbines
- Aero-derivative gas turbines
- Small industrial gas turbines
- Hybrid gas turbines

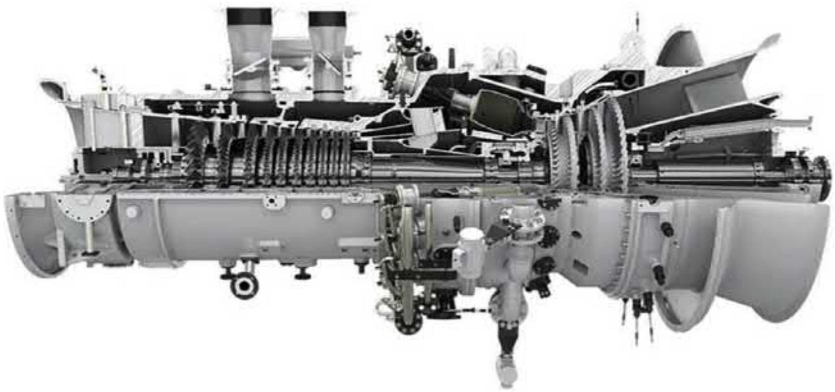


Fig. 12.7 Industrial-type gas turbine.
Courtesy of Westinghouse, Energy Services Division.

12.2.4.2.1 Industrial gas turbines

Heavy-duty industrial CGTs were developed with long-term reliability as the prime goal. They run at lower firing temperatures and lower speeds and can operate on a broad range of fuels. Models are available from about 10,000 to 270,000 hp. On a purchase cost per output basis, modern industrial CGTs are less expensive than comparable to aero-derivative types.

The industrial turbine (refer to Fig. 12.7) is usually built into a heavy cast casing with a horizontal split. It is designed for in-place maintenance. Major overhauls occur about every three years and take 3 to 4 weeks.

Industrial turbines for refinery services are covered in API 616.

12.2.4.2.2 Aero-derivative gas turbines

Aero-derivative Gas Turbines were originally designed for flight service. With derating and minor modifications, they have gained growing acceptance in several industrial and marine applications. Models are normally available from about 4000 to 45,000 hp. Smaller CGTs derived from turbo-prop service are also available but used infrequently. Small units (<1000 hp) are used primarily in producing field applications.

Aero-derivatives are built for small space requirements, high power-to-weight ratios, and high efficiency. They are built in radially split sections which are suspension mounted and can be rapidly removed. Normally the units must be shipped to approved repair facilities to be overhauled.

By rerating the continuous firing temperature and the consequential turbine rating, aero-derivatives have proven reliable enough to run up to two years without overhaul. Also, the maintenance cost is up to twice as expensive as a comparable industrial model. Repair shops need about 2 months to complete a rebuild.

12.2.4.2.3 Small industrial gas turbines

Small industrial gas turbines use compact, radially split designs like aero-derivatives, but employ firing temperatures and bearings that are more typical of industrial units. They are generally used on offshore platforms, pipelines, and other remote locations where they provide a cost-effective compromise. Units are from 1000 to 20,000 hp.

12.2.4.2.4 Hybrid gas turbines

Hybrid gas turbines use the gas generating sections of an aero-derivative CGT. These include the compressor section, combustors, and the turbine stages needed to drive the compressor. The gas generator is attached to an industrial type power turbine that converts the remaining gas energy to power on a second shaft.

12.2.4.3 Steam turbines

Steam turbines are used for driving generators, compressors, and large pumps. Selection of turbines depends upon the load to be driven and the amount and condition of steam available after meeting process requirements. Steam-driven turbines are often used as standby units for driving pumps, compressors, and generators. Sometimes they are used as main drivers for balancing steam in the plant.

Steam turbines can be condensing or noncondensing. Some machines have steam extraction or admission sidestreams. Noncondensing turbines are also known as backpressure or topping turbines.

A condensing turbine may be selected when the process requirements result in an excess of low-pressure steam. It is also selected when no high-pressure steam is available.

A noncondensing turbine, also known as a backpressure turbine, is selected when process steam demands are compatible with the amount of low-pressure steam vented from the exhaust. The noncondensing turbine is most frequently selected because of

- Lower capital cost
- Simple construction
- Suitability for higher speeds
- Generally more reliable
- Less complex machinery and less support systems required

Advantages of condensing turbines:

- Require less steam because the energy drop of the steam is larger
- Only one steam level is affected for a change in power requirements
- Moderately efficient use of medium to low grade energy

Disadvantages of condensing turbines:

- Require high specific volumes of steam
- Cost extra for a condenser and other auxiliary equipment
- Have lower reliability because they use blades, increasing the risk of blade failure
- Greater blade erosion due to operation with wet steam

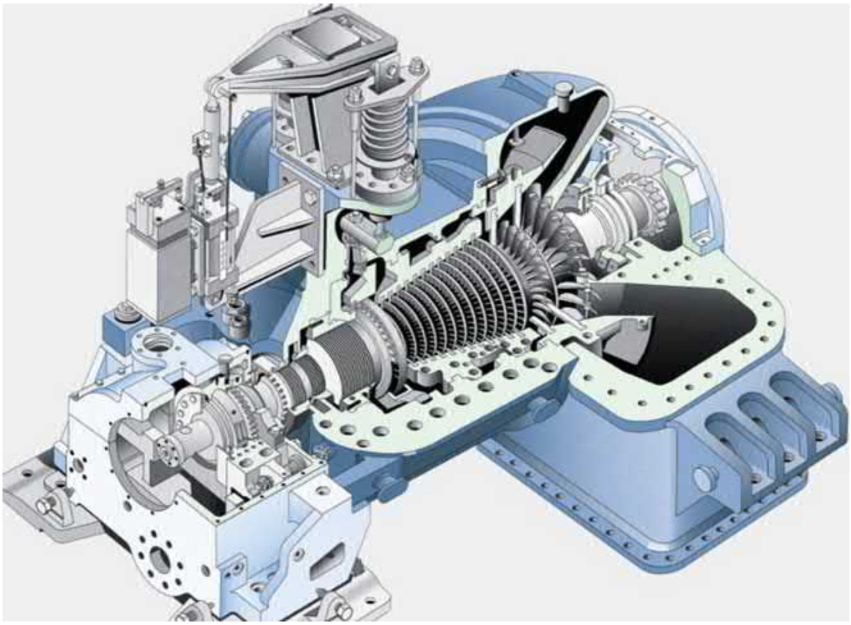


Fig. 12.8 Typical multistage steam turbine.
Courtesy of Siemens energy and Automation, Inc.

Ratings vary from a few horsepower for a single-stage, general-purpose turbine, to 50,000 hp or more for special-purpose, multivalve, multistage turbines. Fig. 12.8 shows a typical multistage steam turbine.

General-purpose steam turbines for refinery services are covered in API 611. Special-purpose turbines are covered in API 612.

12.2.4.4 Expansion turbines

An expansion turbine converts gas or vapor into mechanical work as the gas or vapor expands through the turbine. There are two types of expansion turbines: axial flow and radial flow.

Axial-flow expansion turbines are similar to conventional steam turbines. They may be single stage or multistage with impulse or reaction blading, or some combination of the two. Turbines of this type are "power recovery turbines." They are used where flow rates, inlet temperatures, or total energy drops are very high.

Radial-flow expansion turbines are normally single stage, resembling a centrifugal semiopen compressor impeller. Radial-flow expansion turbines are used primarily for low volume service and are frequently used in hydraulic systems, mainly for power recovery.

A turbo expander is a machine with an expander on one end of the shaft and a compressor on the other end, all as one unit. The expander drives the compressor.

12.2.5 Power transmission devices

Power transmission devices are an integral part of many drive trains, although they are not "prime movers," as are previous drivers mentioned. Common power transmission devices include

- Gear drives
- Belt drives
- Clutches (not covered)

12.2.5.1 Gear drives

Gear drives can be classified by shaft arrangement, type of gearing, ratio change, and horsepower. Three broad types are as follows:

1. *Enclosed gear unit*: The driver is mounted on the same structure.
2. *Gear motor*: A combination of an enclosed gear unit and a motor, with the frame of one component supporting the other, and with the motor shaft common with or directly coupled to the input shaft of the unit.
3. *Shaft mounted speed changer*: An enclosed gear unit mounted directly on and supported by the input shaft of the driven machine, with a reaction member to prevent rotation of the housing. The gear unit may or may not support the driver.

Fig. 12.9 shows a classification of available drives by shaft arrangement, types of gearing, ratio range, and horsepower range.

Most gear drives are manufactured per standards and practices developed by the American Gear Association (AGMA). Industry Standard API 613 covers large, special-purpose gear drives. Small gears are covered in various AGMA standards.

12.2.5.2 Belt drives

Belt drives are used in low-horsepower applications as an alternative to gears. Though less than gears, they offer benefits of low cost and flexibility. They are covered in API 1B, "Oil Field V-belts."

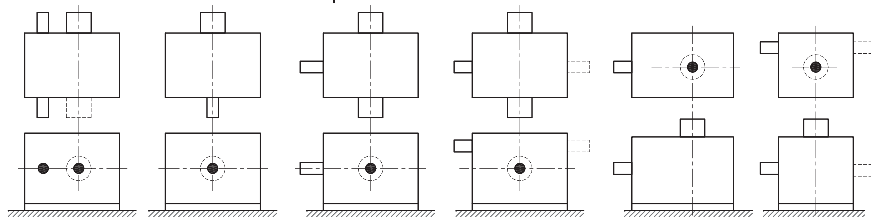
12.3 Application and selection criteria: Driver categories

12.3.1 Overview

Sections 14.3 and 14.4 contain background information to help you select the best category of driver (motor, steam turbine, gas turbine, I/C engine). Once the category is selected, one should refer to the respective chapter of this manual. For example, if a motor is the category chosen, then one can refer to Chapter 13 to determine whether it should be induction or synchronous motor, voltage rating, enclose type, and so on. Note that motors are the most common selection due to their generally lower capital and operating cost, simplicity, and relative reliability.

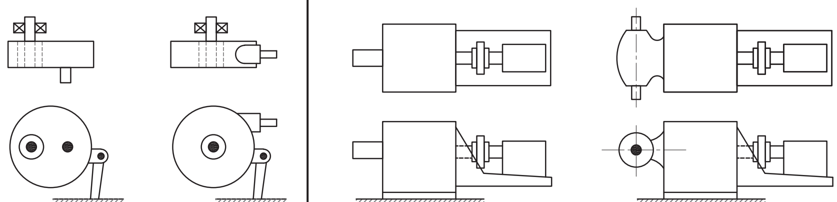
Common speed reducer mounting arrangements

Base-mounted speed reducers and increasers



Shaft arrangement	Horizontal parallel offset	Horizontal concentric	Horizontal right angle	Horizontal right angle	Vertical right angle	Vertical right angle
Gearing	Helical or spur	Helical or spur	Bevel or bevel and helical	Worm	Bevel or bevel and helical	Worm
Ratio range	1.5 to 300:1	3 to 70.1	1.5 to 500:1	3 to 10.000:1	1.5 to 500:1	3 to 10.000:1
Hp range	5 to 3000	5 to 400	5 to 2000	3 to 400	5 to 2000	3 to 400

Shaft-mounted speed reducer / Gear motors



	Shaft-mounted speed reducer		Gear motors	
Shaft arrangements	Horizontal parallel offset	Horizontal right angle	Horizontal concentric	Horizontal right angle
Gearing	Helical or spur	Worm	Helical or spur	Bevel or bevel and helical
Ratio angle	4 to 25:1	5 to 70:1	1 to 300:1	1.5 to 500:1
Hp range	1 to 250	Fractional to 10	1 to 125	1 to 125

Fig. 12.9 Types of gear drives.
Courtesy of Power Transmission design Handbook, 1995, Penton Publishing.

12.3.2 General selection process

The text and figures that follow should enable one to select the best driver for most common applications. For complex or unusual applications, one should consult a mechanical equipment specialist.

The following is a general approach to selecting a driver category for a given application:

1. Assemble all the applicable technical data on the driven load, such as:
   - Horsepower (design and expected range of operation)
   - RPM (design and expected range of operation, if not constant speed)
   - Size and weight
   - Start-up inertia
2. Review the selection criteria outlined in Section 14.3.3, compare these to one's application and make a decision based on one's local priorities. Section 14.3.4 gives typical applications, which one can use to compare to the specific application.

12.3.3 Summary of selection criteria

Fig. 12.10 is a summary of the specific selection criteria for the most common categories of drivers. Since the priorities of each criterion vary, and often conflict, one will need to review each in the context of the specific application. The energy available and the type of driven equipment will generally dictate the driver selection. If there are multiple sources available, one must consider the economics and type of equipment being driven.

Furthermore, Fig. 12.11 summarizes four general areas of consideration to assist in the selection process:

- Economics
- Reliability
- Operability
- Operating center capabilities and restrictions

The relative importance of these factors will vary. For example, reliability may be the most important factor at remote or offshore facilities. Similarly, the most important factors for unspared critical drives may be reliability and operability. Plot area and weight are more important on offshore applications.

12.3.4 Economics

A recommended economic evaluation combines capital and operating costs over a specific evaluation period to determine the low cost alternative. Capital cost is the initial investment required; operating costs primarily include energy and maintenance costs. *Note that in most applications, the energy costs are the major portion of the overall Life-Cycle Cost (LCC).* However, where electricity or fuel has little value, such as offshore, energy cost may not be a major contribution to operating expense.

Fig. 12.12 shows simplified cost comparisons between driver selections, without consideration of the time value of money. This chart is included to show typical

Criteria	Local priority			Driver type																	
				Electric motors						Steam turbines						Internal comb (I/C)					
				Induction			Synchronous			Non-condensing			Condensing			Gas turbines			Recip. engines		
	CRIT	IMP	Not IMP	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor	Good	Fair	Poor
Reliability				X			X			X			X				X				X
Efficiency				X			X				X			X				X		X	
Noise					X		X				X			X				X			X
Variable speed capability						X		X		X			X			X				X	
Req'd oper. speed 1200 RPM					X		X					IMPR			IMPR			IMPR	X		
Req'd oper. speed 1200 to 3600 RPM				X				X			X				X			IMPR		X	
Req'd oper. speed 3600 RPM					X			X		X			X			X					X
Plot area required				X				X		X				X			X				X
Shaking forces transmitted to surroundings				X			X			X			X			X					X
Foundation size				X				X		X				X			X				X
Horsepower to weight capability					X			X		X				X		X					X
Startup simplicity				X			X			X				X			X			X	
Sidestream inlet or exhaust cap.						IMPR			IMPR	X			X					IMPR			IMPR
Mult. fuel or energy source cap.						IMPR			IMPR			IMPR			IMPR	X				X	
Complexity of basic machine				X				X			X			X				X			X
Complexity of control systems				X				X			X				X			X		X	
Complexity of lube-oil systems				X			X				X			X				X		X	
Complexity of cooling systems				X			X				X			X				X		X	
Complexity of piping design				X			X				X			X				X		X	
Maintenance costs				X				X		X				X				X			X
Purchase price below 100 HP				X					IMPR		X				IMPR			IMPR		X	
Purchase price below 500 HP				X				X		X					IMPR			IMPR		X	
Purchase price below 2000 HP				X			X			X					X			X		X	
Purchase price above 2000 HP				X			X			X				X				X			X

Notes: CRIT = Critical
IMP = Important
IMPR = Impractical
Not IMP = Not important

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Fig. 12.10 Commonly used criteria for selecting driver categories.

Economics	This is the evaluation of the purchase, installation, and operating costs of different driver options.
Reliability	The degree of dependability required of the driver may be determined by some of the following factors: • Service criticality • Availability of reliable fuel and power • Maintenance frequency requirements
Operability	The features of different drivers which make them more or less suitable for the specific application. These factors may include: • Driver operating speed • Speed control capability • Need for remote starting and stopping • Availability of skilled operating and maintenance personnel • Equipment sparing philosophy
Operating center capabilities and restrictions	The impact different driver options have on the facility utility balance, loading, and maintenance functions. These factors may include: • Energy sources • Facility energy balance • Future energy needs • Cooling water availability • Space limitations • Environmental restrictions • Operation and maintenance preferences

Fig. 12.11 Summary of general selection criteria for driver categories.

order-of-magnitude numbers only. Do not use it to select specific driver applications or as a substitute for a LLC analysis, which is the appropriate method to select a driver.

12.3.4.1 Capital costs

Capital cost is the total initial investment required for each driver option. The investment includes the equipment purchase and various installation costs. Note that installation costs are usually significantly larger than the purchase price.

12.3.4.1.1 Equipment purchase price

Equipment price can be obtained from equipment manufacturers. However, one should be cautious of the data provided by local salesmen. The cost of equipment can vary considerably with specification requirements and local salesmen may not be sensitive to the impact of specifications.

Some common examples of specification impact on price include:

Driver horsepower	50	250	500	1000	2000	5000	10,000
Driver type							
Electric motors							
Purchase price	3400	22,000	30,000	42,000	65,000	100,000	250,000
Energy cost	65,700	316,000	630,000	1,250,000	2,470,000	6,000,000	12,000,000
Total cost motors	89,100	338,000	660,000	1,292,000	2,535,000	6,100,000	12,250,000
Steam turbines							
Purchase price	20,000	29,000	30,000	130,000	310,000	490.000	850,000
Steam rate (1bs/HP.hr)	70	50	50	20	20	15	15
Energy cost	55,200	213,000	402,000	2.100,000	3,800,000	8,300,000	15,450.000
Total cost steam turbines	75,200	242,000	432,000	2,230,000	4,110,000	8,790,000	16,300.000
Reciprocating engines							
Purchase price	27,500	50,000	1 12,000	246,000	480,000	2,300,000	4,000,000
Fuel rate (BTU/HP»hr)	10,000	8000	7750	7200	7100	6300	6700
Energy cost	39,400	157,700	305,500	568,000	1,120,000	2,500.000	5,300.000
Total cost recip. Engines	66,900	207,700	417,500	814,000	1,600,000	4,800,000	9,300,000
Gas turbines							
Purchase price		-IMPRACTICAL-		550,000	650,000	1,250,000	2,750,000
FUEL RATE (BTU/HP.hr)				10,700	9500	9050	7700
Energy cost				850,000	1,500,000	3,600.000	6,100.000
Total cost gas turbines	0	0	0	1.400,000	2,150,000	4,850,000	8,850,000

Note: Totals do not include steam/fuel rates.

Assumptions:

1. This figure does not consider the time value of money for capital or operating cost.
2. Energy cost is based on a four year period and a 90% operating factor.
3. Electric motors are 460 V up to 250 hp, 2300 V to 2000 hp, and 4000 V above 2000 hp.

Motor enclosure is TEFC for 2000 hp and below and WP2 above 2000 hp.
All motors are horizontal. Motor speed is 3600 RPM for 500 HP and below.
Motor speed is 1800 RPM above 500 hp.

4. Cost of electric power is 0.05 $/kW-h
5. Steam conditions are 150 psig inlet to 40 psig outlet for 500 hp and below.

Steam conditions are 600 psig inlet to 40 psig outlet above 500 hp.
Turbine speed is 3600 rpm for 1000 hp and below, and API 611 applies.
Turbine speed is 5000 rpm above 1000 hp, and API 612 applies.
Turbines above 1000 hp shall have NEMA D governors.
Turbines 1000 HP and below shall have NEMA A governors.
Turbine selections are based on the maximum possible efficiency.

6. The value of 150-to-40 psig steam is 0.50 $/1000 lbs.
7. Costs for reciprocating engines are for separate engines, not for integral compressor engines. When selecting a compressor, integral engines may also be considered.
8. The cost of gas is $2.50/million btu.
9. Typical efficiencies are assumed for each driver type and size.

Fig. 12.12 Typical simplified driver cost comparisons (1990).

12.3.4.1.1.1 Electric motors Motors are built as general and special purpose in distinct horsepower sizes. Since driven equipment performance may not be known, the motor rating may change during the evaluation of driven equipment.

Company specifications and good practice require a margin between the motor capability and driven equipment that include process variations.

The ambient conditions for the driver location will impact the motor enclosure required. Depending on the power rating of the driver, the enclosure may be a special design to comply with ambient area classification or site conditions specified.

The speed of the motor estimated may not be the speed selected. Field experience is mixed for motors rated 600 hp and larger operating at 3600 rpm. A substantial number of these motors have had rotors with "thermal instability" or a tendency to change mechanical balance state as they heat up. These problems can be eliminated by imposing vibration limits and verified by rated temperature testing of the motor at the manufacturer's plant. Some manufacturers have a good record producing 3600 rpm motors, but occasionally a rotor is made which demonstrated thermal instability and must be remanufactured. Enough lead time must be allowed in the equipment procurement or project schedule to permit a new rotor to be made, typically 3 to 4 months. Motors with rated speeds of 1800 rpm and below usually have very few mechanical problems. Selecting a slow speed (1800 rpm) motor and gear box may be more reliable but also more expensive than a high-horsepower (3600 rpm) motor.

When retrofitting or adding an electric motor to an existing system, it is important to consider the transformer rating and the cost of substation upgrades, if required.

12.3.4.1.1.2 Steam turbines The steam rate of the turbine will impact the initial cost. Manufacturers usually quote two options, low capital cost with lower efficiency or low steam rate with higher efficiency. The low capital cost selection will have fewer stages and/or smaller diameter wheels resulting in higher steam use. The low-steam-rate selection will have more stages and/or larger diameter wheels. For spare unit drives, low capital cost turbines are usually the first choice since they are rarely required to operate. Turbines selected for continuous duty are almost always the lowest steam rate choice due to the savings in steam usage.

12.3.4.1.1.3 Reciprocating engines The speed of the engine estimated may not be the speed selected. Since engine power capability increases with the speed, vendors will want to provide the highest speed option available. Engine reliability decreases with increasing speed. Speed limits may affect the engine selection.

Most specifications require a 10% to 20% margin between the driven equipment horsepower requirement and the driver capability. Vendors may overlook this requirement when estimating the driver.

Some installations may require reduced NO_x emissions and a special "clean burn" engine design may be required. This design will be more expensive and may reduce engine performance.

12.3.4.1.1.4 Gas turbines Variations in atmospheric conditions (e.g., high temperature, high humidity, and high elevation) will have a major impact on the gas turbine power output. The difference in engine performance for the most favorable site conditions and the most severe site conditions may result in the selection of a larger engine.

Most mechanical drive specifications require a 10% margin between the driven equipment horsepower requirement and the driver capability. Vendors may overlook this requirement when estimating the driver.

12.3.4.2 Installation costs

Installation costs include foundation requirements, piping, electrical wiring, fuel piping, instrument and control interface, field mechanical completion, testing and preventive maintenance required prior to start-up. It is not unusual for the installation cost to be between two and three times the equipment purchase price.

Installation cost estimates for different driver options must be generated separately for each driver option. Many companies and industry organizations have factors for installation of drivers and other equipment in new plants. These estimating factors do not apply to installations in existing facilities.

An indirect installation cost is the cost to modify or expand existing utility systems to support the new driver and driven equipment. This is common for electric motors or steam turbines. For example, to provide electrical power to a new electric motor in a system that is already utilized to its maximum, a new or expanded electrical distribution system will need to be installed. Since there will be other users of the new or expanded electrical system, only a portion of the cost should be assigned to the selected electric motor.

12.3.4.3 Operating expenses

Operating expense consists of maintenance and energy costs. Maintenance will vary depending on the location of the facility and can be estimated from past experience of the facility. For example, offshore maintenance is much more costly than onshore; and maintenance at remote onshore facilities is more costly than for centrally located facilities. The availability of skilled labor, transportation cost, spare parts cost, and the type of equipment selected have a major impact on maintenance costs.

The frequency of driver maintenance from most reliable to least reliable are

- Electric motors
- Steam turbines
- Gas turbines
- Reciprocating engines

12.3.4.4 Utility costs

The utility cost (liquid or gas fuel, electric power or steam) can be estimated based on the power requirements quoted by the driven equipment manufacturer combined with the energy consumption of the driver.

Common expressions of units for driver energy consumption are

- Gas turbine fuel (btu/hp-h)
- Steam (lbs/hp-h)
- Electric power (kW-h)

- Reciprocating (btu/hp-h)
- Engine Fuel [gal/hp-h (assuming a heat value of fuel, btu/gal)]

It is important to include an operating factor in any calculation of utility cost. Operating factor is the percentage of time the driver will operate. For primary drivers this is usually between 90% and 95% and standby drivers may be as low as 5% to 10%.

12.3.4.5 Economic analysis methods

Once investment and operating costs have been determined, an economic comparison of two or more drivers can be done. Two methods commonly used are

- Comparative cost method
- Rate-of-return analysis

12.3.4.6 Comparative cost method

The comparative cost method is simple and useful for making rough driver comparisons by taking the total of capital cost and operating cost for each option over a specified period. It gives "order-of-magnitude" comparisons but should not be used for final selections.

The following example illustrates this method. The evaluation period is usually a minimum of four years but could be as high as the project life.

	Driver A	Driver B
Capital cost		
Equipment cost	$2500	$3500
Installation cost	*$7500*	*$7500*
Total investment	$10,000	$11,000
Operating cost/year	*$1500*	*$1300*
Ten year total cost	$25,000	$24,000

The comparative cost method shows "Driver B" as the best choice based on simple addition of the costs. This simplified approach ignores factors such as depreciation, taxes, and the time value of money. These are very important considerations, and therefore, this simple method should only be used for order-of-magnitude estimates.

12.3.4.7 Rate-of-return analysis

The rate-of return analysis treats the additional capital of the more expensive driver as an investment and any savings in operating cost as a revenue. With this data, a rate of return for the more expensive driver can be calculated. If the less expensive investment also has lower operating costs, then no analysis is necessary.

In the aforementioned example, an additional $1000 for Driver B will save $200 per year in operating cost. This is the same as a nondiscounted 20% before-tax return on the additional investment. If the Company can earn greater than 20% on its investments, then Driver A is the best choice, if not, then Driver B is the best choice.

In general, the appropriate analysis is to compute the net present worth of each option, and the rate of return as described before. These numbers should allow one to make a choice of the most economic driver. Describing these computations is beyond the scope of this text.

12.3.5 Reliability

Reliability is the degree of dependability required of a driver, varying from nonessential to critical. The following factors should be considered:

- Service criticality
- Availability of reliable fuel and power
- Relative equipment reliability

12.3.5.1 Service criticality

One should complete a design review of critical machinery to minimize the probability of mechanical failure. Equipment selected for "critical service" is generally defined as that equipment whose unscheduled shutdown (due to mechanical failure or loss of the energy source) will result in one of the following:

- Endanger the safety of facility personnel
- Cause release of toxic material into the environment
- Cause extensive damage to major equipment
- Produce an excessive loss of revenue through an unplanned shutdown or curtailed operation on the facility
- Produce an excessive loss of revenue through extended loss or degradation

12.3.5.2 Availability of reliable fuel and power

Energy source reliability is largely dependent upon who controls the energy production. For example, company steam systems are usually more reliable than utility-supplied electrical power because

- Steam systems are usually controlled by the operating center while purchased power electrical systems may be vulnerable to external failures.
- Electrical system voltage dips can cause motors to stall and shutdown. Steam turbines will usually continue to run at reduced speed with sagging steam system pressure.

Steam is normally not available offshore or at remote upstream locations. The most reliable source of power at these locations is usually multiple company-operated gas turbine-driven generators or reciprocating engine generators.

Purchased power is sometimes augmented with in-house power generating facilities which allow the use of electric motors as critical service drivers. Since in-house power generation is usually limited, the number of drivers connected to this system is also limited.

The best guide for energy source reliability is an accurate history of failures in the existing energy system, if available.

Electric motors	Induction motors are very reliable For maximum reliability, induction motors should be limited to 1800 rpm when the nameplate horsepower is 600 hp or above. The company has a history of poor performance from large (≥600 hp) 3600 rpm (2-pole) motors. Reliability of 3600 rpm motors can be increased by selective supplier choice and factory testing Slow speed induction and synchronous motors are very reliable
Steam turbines	Steam turbines are the next most reliable driver after motors Depending on steam conditions, single-stage steam turbines can reliably produce up to 800 to 1000 hp Multistage steam turbines are capable of thousands of horsepower, but compared to electric motors the degree of mechanical complexity decreases their reliability
Gas turbines	Gas turbines are used more for generator drives on co-generation projects and for upstream power generation Due to inlet air quality, gas turbines require regular offline washing Subject to periodic inspections. When required, overhauls must be scheduled
Internal combustion engines (I/C)	I/C engines are considered the least reliable of the four major driver types, requiring frequent shutdowns for maintenance They can burn some produced gases and/or crude oils and provide nonelectrical driver capability at horsepower ranges lower than gas turbines

Fig. 12.13 General equipment reliability comparisons of driver categories.

12.3.5.3 Relative equipment reliability

Different drivers have generally different reliability characteristics. These are summarized in Fig. 12.13.

12.3.6 Operability

Some of the factors that affect a driver's operability are as follows:

- Driver operating speed
- Speed control
- Remote starting and stopping
- Availability of skilled operating and maintenance personnel
- Sparing philosophy

12.3.6.1 Driver operating speed

The first operability consideration is to determine the speed required of the driver to match the driven equipment operating speed. When possible:

- Use a driver that operates at the same speed as the driven equipment
- Avoid the use of gears, belts, or other speed changing equipment
- Minimize the number of components in a train to maximize reliability

12.3.6.2 Speed control

The process conditions for some applications require flow and pressure variations. These variations can be achieved by throttling, bypassing, and speed control. Throttling and bypassing in high energy applications are expensive. The ability to control the driver speed to meet changing process conditions usually results in substantial energy savings. Some common application guidelines for adjustable speed drivers are as follows:

- Drivers for large centrifugal compressors commonly have adjustable speed control.
- Steam turbines, reciprocating engines, and gas turbines are frequently the most used drivers for adjustable speed operation.
- Occasionally, adjustable speed drivers are used on positive displacement pumps.
- Electric motors are usually installed as constant speed drivers but are readily available with adjustable speed when needed.
- Electric motor options for adjustable speed include adjustable frequency, wound rotor, and DC motors. However, the first cost of adjustable speed motors is significantly higher than constant speed motors. This cost can usually be recovered by reduced operating costs where speed control is needed.

One should be sure to consider the impact of adjustable speed operation on the driver and driven equipment. Speed changes from 30% to 5% of rated speed are not unusual. One should evaluate the effects of speed changes on critical speeds, structural resonant frequencies, pressure ratings, and mechanical ratings to avoid operation and maintenance issues.

12.3.6.3 Remote starting and stopping

The ease of starting and stopping drivers is only important when choosing spare drivers, and in particular, critical sparing applications. In these cases, the driver must start up quickly for safety or to minimize process upsets.

Generally accepted remote starting and stopping guidelines for drivers include

- Electric motors are the easiest drivers to start up and shut down. Both manual and automatic starting can be used. The manufacturer's recommendation on the frequency of restarts must be followed to avoid overheating the rotor.
- Noncondensing steam turbines are the next easiest to start. These backpressure turbines are easily manually started and can normally be automatically started. Regular maintenance is required on their automatic start system to ensure proper operation.
- Gas turbines and reciprocating engines usually start up well, depending on the quality of the starting system. Reciprocating engines are commonly used as standby fire pump and emergency generator drivers. Due to the initial cost of gas turbine drivers, they are not commonly used as drivers for spare machinery except for isolated installations such as offshore installations.
- Condensing turbines are usually the most difficult to start up due to the amount of equipment involved. A condensate pump and all associated instrumentation must be checked prior to starting. Condensing turbines are not used for automatic start applications as they are only suitable for continuous duty.

12.3.6.4 Availability of skilled operating and maintenance personnel

The skills and experience of the operation and maintenance personnel should be considered when selecting a driver. Selecting equipment similar to that already installed in the facility will ease the transition and acceptance by operation and maintenance personnel. This should be discussed with the appropriate operations and maintenance representatives, who usually have important input to the decision.

12.3.6.5 Availability of spare parts

The selection of a driver is sometimes determined by the availability of spare parts which is common with drivers already installed in the facility. This reduces the number of parts required in storage and improves the operation and maintenance consistency. If an identical driver is not selected, a driver from the same manufacturer may be preferred by operation and maintenance personnel because of the quality of service provided.

12.3.6.6 Sparing philosophy

Generally, low-cost equipment, such as pumps, installed in critical services are spared. The cost of spare equipment is much less than the potential loss of product.

Depending on the utility sources available at the facility, a sparing philosophy may be established. For example, many refineries utilize a combination of steam turbines and electric motors as spare drivers. Company-generated power sometimes supplements purchased power and is dedicated to critical spare electric motor drives. Steam is usually available for spare drivers, even during temporary electrical power failures. If both the main driver and the spare driver are electric motors, the two motors should be supplied from independent electrical circuits.

The sparing philosophy of a facility may change if a major expansion or upgrade is planned. The availability of electrical power, cooling water steam, and liquid or gas fuel suitable for burning in gas turbines or reciprocating engines should be evaluated for the expansion, plant flexibility, and future needs.

Assuming facility turndown is acceptable during compressor outage, large, very expensive, reciprocating compressors and their drivers are not usually 100% spared. Instead, due to the flexibility in loading of the compressors, two 60% or 70% units are installed. Otherwise, alternative arrangements such as two 100% or three 70% units may be installed.

Centrifugal compressors and their drivers are rarely spared due to cost, complexity, and history of good reliability.

12.3.7 Facility capabilities and restrictions

Facility capabilities refer to the local experience, limitations, and restrictions that may affect the driver selection. These factors include

- Energy sources
- Facility energy balance
- Future energy needs
- Cooling water availability
- Space limitations
- Environmental restrictions
- Operation and maintenance preference
- Vibration

12.3.7.1 Energy sources

Energy sources will vary between facilities and locations. For example, steam is not available on offshore platforms or in some remote locations. In addition, electrical power may be limited due to local utility capabilities or inadequate company-produced power. An assessment of the available energy sources and impact of their use may limit driver options.

12.3.7.2 Facility energy balance

When selecting a new driver, one should consider the effect on the existing utility systems (usually steam, electrical power, cooling water, fuel gas, and/or fuel oil). The effects on utility balances and loads are critical. For electrical drivers, an up-to-date electrical system single line diagram with equipment ratings and loads indicated is required to assess its impact. Similarly, an up-to-date stream balance is required to assess its impact of installing a new steam turbine.

12.3.7.3 Future energy needs

An understanding of the expansion or modification plans of the facility may impact the driver selection. The installation of a driver which is comparable with the current and future development plans for the facility is essential to optimize energy use and efficiency.

12.3.7.4 Cooling water availability

Cooling systems are required for reciprocating engines, gas turbines, and steam turbines. Most electric motors do not require water cooling. Reciprocating engines are commonly supplied with air coolers for the internally circulated coolant, but a source of makeup water is recommended. Gas turbines require either water or air coolers for lube oil temperature control and a source of fresh wash water for cleaning turbine internals. Steam turbines require lube oil coolers or once through water cooling of the bearing housing oil. The availability and quality of cooling water may impact the driver selection.

12.3.7.5 Space limitations

Space limitations have an impact on most jobs both onshore and offshore. This limitation is often covered by buying equipment and drivers packaged by contractors. However, tightly packaged machinery creates problems, such as lack of access for maintenance. Drivers capable of the power required for a service may not be acceptable due to space or weight limits. The evaluation of possible drivers for an application must include a critical review of packaging techniques, space available, and maintenance access requirements.

12.3.7.6 Environmental restrictions

Most domestic sites have restrictions based on minimizing environmental impact. These restrictions may include limits on emissions (from reciprocating engines and gas turbines), noise, height, paint color and type, lighting requirements, disposal, and others. The local requirements will vary and usually get more strict with time.

12.3.7.7 Operation and maintenance preference

Each facility has experience with the equipment which has been installed and operating. This experience may be good or bad, resulting in strong preferences to select or avoid certain equipment and drivers. Maintenance, technical, and operation personnel should be given an opportunity to review and approve proposed driver selections.

12.3.7.8 Vibration

Vibration magnitudes may sometimes influence the driver selection. For example, electric motors vibrate less than reciprocating engines and are therefore more desirable for offshore platforms.

12.4 Typical applications

12.4.1 Application tables

Table 12.1 is a summary of typical driver applications, and Tables 12.2–12.7 summarize typical application ranges of specific drivers and additional selection notes.

12.4.2 Electric motors

12.4.2.1 Induction motors

The induction motor is usually the first choice for an electric driver because of its rugged, simple and extremely reliable design, and low first cost. It accelerates high load inertias more readily and quickly, and is more reliable under transient load conditions than synchronous motors. The advantages of induction motors are as follows:

Table 12.1 Common combinations of drivers and driven equipment

Driven equipment	Electric motors		Steam turbines		Gas turbines	I/C engines
	Induction	**Synchronous**	**Backpressure**	**Condensing**		
Centrifugal pumps						
Single stage	B,D,G,I	–	D,G	–	–	D,G
Multistage	D,G	–	D,G	–	D,G	D,G
Positive displacement pumps						
Plunger type	B,G	–	–	–	–	D,G
Rotary gear	B,D,G	–	–	–	–	–
Rotary screw	B,D,G	“	–	–	–	D,G
Centrifugal compressors						
Single stage	D,G	–	D	–	–	–
Multistage	D,G	D,G	D,G	D,G	D,G	–
Positive displacement compressors						
Reciprocating	B,D,G	D,E,G	–	–	–	D,E,G,I
Rotary screw	B,D	–	–	–	–	B,D,G
Fans	B,D,G	–	D,G	G	–	
Generators	–	–	D,G	D,G	D,G	D,E,G
Vacuum pumps	B,D,G	–		–	–	
Mixers	B,D,G	–		–	–	

Key for drive arrangement options: *B*, belted; *D*, directly coupled; *E*, engine-type direct-connected; *G*, geared; *I*, integral.
Notes:

(1) Gas turbines and reciprocating engines are seldom applied with a gear to the driven equipment.
(2) There may be other combinations of applied with a gear to the driven drivers and driven equipment in use.
(3) However they are less frequently used. when motors are the selected driver the most common drive arrangement is direct coupled. the other options listed in population would be geared, belted, then integral.
(4) When steam turbines are selected for pump drivers, they are usually direct coupled. otherwise, the steam turbine driver is usually applied with a gear to the driven equipment.
(5) Belt drives are not common in refineries and can contribute to bearing problems.

Table 12.2 Induction motor—Application ranges and selection notes

Application ranges		
Horsepower	Typical	1/4 to 5000
	High	10,000
Voltage	Typical	230 to 4000
	5 to 250 hp	230 to 460
	250 to 500 hp	460 to 2300
	500 to 5000 hp	2300 to 4000
	5000+ hp	4000 to 13,200
Speed (rpm)	Typical	720 to 3600

Selection notes:

- The most common driver selection for all types of driven equipment.
- Are more efficient than steam turbines, gas turbines, or reciprocating engines.
- Are commonly packaged with driven equipment by the driven equipment vendor.
- Can have serious design, manufacturing, and testing problems at 2-pole speeds (3600 rpm) greater than or equal to 600 hp.
- Are available in several different enclosures (ODP, WP1, WP2, TEFC, TEWAC, etc.) which have a substantial impact on price, delivery, and reliability.
- Control and monitoring systems are simple compared to other driver options.
- Are more reliable than almost all other drivers.

Table 12.3 Synchronous motor—Application ranges and selection notes

Application ranges		
Horsepower	Typical	500 to 10,000
	High	40,000
Voltage	Typical	2300 to 13,800
	500 to 5000 hp	2300 to 4160
	5000+ hp	4000 to 13,800
Speed (rpm)	Typical	300 to 1200

Selection notes:

- The most common driver selection for slow speed (below 1200 rpm) driven equipment except generators.
- Is the most efficient driver. This is due to the unity (1.0) power factor which is commonly provided with these motors.
- Can be used for system power factor correction.
- Are commonly packaged with driven equipment by the driven equipment vendor.
- Are available in several different enclosures (ODP, WP1, WP2, TEFC, TEWAC, etc.). Most installations utilize the WP2 enclosure.
- Control and monitoring systems are simple compared to steam turbines, gas turbines, or reciprocating engines, but are usually more complicated than systems required for induction motors.
- Are commonly applied as drivers for equipment with pulsating loads (e.g., reciprocating compressors) because of their high rotor inertia. A rule of thumb: select a synchronous motor instead of induction motor if the nameplate horsepower of the motor exceeds the rated speed.

- Maintenance, purchase, and installation costs are lower.
- There are many combinations of nameplate power, voltage rating, and enclosures available, which makes them the predominate choice for all types of pump, compressor, blower, and fan drives.

Table 12.4 Noncondensing steam turbine—Application ranges and selection notes

		Single stage	Multistage
Application ranges			
Horsepower	Typical High	50 to 500 1000	1000 to 5000 10,000
Steam conditions (psig)			
	Typical inlet Low/high Typical exhaust Low/high	600 150/700 150 50/300	600 150/1500 150 50/600
Speed (rpm)	Typical	3600 to 5000	3600 to 12,000

Selection notes:

- Can have overlapping horsepower coverage with different steam rates for each selection. Operating factor, steam availability, and cost usually determines if the low-cost, high-steam-rate selection, or high-cost, low-steam-rate selection is the best choice.
- Steam temperatures above 850°F will require special materials.
- Steam purity becomes a major issue when temperature exceeds 700°F and pressure exceeds 700 psig to avoid any blade deposits.
- Maximum efficiencies for single stage are usually achieved when inlet to exhaust pressure ratio is between two and three.
- Higher ratios are common but are usually less efficient.

Table 12.5 Condensing steam turbine—Application ranges and selection notes

Application ranges		
Horsepower	Typical High	1000 to 10,000 15,000
Steam conditions (psig)		
	Typical inlet Low/high Typical exhaust	300 to 600 50/700 Condenser pressure (4 inch Hg)
Speed (rpm)	Typical	3600 to 10,000

Selection notes:

- Most commonly selected for generator or centrifugal compressor drives.
- Are usually installed with driven equipment by an engineering contractor, rarely by the equipment vendors.
- Control and monitoring systems are more complicated than backpressure steam turbines due to the associated system and equipment requirements.
- Are expensive to purchase, install, and maintain, therefore may not be the economical choice.
- Are usually selected to take advantage of an excess of low pressure steam.
- An elevated foundation is usually required to permit the use of a bottom exhaust nozzle orientation into the condenser mounted under the turbine.
- Steam temperatures above 850°F will require special materials.
- Steam purity becomes a major issue when the temperature exceeds 700°F and the pressure exceeds 700 psig to avoid blade deposits.

The major disadvantage is that their operability is dependent on a reliable source of power. For this reason, it is common practice to use company-produced power or steam turbine drivers for spare critical drivers. In some operations internal combustion engines are also used as spares.

Table 12.6 Gas turbines—Application ranges and selection notes

Application ranges		
Horsepower	Typical	1000 to 100,000
	High	270,000
Speed (rpm)	Typical	3600 to 15,000
	High	22,000

Selection notes:

- Usually selected for generator or centrifugal compressor drives.
- Are usually packaged with driven equipment by a packager which may not be the original equipment manufacturer, otherwise the packager is the gas turbine vendor.
- Control and monitoring systems are more complicated due to the associated systems (starting, fuel supply, lube oil, and protection) and equipment requirements.
- Are expensive to purchase and maintain, thereby impacting the economics.
- Are usually selected to take advantage of an available supply of natural gas (remote or offshore) or liquid fuel.
- It is important to consider the turbine performance at the worst-case ambient conditions (high temperature, high elevation, and high humidity) for power output capability relative to driven equipment requirements.
- Sometimes used as the spare machine driver where the main machine is electrical and steam is not available.

Table 12.7 Reciprocating gas and diesel engines—Application ranges and selection notes

Application ranges		
Horsepower	Typical	200 to 2000
	High	6000
Speed (rpm)	Typical	300 to 2100
	High	3600

Selection notes:

- Most commonly selected as drivers for centrifugal and positive displacement pumps, reciprocating and rotary compressors, generators, and fire water pumps.
- Are usually packaged by the driven equipment manufacturer or by the engine distributor.
- Control and monitoring systems are more complicated due to the associated systems (starting, fuel supply, lube oil, and protection) and equipment requirements.
- Are expensive to purchase and maintain, therefore may not be the economic choice.
- Are usually selected because of their portability, or to take advantage of an available supply of natural gas (remote or offshore) or liquid fuel.
- It is important to consider the engine performance at the worst-case ambient conditions (high temperature and altitude) for power output capability relative to driven equipment requirements.
- Engines typically have several ratings. These may include ratings such as intermittent, industrial, continuous, standby, fire pump, marine, and so on. Engines are normally bought based on their continuous duty rating and require at least a 10% margin between this rating and the driven equipment power requirement.
- Sometimes used as the spare machine driver where the main machine is electrical and steam is not available.

For a minimum investment when specifying induction motors, one should ensure the motor is not excessively oversized. The type of motor enclosure is usually dictated by the facility electrical area classification, the site conditions, and the noise attenuation required. Motor enclose type has a significant effect on the cost. Engineering judgment is required to select the minimum enclosure required. For more details, refer to Chapter 13.

12.4.2.2 Synchronous motors

The purchase and maintenance cost of synchronous motors is higher than that of induction motors. They are usually selected for slower speed and higher horsepower applications. However, synchronous motors have the following advantages over induction motors:

- Efficiency at full rated and partial loads is higher.
- Torque characteristics can be designed to more closely match the load and power supply systems.
- Starting or inrush currents are generally lower.
- Their economic design is for low speeds and high horsepower.
- They are available with unity or leading (overexcited) power factor for system power factor improvement. Induction motors operate at a lagging power factor.
- Their high inertia makes them well suited for pulsating loads such as reciprocating compressors.

12.4.3 Steam turbines

12.4.3.1 Backpressure or noncondensing turbines

Centrifugal machines can use steam turbines as the prime mover. The steam turbine can be backpressure or condensing. Backpressure turbines (refer to Fig. 12.14) exhaust wet steam at some intermediate pressure. Condensing turbines have a vacuum exhaust condition and a condensing system at the outlet (refer to Fig. 12.15).

Steam turbines are variable speed drivers. This is an advantage for centrifugal machines because capacity can be controlled by varying the running speed. Steam turbines are considered more reliable than motors because they do not fail when the power fails. This is especially important in "Quench" service where a compressor

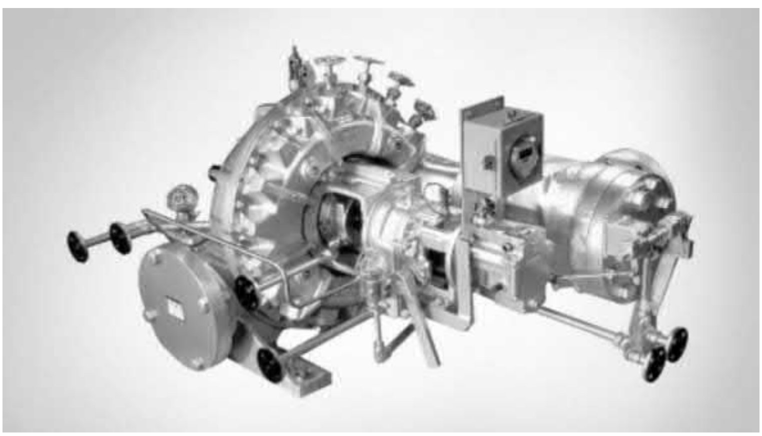

Fig. 12.14 Noncondensing (backpressure) turbine.

Fig. 12.15 Condensing turbine.

provides cooling gas to the system. Loss of the compression can lead to a temperature excursion. Steam turbines are less efficient than motors, but the overall cycle efficiency can be made competitive if the steam system is in balancer.

Backpressure turbines are the most common steam turbines selected. Their inlet steam pressure is usually the highest or an intermediate facility pressure level. The exhaust pressure levels are either intermediate or low plant pressures. For most applications, these turbines are single-stage, "Curtis wheel" designs which are selected to match the RPM of the driven equipment, frequently 3600 rpm.

Noncondensing backpressure turbines are usually smaller turbines under 1000 hp that are single stage. Steam turbines that drive pumps are usually noncondensing. They are usually purchased in accordance with API 611. Occasionally, noncondensing backpressure turbines are used to drive generators in refinery utility plants. These are called topping turbines and improve overall efficiency. They are in accordance with API 612.

Pumps are generally driven by smaller noncondensing turbines, and compressors are generally driven by larger condensing turbines.

12.4.3.2 Condensing turbines

Condensing turbines often use the abundance of low pressure steam (usually 50 or 150 psig (345 or 1034 kPa)) as inlet steam and exhaust to a vacuum condenser. These turbines are seldom the economic choice except as primary drives for high-horsepower critical services. Because of their size, complexity, and cooling water requirements, condensing turbines are expensive to purchase and install. Condensing turbines are usually larger drivers used for centrifugal compressors. They are purchased in accordance with API 612.

12.4.3.3 Multiple extraction and admission turbines

Multiple extraction and admission turbines are built to take advantage of different inlet steam sources (at different pressure) and/or exhaust at multiple steam pressures. A flanged connection is required for each inlet or exhaust pressure point on the turbine. For example, a condensing turbine with 150 and 50 psig (1034 and 345 kPa) inlet will have three connections (two inlet and one exhaust).

This type of turbine can potentially replace or augment steam let-down stations, but they are expensive to purchase and install, and have complicated control systems. Flash-type geothermal power plants utilize this type as the main generator driver.

12.4.4 Gas turbines

Gas turbines are normally applied as drivers for centrifugal compressors, generators, and large pumps. Many upstream gas turbines are installed at remote onshore facilities and offshore platforms. These facilities have a reliable and economic source of fuel, usually produced gas, available in sufficient quantities.

Gas turbines are used as generator, compressor, or pump drives. They are also used in cogeneration facilities. The turbine usually drives a generator and the exhaust heat from the turbine is used to produce steam or warm a heat transfer oil.

12.4.5 Reciprocating engines

Liquid fueled engines are commonly used as drivers for spare fire water pumps. These engines are also used to drive emergency storm water pumps and emergency generators. Reciprocating gas engines, both packaged high speed and integral, are used as reciprocating compressor drivers. These gas engines are not normally used for firewater service because of the difficulty in storing fuel for use during an emergency.

Except in remote locations where steam and electric power are not practical or in locations where fuel is priced very low, reciprocating engines are seldom selected as drivers because:

- Maintenance and capital costs are high.
- They usually require shelter.
- They are large, heavy, noisy, and vibrate considerably compared to other driver options.
- They are less reliable than other drive options.
- Emissions reduction equipment is often required which increases the cost and may decrease the reliability of the entire machine.

12.5 Packaging

12.5.1 Purpose

Packaging is a construction technique to modularize a driver and driven equipment for large projects in order to minimize construction time and labor costs.

Modular construction is used extensively for offshore platforms and onshore facilities located at remote sites.

Packaging consists of placing the driver, driven machine, auxiliaries, and control system on one or more packages. Piping, tubing, and wiring are routed between the various equipment items on each package and are then brought to a minimum number of terminal points at the perimeter of the package. These terminal points facilitate connecting the package to the system at the job site.

The size and complexity of packaged equipment ranges from simple air compressor package rated at a few hundred horsepower to a large and complicated package containing a gas turbine driving a generator rated at 70,000 hp.

12.5.2 Modular construction

Lifting or moving apparatus. Table 12.8 summarizes which drivers are typically packaged with driven equipment.

Tables 12.9 and 12.10 list estimating weights and dimensions for various drivers with reciprocating compressors. Table 12.11 presents typical dimensions and weights for integral gas engine-driven reciprocating compressors. Table 12.12 presents weights and dimensions for packaged pumps similar to Table 12.11. Table 12.13 provides information for turbine drivers and centrifugal compressors with their enclosure.

The only process driver/driven machines usually not packaged are large reciprocating compressors and large centrifugal fans. However, they may have packaged auxiliary systems, such as a lube oil system.

12.5.3 Advantages and disadvantages

Table 12.14 presents the advantages, disadvantages, and limitations of packaging.

Table 12.8 Typical combinations of drivers packaged with driven equipment

Driven unit	Motors	Reciprocating engines	Gas turbines
Pumps			
Positive displacement	X	X	
Centrifugal	X	X	X
Compressors			
Screw	X		
Integrally geared centrifugal air	X		
Centrifugal process and refrigeration	X		
Centrifugal	X		X
High-speed reciprocating	X	X	
Centrifugal fans	X	X	
Engine generator sets		X	
Generators			X

Table 12.9 Packaged high-speed reciprocating compressors with coolers—Typical weights and dimensions

Driver	Horsepower	$L \times W \times H$ ft	Weight kips[a]	MMSCFD[b]
Engine	1200	$34 \times 13 \times 16$	100	19.1
Engine	565	$35 \times 12 \times 14$	60	2.3
Engine	500	$30 \times 12 \times 14$	47	4.7
Engine	450	$30 \times 18 \times 18$	55	1.8
Engine	415	$30 \times 12 \times 13$	48	5.2
Motor	350	$20 \times 12 \times 14$	51	2.0

[a]One kip = 1000 pounds.
[b]MMSCFD = million standard cubic feet per day.

Table 12.10 Packaged high-speed reciprocating compressors without coolers—Typical weights and dimensions

Driver	Horsepower	$L \times W \times H$ ft	Weight kips[a]	MMSCFD[b]
Engine	2600	$38 \times 12 \times 10$	106.0	13.2
Engine	1000	$35 \times 13 \times 12$	71.0	9.5
Turbine	2000	$35 \times 12 \times 12$	100.0	17.0
Turbine	1000	$50 \times 12 \times 12$	80.0	11.6
Motor	2000	$22 \times 12 \times 10$	89.5	12.5
Motor	900	$26 \times 12 \times 8$	49.0	12.5

[a]One kip = 1000 pounds.
[b]MMSCFD = million standard cubic feet per day.

Table 12.11 Packaged integral compressors—Typical weights and dimensions

Horsepower	$L \times W \times H$ ft	Weight kips[a]
1000	$23 \times 14 \times 12$	100
1500	$27 \times 14 \times 12$	140
2000	$33 \times 16 \times 15$	190
2400	$33 \times 16 \times 15$	210

[a]One kip = 1000 pounds.

12.6 Foundations

12.6.1 Purpose

Foundations dampen vibration generated by the machinery and reduce transmission of these forces to the surrounding equipment. Foundations also keep machinery in alignment.

Table 12.12 Packaged pump with driver—Typical weights and dimensions

Driver	Horsepower	RPM driver	RPM pump	$L \times W \times H$ ft	Weight kips[a]	GPM	Discharge pressure (psi)
Motor	20	1800	1800	$5 \times 2 \times 3$	2	60	150
Motor	100	1800	3600	$10 \times 5 \times 6$	12	250	415
Motor	500	3600	3600	$20 \times 12 \times 12$	45	1000	520
Motor	500	1800	3600	$26 \times 12 \times 12$	52	1100	470
Engine	500	900	3600	$30 \times 12 \times 12$	60	1100	500
Engine	750	1200	3600	$36 \times 12 \times 15$	85	1500	520
Motor	1000	1800	1800	$30 \times 14 \times 14$	70	2200	470

[a]One kip = 1000 pounds.

To perform these essential functions throughout the life of the installation, the foundation must be sized to support the weight of the machinery while imposing a tolerable bearing pressure on the soil or structure. It must be properly designed so that the system, consisting of the foundation, soil, machinery, and piping, will not vibrate excessively.

The purchaser of the machinery, usually the company, is normally responsible for the design of the foundation. The vendor or manufacturer of the driver is seldom asked to take on this responsibility because they do not have specific knowledge about the soil conditions at the site or knowledge of the driven machine.

In addition to knowing the dimensions and weights of the machinery to be supported, foundation designers must also know the magnitude, direction, and frequency of the dynamic forces that the machinery will exert on the foundation.

12.6.2 Dynamic forces of rotating machines

12.6.2.1 General considerations

Rotating machines (gas turbines, steam turbines, motors, generators, and gear boxes) normally exert much smaller dynamic forces than reciprocating machines. In any case, these forces must be accounted for to avoid a potentially serious vibration problem during operation.

A fault in the design of a concrete foundation is difficult to correct after the concrete has been poured. There is no easy way to add mass, alter the stiffness, or adjust the damping to change the natural frequency of a concrete foundation in an effort to move the system away from a condition of resonance. In a few cases, it has been necessary to break out an existing foundation and pour a redesigned foundation to solve a serious vibration problem. Such instances are expensive and time consuming.

Guidelines have been developed through the years for the allowable vibration of the foundation; however, criteria for defining the forces to be used in foundation

Table 12.13 Packaged turbine-driven centrifugal compressor—Typical weights and dimensions

ISO horsepower	Number of shafts	Speed (rpm)	Turbine-compressor skid dimensions $L \times W \times H$, ft	Ancillary equipment dimensions $L \times W \times H$, ft	Approximate weight (kips)[a]	Skid weight (kips)[a] per additional foot	With enclosure type add kips[a]	Engine control cab add kips[a]
4250			34.5 × 8.0 × 20.0		33	0.8		
16,000			59 × 10.0 × 24.0		96	1.6		
26,500			61.0 × 10.0 × 24.0		105	1.8		
4900	Single		30.0 × 8.0 × 8.0	21.4 × 7.8 × 10.7	74			
4900	Dual		34.0 × 8.0 × 8.0	21.4 × 7.8 × 10.7	81			
10,600		8140	48.5 × 8.0 × 11.8	38.0 × 18.0 × 15.1	107.5	1.2		
3830		15,700	27.2 × 7.8 × 8.3	21.4 × 7.8 × 10.0	52.1	0.6		
1165		22,300	23.1 × 5.8 × 7.3	12.0 × 6.0 × 4.8	18.5	0.4		
4900	Single		26.0 × 8.0 × 8.0		55.0		Open side-4.1 Total end. 5.3	0.5
4900	Dual		29.0 × 8.0 × 8.0		66.0		Open side-4.1 Total side 5.3	0.5
2500	Single		28.0 × 8.0 × 8.5		33.0		Open side-3.4 Total end. 5.0	0.5
2500	Dual		33.0 × 8.0 × 8.5		48.0		Open side-3.4 Total end 5.0	0.5
1875			25.0 × 8.0 × 8.6		32.0		Open side-4.1 Total end 5.3	0.5

[a]One kip = 1000 pounds.

Table 12.14 Advantages and disadvantages of packing

Advantages	**Disadvantages**
Usually lower installation costs. Packaging done onshore is less costly than installing individual machines offshore	Difficult to write specifications to cover all proposed driver/driven equipment with accessories for modularized equipment. This may cause extras later. Technical bids are difficult to compare on an even basis
Faster track with turnkey projects as packaging is a technique used to minimize construction time	Usually insufficient prequalification of third-party packagers
Compact skids requiring minimum platform space and land plot for onshore facilities	Packagers are often technically ill equipped and operate on a tight financial budget. They also cannot be counted on for good technical coordination of subvendors
Easier to shop test assembled driver/driven load trains on skids	Transportation limits size and weight
Clearer vendor responsibility for equipment on skids	Shipping damage risks are greater without proper precautions
Easier to relocate to another facility	Partial field assembly is frequently required
	Baseplate stiffness for skids must be based on the demands of overall rigidity, lift, resonance, and thermal effects. Packagers may try to cut corners on baseplate stiffness by using lighter material which could eventually create serious operating problems
	Field cleanup and alignment is still necessary
	Provisions must be built-in for operability and maintenance accessibility

design have been lacking. A misunderstanding between the foundation designer and the machine manufacturer regarding the unbalanced forces to be allowed for in the design has contributed to many foundation vibration problems. These problems have been caused by not designing for the actual dynamic forces but rather for some lower value due to communication problems between the foundation designer and the machine manufacturer.

Depending on how the question concerning unbalanced force is asked, the manufacturer might respond with the rotor's residual unbalance from the dynamic balancing machine. The balancing machine tolerance is a very small number that might be only 1/20th of the actual force rated speed. At other times arbitrary values are assumed for foundation design, yet they are typically not representative of actual conditions.

12.6.2.2 Dynamic forces

For adjustable speed drivers, it is more difficult to avoid the natural frequencies of structural supports. This is important to consider when designing for offshore installations.

The dynamic force generated by the rotor is typically related to its running speed and the amplitude of vibration. Because of the complexity of the subject, it is impossible to accurately predict the behavior of a rotor system with one or two simple equations.

However, standards have been developed for allowable limits of vibration for new machinery. One of the most widely used standards is the API limit for dynamic and rotary machines:

$$A_V = 2$$

or

$$= [12,000/N]^{0.5} \tag{12.1}$$

whichever is less

where

A_V = peak-to-peak amplitude (displacement) of shaft vibration in mils (0.001 inches)
N = rated speed, rpm

Note: Below 3000 rpm the limit is 2 mils except for electric machines.

12.6.2.3 Vibration limit converted to dynamic force

The following equation may be used for calculating the force used in the foundation design. This equation is based on a vibration three times the amplitude calculated from Eq. (12.1). A safety factor of three is recommended because that is about the maximum vibration level where a machine would be allowed to continue to operate.

$$F = 4.3 \times 10^{-8} \left(N^2\right)(W_R)(A_V) \tag{12.2}$$

where

F = dynamic force, pounds
N = rpm
W_R = weight or rotor, pounds
A_V = vibration level, mils, from Eq. (12.1)

The force calculated is a rotor vector, and it should be assumed that it is acting perpendicularly at the center of the rotor. It should also be assumed that there will be a 50% reaction at each bearing from the unbalanced rotating force. The reactions at the machine's hold-down bolts can then be resolved.

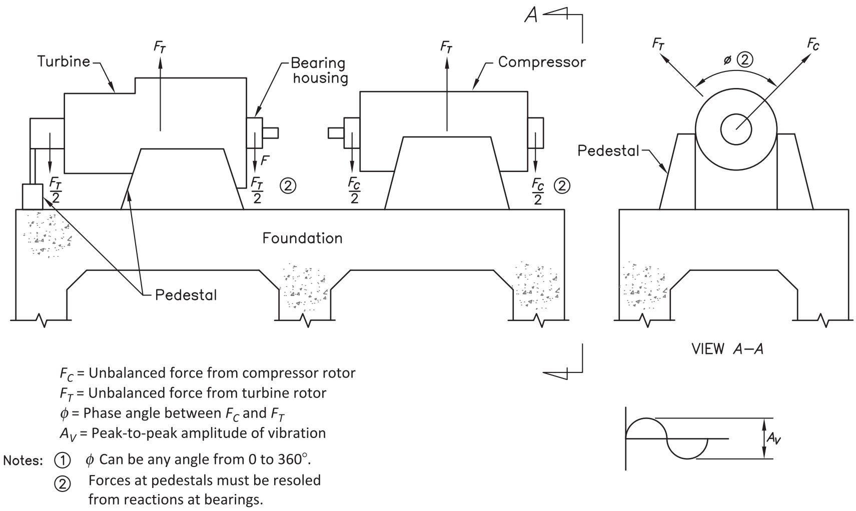

Fig. 12.16 Unbalanced forces from turbine and compressor rotors.

Fig. 12.16 shows the resolution of these forces to foundation reactions. The latter reactions are transmitted to the foundation via soleplates or baseplate and anchor bolts.

Although Eq. (12.2) is quite conservative, foundation designers occasionally add a factor above the dynamic force. Five times the API vibration limit has been used as a design criterion in some cases where there were special concerns about the design. This would provide a safety factor of 1.67 beyond Eq. (12.2). To make the calculation, substitute 7.1 for 4.3 in Eq. (12.2).

12.6.2.4 Other considerations

Sometimes the question arises about whether foundation would survive if a large chunk of metal, such as a piece of an impeller or turbine blade, were thrown off the rotor while running at full speed. Another question might be whether the foundation should be designed to accommodate such an occurrence. Foundations usually will survive such accidents, although some repairs to anchor bolts, hold-down bolts, or bearing pedestals may be necessary. Such occurrences are generally not taken into account in the design. The forces involved are extremely high, and it is impossible to predict their magnitude. Bolting and structures are normally checked for adequacy at 10 times rated torque. This value is often used on turbine generator foundations, because a short circuit can cause an instantaneous torque increase to that level. Similarly, a rotor might cause such a torque increase in the event of a severe rub.

Generally, it is recommended that the natural frequency of the foundation system be at least 30% above or below the frequency of any driver operating speed.

As a general rule of thumb, the weight of the foundation should be no less than three times the weight of the rotating machinery it supports. Induction motor drivers 600 hp, 3600 rpm and larger should be mounted directly on concrete foundations. This helps to avoid rotor resonance difficulties that can be caused by a less rigid base, lowering the rotor critical speed to near the operating speed. Also, foundation and structural supports should be symmetrical around the centerline of the rotor.

12.6.3 Unbalanced forces and moments—Reciprocating drivers or drivers for reciprocating compressors

12.6.3.1 General considerations

When drivers are reciprocating or are on the same foundation with a driven load which is reciprocating, the foundation design must include the forces generated by the reciprocating machines. Reciprocating machines generate primary and secondary forces and couples as a result of unbalanced rotating masses and unbalanced reciprocating that accelerate and decelerate with each revolution. Primary forces occur at the running speed frequency and secondary forces at twice running speed. These forces and couples react at the main bearings and the resulting forces are transmitted to the foundation via the frame and bolting. These forces and moments are called shaking forces.

Reciprocating compressor pistons are usually of unequal size, thus it is difficult, if not impossible, to balance the masses. Weights can be added to crossheads and/or piston material can be varied to attempt a balancing effect. For example, a large aluminum piston could be placed opposite a smaller cast iron piston in a horizontal opposed type of machine. Counterweights can be added to the crankshaft throws opposite the crankpin to reduce primary forces. Single-cylinder machines are not easy to balance. Two-throw machines are somewhat better. Depending on the crankshaft configuration (i.e., the orientation of the crank throws), a four-throw horizontally opposed compressor can usually be balanced to very low values or primary and secondary forces and moments.

Driver and reciprocating machine designs vary widely in terms of the number and size of cylinders and crankshaft configuration combined with either a large motor or internal combustion engine. There is no "rule of thumb" to predict the magnitude of forces on the foundation. The vendors of both the driver and reciprocating machine must be requested to submit preliminary values of forces and moments for the machines so that a foundation design that includes both machines can be performed.

12.6.3.2 Recommendations

It is normally recommended that the natural frequency of the foundation system be at least 30% above or below the primary (running speed) and secondary (twice running speed) frequencies of the driver and the reciprocating machine.

For reciprocating machines, as a rule of thumb, the weight of the foundation should be at least five times the combined weight of the frame, cylinders, and driver.

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Dr. Maurice I Stewart PE, LLC is Founder and Principle of Stewart Training and Consulting (STC), LLC, an international training and consulting firm. Dr. Stewart has over 45 years of experience that he passes along to help the reader get a multi-discipline approach in solving problems associated with pump and compressor systems. Currently training and teaching all over the world, Dr. Stewart is a former professor of petroleum engineering at Louisiana State University.

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