ANSI/HI 3.1-3.5-2008

Approved American National Standard

ANSI

American National Standard for

Rotary Pumps

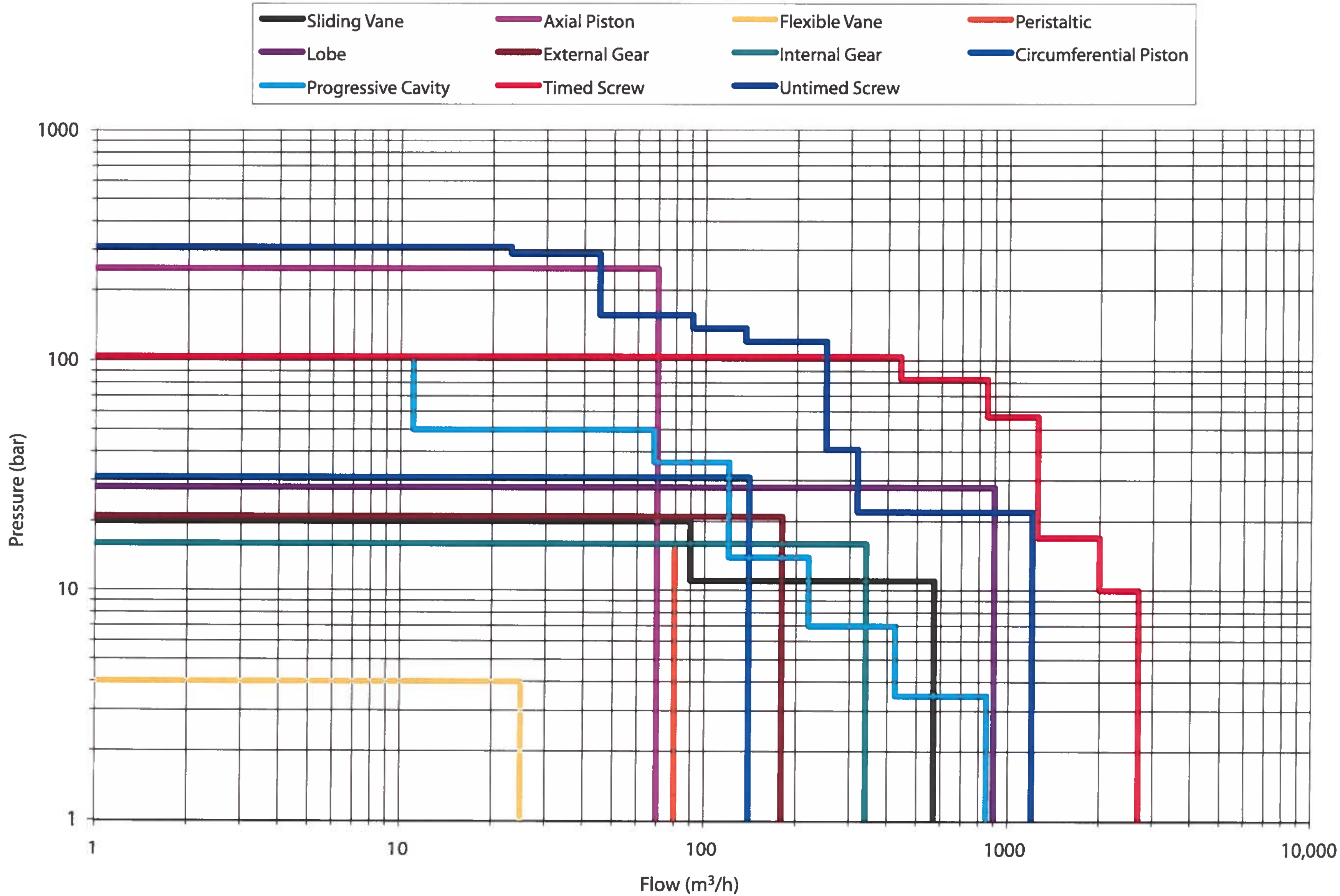
for Nomenclature, Definitions, Application and Operation

ANSI/HI 3.1-3.5-2008

Hydraulic Institute

6 Campus Drive
First Floor North
Parsippany, New Jersey
07054-4406
www.Pumps.org

Rotary pump consolidated range chart (metric)



ANSI/HI 3.1-3.5-2008

American National Standard for

Rotary Pumps

for Nomenclature, Definitions, Application and Operation

Sponsor

Hydraulic Institute

www.Pumps.org

Approved July 15, 2008

American National Standards Institute, Inc.

American National Standard

Approval of an American National Standard requires verification by ANSI that the requirements for due process, consensus and other criteria for approval have been met by the standards developer.

Consensus is established when, in the judgement of the ANSI Board of Standards Review, substantial agreement has been reached by directly and materially affected interests. Substantial agreement means much more than a simple majority, but not necessarily unanimity. Consensus requires that all views and objections be considered, and that a concerted effort be made toward their resolution.

The use of American National Standards is completely voluntary; their existence does not in any respect preclude anyone, whether he has approved the standards or not, from manufacturing, marketing, purchasing, or using products, processes, or procedures not conforming to the standards.

The American National Standards Institute does not develop standards and will in no circumstances give an interpretation of any American National Standard. Moreover, no person shall have the right or authority to issue an interpretation of an American National Standard in the name of the American National Standards Institute. Requests for interpretations should be addressed to the secretariat or sponsor whose name appears on the title page of this standard.

CAUTION NOTICE: This American National Standard may be revised or withdrawn at any time. The procedures of the American National Standards Institute require that action be taken periodically to reaffirm, revise, or withdraw this standard. Purchasers of American National Standards may receive current information on all standards by calling or writing the American National Standards Institute.

Published By

Hydraulic Institute
6 Campus Drive, First Floor North
Parsippany, NJ 07054-4406

www.Pumps.org

Printed in the United States of America

ISBN 978-1-880952-80-1

Recycled paper

Contents

Figures

Tables

Foreword (Not part of Standard)

Scope

The purpose and aims of the Institute are to promote the continued growth of pump knowledge for the interest of pump users and pump manufacturers and to further the interests of the public in such matters as are involved in manufacturing, engineering, distribution, safety, transportation and other problems of the industry, and to this end, among other things:

a) To develop and publish standards for pumps;

b) To collect and disseminate information of value to its members and to the public;

c) To appear for its members before governmental departments and agencies and other bodies in regard to matters affecting the industry;

d) To increase the amount and to improve the quality of pump service to the public;

e) To support educational and research activities;

f) To promote the business interests of its members but not to engage in business of the kind ordinarily carried on for profit or to perform particular services for its members or individual persons as distinguished from activities to improve the business conditions and lawful interests of all of its members.

Purpose of Standards

1) Hydraulic Institute Standards are adopted in the public interest and are designed to help eliminate misunderstandings between the manufacturer, the purchaser and/or the user and to assist the purchaser in selecting and obtaining the proper product for a particular need.

2) Use of Hydraulic Institute Standards is completely voluntary. Existence of Hydraulic Institute Standards does not in any respect preclude a member from manufacturing or selling products not conforming to the Standards.

Definition of a Standard of the Hydraulic Institute

Quoting from Article XV, Standards, of the By-Laws of the Institute, Section B:

"An Institute Standard defines the product, material, process or procedure with reference to one or more of the following: nomenclature, composition, construction, dimensions, tolerances, safety, operating characteristics, performance, quality, rating, testing and service for which designed."

Comments from users

Comments from users of this Standard will be appreciated, to help the Hydraulic Institute prepare even more useful future editions. Questions arising from the content of this Standard may be sent to the Technical Director of the Hydraulic Institute. The inquiry will then be directed to the appropriate technical committee for provision of a suitable answer.

If a dispute arises regarding the content of an Institute Standard or an answer provided by the Institute to a question such as indicated above, the point in question shall be sent in writing to the Technical Director of the Hydraulic Institute, who shall initiate the Appeals Process.

Revisions

The Standards of the Hydraulic Institute are subject to constant review, and revisions are undertaken whenever it is found necessary because of new developments and progress in the art. If no revisions are made for five years, the standards are reaffirmed using the ANSI canvass procedure.

Units of Measurement

Metric units of measurement are used; and corresponding US customary units appear in brackets. Charts, graphs and sample calculations are also shown in both metric and US customary units. Since values given in metric units are not exact equivalents to values given in US customary units, it is important that the selected units of measure to be applied be stated in reference to this standard. If no such statement is provided, metric units shall govern.

Consensus

Consensus for this standard was achieved by use of the canvass method. The following organizations, recognized as having an interest in the Rotary Pump Standard, were contacted prior to the approval of this revision of the standard. Inclusion in this list does not necessarily imply that the organization concurred with the submittal of the proposed standard to ANSI.

Bantrel
Bechtel Power Corporation
Buse, Fred – Consultant
California Polytechnical State University
Flowserve Pump Division
Grundfos Pumps Corporation
Healy Engineering
Intelliquip, LLC
J.A.S. Solutions Ltd.
Malcolm Pirnie
McFarland Pump Company, LLC
Moyno, Inc.
Powell Kugler, Inc.
Sulzer Pumps (US), Inc.
Viking Pump, Inc.
Weir Floway, Inc.

Rotary Committee Members

Although this standard was processed and approved for submittal to ANSI by the Canvass Method, a working committee met many times to facilitate the development of this standard. At the time it was developed, the committee had the following members:

Chair – Alan G. Wild, Moyno, Inc

Committee Members	**Company**
Randolph K. Bennett	Leistritz Corporation
Todd E. Brown	Moyno, Inc.
James B. Casey	Milton Roy Americas
Trygve Dahl	Intelliquip, LLC
David G. McKinstry	IMO Pump
Michael L. Mueller	Flowserve Pump Division
John W. Owen	IMO Pump
Manor M. Parikh	Siemens Water Technologies
John E. Purcell	Roper Pump Company
James K. Simonelli	Roper Pump Company
Fred F. Walker	Weir Floway, Inc.

Other contributors

Name	**Company**
Fred Buse	Consultant
Michael Mulcahy	Tuthill Pump Group
Kathy Parry	formerly of Tuthill Pump Group
Dan Ross	formerly of Tuthill Pump Group

3.1 Types and nomenclature

A rotary pump is a positive displacement pump consisting of a chamber containing gears, cams, screws, vanes, plungers, or similar elements actuated by relative rotation of the driveshaft to casing, and which has no separate inlet and outlet valves. These pumps are characterized by their close running clearances.

There are seven common basic types of rotary pumps identified by the type of pumping element. Relationships between these types of pumps are illustrated in Figure 3.1.

- Rotary pumps
  - Sliding vane
    - Vane in rotor
      - Constant displacement
      - Variable displacement
    - Vane in stator
  - Piston, Axial
    - Constant displacement
    - Variable displacement
  - Flexible member
    - Flexible vane
    - Peristaltic
  - Lobe
    - Single
    - Multiple
  - Gear
    - External, timed and untimed
      - Spur
      - Helical
      - Herringbone
    - Internal
      - Crescent
      - No crescent
  - Circumferential piston
    - Single
    - Multiple
  - Screw
    - Single – Progressing cavity
    - Multiple
      - Timed
      - Untimed

Figure 3.1 — Types of rotary pumps

Capability table

Overview

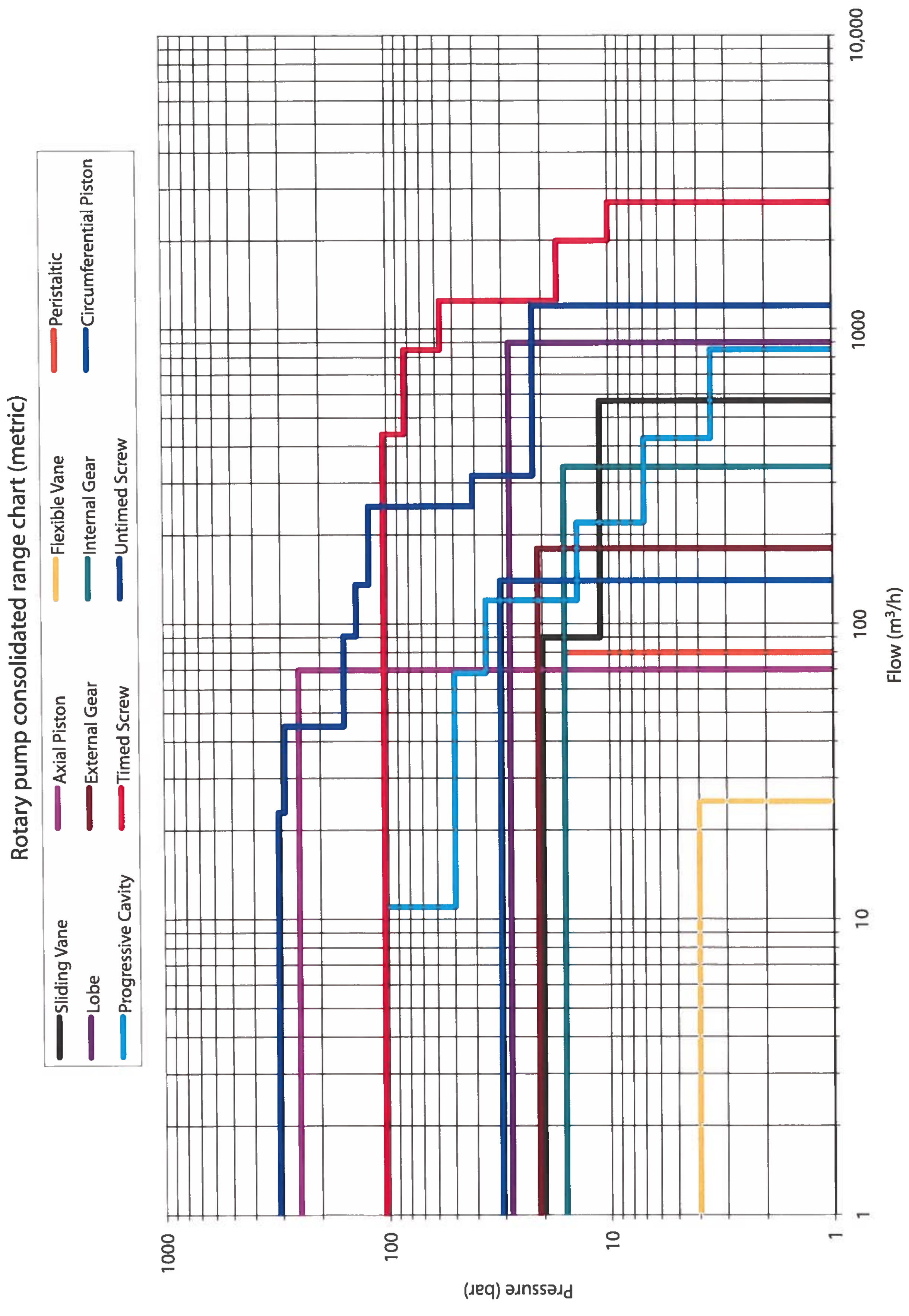
The capability table with individual technology descriptions provides an overview of operating ranges and application characteristics for the industrial rotary pumps covered in this document. These ranges are not absolute because custom designs at alternate speeds, clearances, or with special materials may be available through a thorough understanding of the application between the user and the manufacturer. The maximum parameter listed is only an indication of the range for that parameter and typically not all maximum values can be obtained simultaneously. Relative capabilities (i.e., abrasive handling, shear sensitivity, and pulsation) are highly dependent on application conditions and the pump technology to which the comparison is made. Shear sensitivity relates to the effect the pump has on the liquid. The abrasive handling capabilities ratings allow material enhancements. Consultation with the supplier is recommended to confirm specific application recommendations and to investigate special designs, which are often available to provide unique solutions.

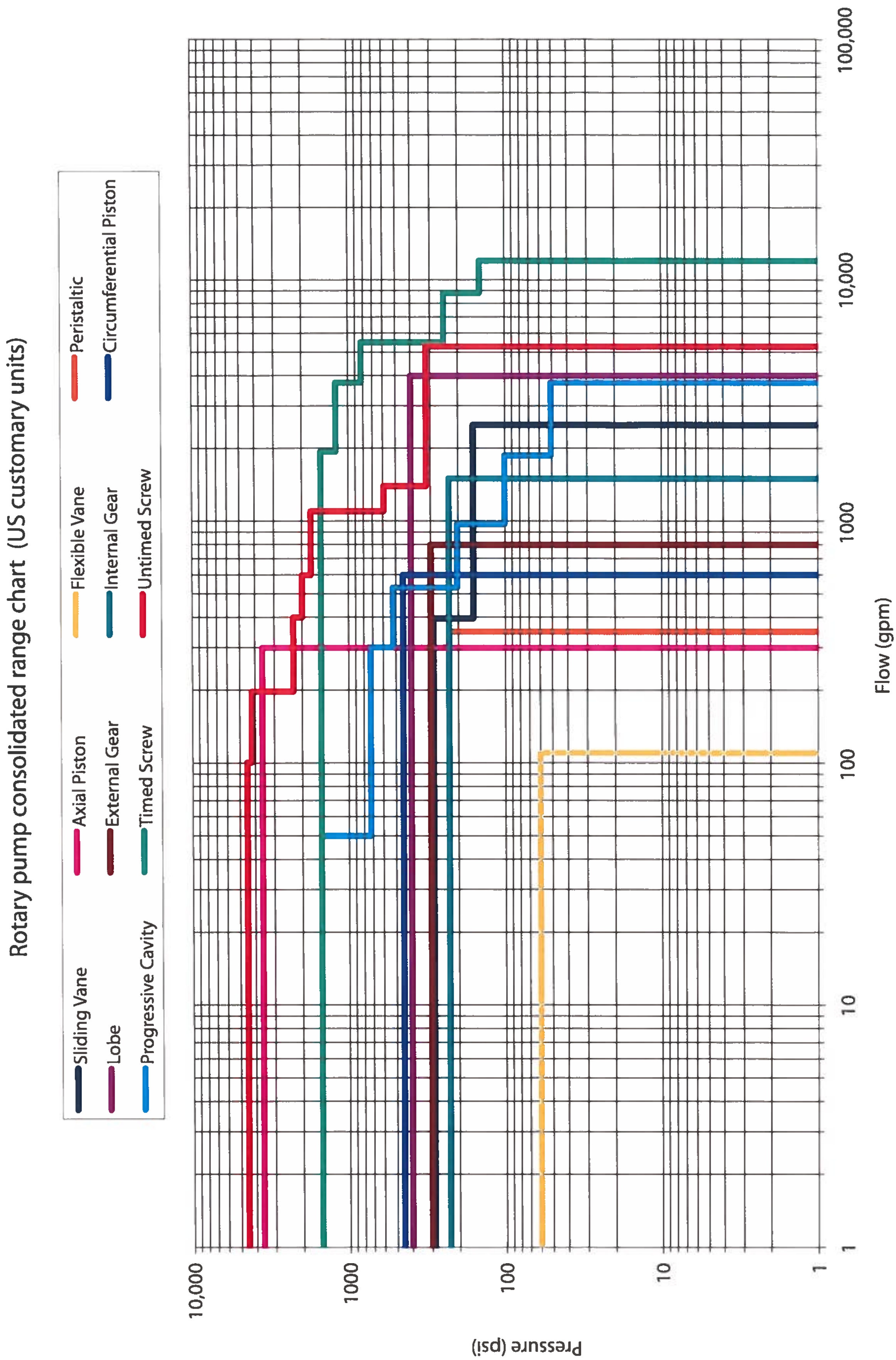
Capability table – Metric

Pump type	Flow m^3/h	Pressure *bar*	Viscosity *cSt* (×1000)	Solids *cm*	Temps. °C	Dry, Self-priming	Wet, Self-priming	Dry running	Reversible	Abrasive handling	Shear sensitive	Pulseless	Power *kW*
Sliding vane	570	20	220	0.08	200	Y	Y	Y	Y	Fair	Fair	Fair	0.75-190
Piston, axial	70	250	0.44	Clear	60	N	Y	Y	Y (special)	Poor	Poor	Poor	0.75-450
Flexible vane	25	4	22	Clear	90	Y	Y	N	N	Fair	Good	Good	0.15-4
Peristaltic	80	16	44	3.3	80	Y	Y	Y	Y	Excellent	Excellent	Poor	0.1-30
Lobe	900	28	440	6.35	177	N	Y	Y	Y	Good	Excellent	Fair	0.75-160
Gear,													
external	180	21	440	0.08	275	Y	Y	Y	Y	Good	Good	Good	0.37-110
internal	340	16	440	0.08	275	Y	Y	N	Y	Good	Good	Good	0.37-110
Circumferential piston	140	31	1000	3.2	275	Y	Y	Y	Y	Good	Excellent	Poor	0.75-150
Progressing cavity	850	104	440	9	205	Y	Y	N	Y	Excellent	Excellent	Excellent	0.1-150
Timed screw	2700	104	990	0.08	320	N	Y	Y	Y	Good	Good	Excellent	3.7-1500
Untimed screw	1200	310	220	Clear	275	N	Y	N	Y (special)	Good	Good	Excellent	0.75-750

Capability table – US customary units

Pump type	Flow *gpm*	Pressure *psi*	Viscosity SSU *(million)*	Solids *in.*	Temps. °F	Dry, Self-priming	Wet, Self-priming	Dry running	Reversible	Abrasive handling	Shear sensitive	Pulseless	HP range
Sliding vane	2500	290	1.0	1/32	400	Y	Y	Y	Y	Fair	Fair	Fair	1-250
Piston, axial	300	3600	0.002	Clear	140	N	Y	Y	Y (special)	Poor	Poor	Poor	1-600
Flexible vane	110	60	0.1	Clear	195	Y	Y	N	N	Fair	Good	Good	0.2-5
Peristaltic	350	230	0.2	1.3	175	Y	Y	Y	Y	Excellent	Excellent	Poor	0.05-40
Lobe	4000	400	2.0	2.5	350	N	Y	Y	Y	Good	Excellent	Fair	1-210
Gear,													
external	800	300	2.0	1/32	525	Y	Y	Y	Y	Good	Good	Good	1-150
internal	1500	230	2.0	1/32	525	Y	Y	N	Y	Good	Good	Good	1-150
Circumferential piston	600	450	4.5	1.25	525	Y	Y	Y	Y	Good	Excellent	Poor	1-200
Progressing cavity	3750	1500	2.0	3.5	400	Y	Y	N	Y	Excellent	Excellent	Excellent	0.1-200
Timed screw	12,000	1500	4.5	1/32	700	N	Y	Y	Y	Good	Good	Excellent	5-2000
Untimed screw	5300	4500	1.0	Clear	500	N	Y	N	Y (special)	Good	Good	Excellent	1-1000


Rotary pump consolidated range chart (metric)
Sliding Vane
Lobe
Progressive Cavity
Axial Piston
External Gear
Timed Screw
Flexible Vane
Internal Gear
Untimed Screw
Peristaltic
Circumferential Piston
Flow (m3/h)
1
10
100
1000
10,000
Pressure (bar)
1000
100
10
1


Rotary pump consolidated range chart (US customary units)
Sliding Vane
Lobe
Progressive Cavity
Axial Piston
External Gear
Timed Screw
Flexible Vane
Internal Gear
Untimed Screw
Peristaltic
Circumferential Piston
Flow (gpm)
1
10
100
1000
10,000
100,000
Pressure (psi)
10,000
1000
100
10
1

3.1.1 Scope

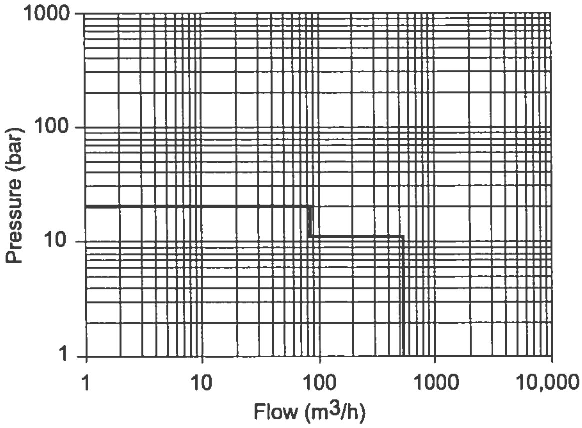
This Standard applies to industrial/commercial rotary positive displacement pumps. It includes: types and nomenclature; definitions; design and application; and installation, operation, and maintenance. It does not include standards on magnetic drives for sealless pumps nor rotary pumps primarily used for fluid power applications.

In order to provide a comprehensive overview of features and attributes of rotary pumps, Paragraphs 3.1.2 through 3.1.8.2.2 contain descriptive material covering the basic technologies highlighted in Figure 3.1. These include a description of each configuration, typical applications, and general operating ranges for commercially available designs for viscosity, flow, and pressure. Operating ranges stated represent only a single hydraulic parameter and do not indicate that all maximum conditions can be met simultaneously. Suppliers should be contacted for such details.

3.1.2 Sliding vane (rigid)

In this rotary pump technology the vane or vanes are moved by a rotor, thereby drawing fluid into and forcing fluid out from the pumping chamber formed in cooperation with the pump casing. These pumps may be made with vanes in either the rotor or stator and with radial hydraulic forces balanced or unbalanced on the rotor. Figure 3.1.2a (page 15) illustrates a vane-in-rotor constant displacement unbalanced pump. Figure 3.1.2b illustrates a vane-in-stator constant displacement unbalanced pump. Vane-in-rotor pumps also may be made with variable displacement pumping elements.

Sliding vane pump range chart



A common design has a number of vanes that are free to move into and out of slots in the pump rotor, which is inside an eccentrically shaped casing that acts as a cam. In this design, when the driver turns the rotor, centrifugal forces, internal pusher rods, and/or pressurized fluids causes the vanes to move outward in their slots and bear against the inner bore of the pump, forming pumping chambers. As the rotor revolves, fluid flows into the area between the vanes (pumping chambers) when they pass the suction port. The fluid is transported around the pump casing until the discharge port is reached. At that point the fluid is squeezed out into the discharge piping.

In variable capacity designs, displacement of the pump is changed by mechanical movement of the cam ring relative to the rotor. Appropriately configured designs do not need relief valves but simply move the cam to compensate for overpressure until the flow is reduced to zero.

Engineered vane materials make these pumps well-suited for low-viscosity, nonlubricating liquids. Such liquids include solvents, fuel oils, gasoline, refrigerants, and liquefied gas. They handle fluid viscosities ranging from 0.5 cSt to 220,000 cSt (1,000,000 SSU), which are used in a wide range of industries from aviation and automotive to textile. In proper configurations they can be used for fluid temperatures from –29°C (–20°F) to 204°C (400°F) and pressures to 20 bar (290 psi).

At volumes below 3 m^3/h (13 gpm) direct four-pole motor drives are possible. In general for larger flows up to 570 m^3/h (2500 gpm) design rotating speeds are typically below 600 rpm. Because of their versatility they are available in a wide range of materials, such as stainless steel, nodular iron, cast iron, bronze, and aluminum.

3.1.3 Axial piston pumps

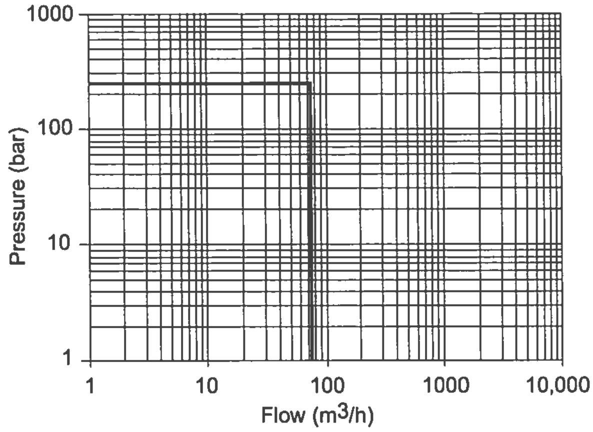
In this pump type fluid is drawn in and forced out by multiple pistons that reciprocate within cylinders. The reciprocating motion is created by a cam plate that is inclined at an angle with the pump centerline and does not rotate.

One end of each piston is held in contact with the cam plate as the cylinder block and piston assembly rotates with the driveshaft. This causes the pistons to reciprocate within the cylinders. The length of the piston stroke is proportional to the angle that the cam plate makes with the pump centerline. Valving is accomplished by rotation of the pistons and cylinders over the inlet and outlet ports.

In fixed displacement axial piston pumps the angle of the cam plate with respect to the pump centerline is fixed. In variable displacement axial piston pumps the angle of the cam plate with respect to the pump centerline can be varied.

Axial piston pumps have relatively low flow rates, 70 m^3/h (300 gpm), but are capable of operating at pressures to 250 bar (3600 psi). Typical applications include the spraying of clean fluids or high-pressure pumping of lubricants. This type of pump will typically operate at synchronous motor speeds. Figure 3.1.3 illustrates a fixed displacement axial piston pump.

Axial piston pump range chart



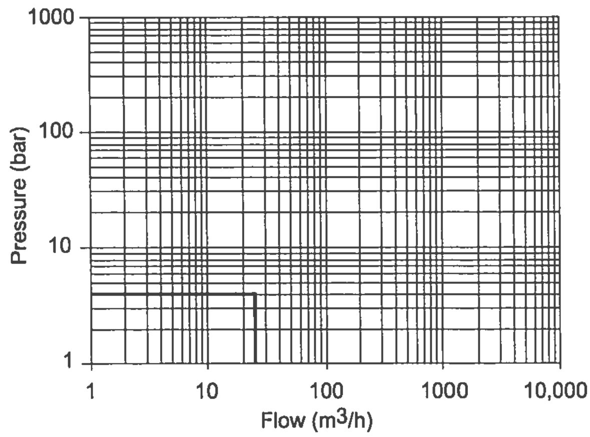
3.1.4 Flexible member

3.1.4.1 Flexible vane

Another member of the rotary family is the flexible vane pump (sometimes categorized as a flexible impeller pump). These designs have a typical range up to 25 m^3/h (110 gpm) and a maximum pressure capability of 4.1 bar (60 psi). They perform a wide variety of transfer duty applications for low-viscosity fluids up to 22,000 cSt (100,000 SSU). Temperature capabilities typically extend to 90°C (195°F) fluids.

The pump uses an elastomer rotating member with enlarged vane tips that form a pumping chamber in conjunction with a casing when the rotor is placed with the shaft centered in the substantially circular casing that incorporates an eccentric section. Discharge forcing action is accomplished as the vane bends in the eccentric section, effectively squeezing liquid from the discharge chamber. This design is shown in Figure 3.1.4.1.

Flexible vane pump range chart



Because of the variety of applications, stationary components are available in various materials, including stainless steel, bronze, steel, cast iron, and nonmetallics. Flexible members are correspondingly available in a broad range including neoprene, nitrile, EPDM, and Viton®.* Some designs have hygienic certifications, and magnetic drive models are available. This allows a very broad range of industries to be served, from food, beverage, and pharmaceuticals; to chemical and paints; to recreational marine. Flexible vane pumps are typically direct-coupled to the drivers and operate at synchronous motor speeds.

3.1.4.2 Peristaltic

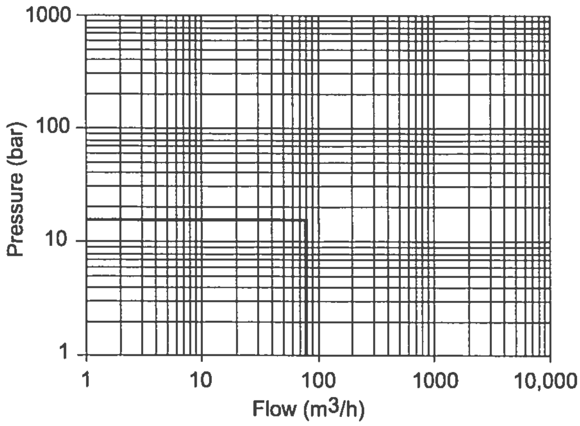
In this type, the fluid pumping and sealing action depends on the elasticity of the flexible member(s). The flexible member may be a tube or a liner. This type of pump is illustrated in Figure 3.1.4.2. The most common type of flexible member pump is the peristaltic pump that has a flexible tube compressed between one or more moving rollers

* Viton® is a registered trademark of DuPont Performance Elastomers.

or shoes and a fixed track. The track is curved and the rollers or shoes rotate about an axis coincident with the center of the radius of curvature of the track. The roller or shoe compresses the tubing and pushes the fluid in front of the roller or shoe towards the discharge end of the tubing. The tubing behind the roller expands to full shape and fills with more fluid. The most common peristaltic pumps have two or three rollers or shoes, which permits closure of the tubing between the suction and discharge ends at all times.

The primary advantage of peristaltic pumps is that the fluid contacts only the tubing. Peristaltic pumps are self-priming, do not require seals and valves, and are reversible. They are used in pharmaceutical, chemical, food, and beverage production, and a number of industrial applications. Small peristaltic pumps are used in various medical applications, and the industrial models can be used for pumping slurries, abrasive fluids, fluids with solids in suspension, and low- to medium-viscosity fluids. Peristaltic pumps are available with flow rates up to 80 m^3/h (350 gpm) and differential pressures to 16 bar (230 psi). The smaller models typically operate at speeds below 200 rpm and the larger models are limited to speeds below 100 rpm.

Peristaltic pump range chart

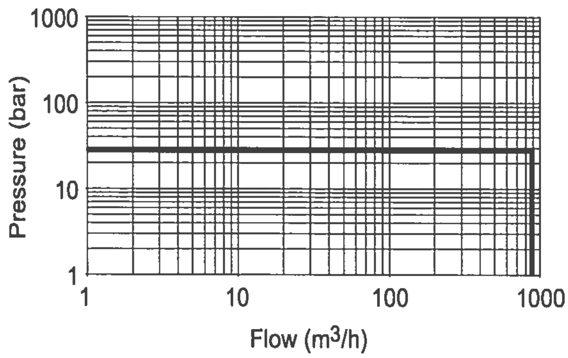


Because only the tubing contacts the fluid, it is available in a variety of materials to ensure compatibility with the fluid being pumped. The life of the tubing depends on the fluid pumped, differential pressure, pump speed, and tubing material.

3.1.5 Lobe

In this design, fluid is carried between rotor lobe surfaces and the pumping chamber from the inlet to the outlet. The rotor surfaces cooperate to provide continuous sealing. The rotors must be timed by separate means. Each rotor has one or more lobes. Figures 3.1.5a and 3.1.5b illustrate a single- and three-lobe pump, respectively.

Lobe pump range chart



Lobe pumps are available in a number of configurations and are used in a variety of applications and industries. They can pump a variety of fluids, including most low- to medium-viscosity fluids such as slurries, solids in suspension, and shear-sensitive fluids. If wetted by injecting fluid into the pumping chamber prior to starting, they can self-prime, operate dry for brief periods of time, and handle relatively large solids. They are frequently used to handle food products because of their ability to handle solids without damaging the product and their ability to be readily cleaned.

Lobe pumps are available in flow rates up to 900 m^3/h (4000 gpm) and can pump fluids with viscosities of 440,000 cSt (2,000,000 SSU). Specific models can operate at temperatures to 177°C (350°F), differential pressures up to 28 bar (400 psi), and can pump fluids with viscosities of 2,000,000 SSU. With smaller lobe pumps (≥47 m^3/h [208 gpm]), speeds of 1000 rpm are possible. As the pump capacity per revolution increases, speeds are reduced. Larger lobe pumps typically operate at speeds of 600 rpm or less, and operating speeds and flow rates are reduced as the fluid viscosity increases.

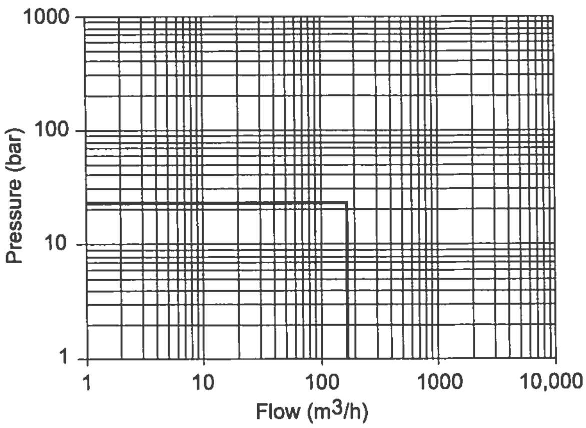
3.1.6 Gear

In this type of pump, fluid is carried between gear teeth and displaced when they mesh. The surfaces of the rotors cooperate to provide continuous sealing and either rotor is capable of driving the other.

3.1.6.1 External gear

The external gear pump is a positive displacement pump composed of a casing with two meshing gears with external teeth. One gear is driven by the shaft coupled to a driver. This gear drives the other gear. The rotation of the gears is such that the liquid comes into the inlet port and flows into and around the outer periphery of the two rotating gears. As the liquid comes around the periphery it is discharged to the outlet port (Figure 3.1.6.1). The flow of the pump is regulated by the size of the cavity (volume) between the teeth and the speed of the gears. Flow from the outlet is further regulated by the amount of liquid that slips back to the inlet port. The amount of slip depends on the side clearance of the gears to the casing, the peripheral clearance of the gear and bore in the casing, gear-to-gear clearance, developed pressure, and viscosity of the liquid. The lower the viscosity, the greater the slippage. Slippage approaches zero at 5000 SSU. As the viscosity increases, the pump speed is lowered to allow the liquid to fill the space between the rotating teeth. Viscosity range is 2 to 400,000 cSt (40 to 2,000,000 SSU).

External gear pump range chart



Most external gear pumps use spur, helical, or herringbone gears. The helical and herringbone gears will deliver more flow and higher pressure. They are quieter than the spur gears but may require more net inlet pressure than a spur gear.

The most common uses for these pumps are to supply fuel oil for burners, gasoline transfer, kerosene, fuel oil, and diesel oil. They are used for hydraulic devices such as elevators and damper controls. They also pump coolants, paints, bleaches, solvents, syrups, glues, lard, greases, asphalt, petroleum, and lube oils and are used in general industrial applications.

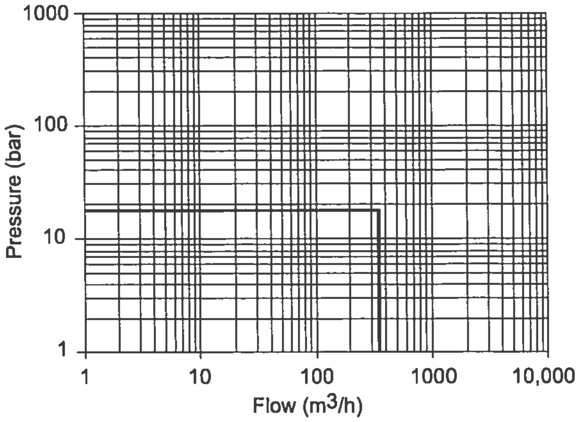
External gear pumps can handle small suspended solids in abrasive applications but will gradually wear and lose performance. Materials of construction are dictated by the application and are available in cast iron, ductile iron, bronze, cast steel, and stainless steel. Because of their broad application scope, numerous optional designs are available.

Rated (normal) performance range is 1 to 180 m^3/h (5 to 800 gpm), 3.5 to 21 bar (50 to 300 psi), and 0.37 to 75 kW (0.5 to 100 hp). Small external gear pumps frequently operate at four-pole motor speeds (1800 rpm) and have operated at two-pole speeds (3600 rpm). As the pump capacity per revolution increases, speeds are reduced to less than 500 rpm. Operating speeds and flow rates are reduced as the fluid viscosity increases.

3.1.6.2 Internal gear

The internal gear pump is a rotary flow positive displacement pump design, which is well-suited for a wide range of applications due to its relatively low speed and inlet pressure requirements. These designs have only two moving parts and hence have proven reliable, simple to operate, and easy to maintain. They are often a more efficient alternative than a centrifugal pump, especially as viscosity increases.

Internal gear pump range chart



Internal gear pumps have one gear with internally cut gear teeth that mesh with the other gear that has externally cut gear teeth. Pumps of this type are made with (Figure 3.1.6.2a) or without (Figure 3.1.6.2b) a crescent-shaped partition. Either gear is capable of driving the other, and the design can be operated in either direction. Designs are available to provide the same direction of flow regardless of the direction of shaft rotation.

As the gears come out of mesh on the inlet side, liquid is drawn into the pump. The gears have a fairly long time to come out of mesh allowing for favorable filling. The mechanical contacts between the gears form a part of the moving fluid seal between the inlet and outlet ports. The liquid is forced out the discharge port by the meshing of the gears.

Internal gear pumps are commercially available in product families with flows from 1 to 340 m^3/h (5 to 1500 gpm) and discharge pressures to 16 bar (230 psi) for applications covering a viscosity range of 2 to 400,000 cSt (40 to 2,000,000 SSU). Internal gear pumps are made to close tolerances and typically contain at least one bushing in the fluid. They can be damaged when pumping large solids. They can handle small suspended solids in abrasive applications but will gradually wear and lose performance. Materials of construction are dictated by the application and include cast iron, ductile iron, bronze, cast steel, and stainless steel.

Small internal gear pumps frequently operate at four-pole motor speeds (1800 rpm) and have operated at two-pole speeds (3600 rpm). As the pump capacity per revolution increases, speeds are reduced. Larger internal gear pumps typically operate below 500 rpm. Operating speeds and flow rates are reduced as the fluid viscosity increases.

Pinion-drive internal gear pumps are a distinctive subclass with unique operating characteristics. They are typically direct-drive arrangements operating at two-, four-, and six-pole speeds for flows below 750 L/min (200 gpm) on clear to very light abrasion, low-viscosity, hydrocarbon-based fluids. They are available in single or multistage module designs capable of pressures to 265 bar (4000 psi).

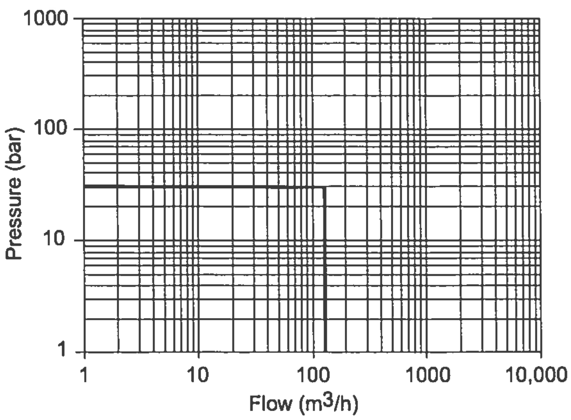
Internal gear pumps are applied in petrochemical, marine, terminal unloading, asphalt, chemical, and general industrial applications for transfer, lubrication, processing, and low-pressure hydraulics handling a wide range of fuel oils, lube oils, and viscous chemicals (both corrosive and noncorrosive). Because of their broad application scope, numerous optional designs are available, such as close-coupled, abrasion resistant, and API Standard compliance considerations.

Figures 3.1.6.2a and 3.1.6.2b illustrate internal gear pumps with and without the crescent-shaped partition, respectively.

3.1.7 Circumferential piston

The circumferential piston pump is a rotary flow positive displacement pump design, which is well-suited for a wide range of applications due to its relatively low speed and inlet pressure requirements and large cavities. These designs have only two moving parts within the fluid chamber and hence have proven reliable. There is no sealing contact between the piston surfaces, which distinguishes this design from gear and screw pumps. External timing gears synchronize the circumferential pistons. As the circumferential piston rotates on the inlet side, the expanding volume draws the liquid into the pump. The liquid is forced out the discharge port by the collapsing cavity on the discharge side.

Circumferential piston pump range chart



Circumferential piston pumps are commercially available in product families with flows to 140 m^3/h (600 gpm) and discharge pressures to 31 bar (450 psi) for applications covering a viscosity range of 50 to 1,000,000 cSt (200 to 4,500,000 SSU). Circumferential piston pumps are made to close tolerances. They can pump almost any product that can be moved and can handle rather large solids and shear-sensitive fluids. They are suitable to run dry for extended periods. Shaft supports often are external from the fluid chamber allowing for higher pressure capabilities. Materials of construction are dictated by the application and include cast iron, ductile iron, cast steel, stainless steel, and many exotic materials.

With smaller circumferential piston pumps speeds of 1800 rpm are possible. Larger circumferential pumps typically operate at speeds of 500 rpm or less. Operating speeds and flow rates are reduced as the fluid viscosity increases.

Circumferential piston pumps are used in petrochemical, paper, marine, wastewater, food processing, tank and terminal unloading, asphalt, chemical, and general industrial applications for transfer and processing handling a wide range of liquids and viscous chemicals (both corrosive and noncorrosive). They are particularly suited for high viscosities, shear-sensitive fluids, and applications that may run dry for a period of time or require higher pressure capability than an internal gear or lobe pump can provide.

Figure 3.1.7 illustrates a circumferential piston pump.

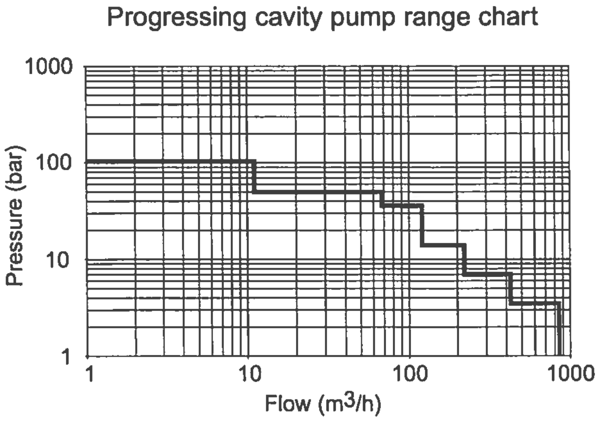
3.1.8 Screw

In this pump type, fluid is carried in spaces formed by the screw(s) and the screw housing and is displaced axially from suction to discharge as they mesh.

3.1.8.1 Single screw (progressing cavity)

Single-screw pumps (commonly called *progressing cavity pumps*) illustrated in Figure 3.1.8.1, have a rotor with external threads and a stator with internal threads. In the simplest form of progressing cavity pump a single-threaded inner member (rotor) rotates inside a double-threaded outer member (stator). The geometry of the rotor and stator are such that cavities are created between the rotor and stator. In each revolution of the rotor two cavities are formed that progress from one end of the rotor and stator pair to the other end. The geometry of the rotor and stator also causes the rotor to rotate eccentric to the axis of rotation. In most progressing cavity pumps the stator is made of an elastomeric material and the rotor is made of a rigid material. The elastomeric stator attaches to the rotor with a compressive fit between the rotor and stator. Progressing cavity pumps are also available with rigid stators that fit on the rotor with a clearance. Progressing cavity pumps with rigid stators are suited for pumping nonabrasive, medium- to high-viscosity fluids at pressures to 200 bar (2900 psi).

Progressing cavity pumps can pump a wide variety of fluids, from less than 1 SSU viscosity to over 2,000,000 SSU viscosity. They can handle fluids containing abrasives and solid particles up to 9 cm (3.5 in.) in diameter, and can handle multiphase fluids with up to 99% gas. They are capable of self-priming and can suction lift fluids up to 8.5 m (28 ft). They can be used to pump practically any fluid that is compatible with the materials of construction. Progressing cavity pumps are available with flow rates over 850 m^3/h (3750 gpm). Standard industrial models are available with differential pressure capabilities up to 70 bar (1040 psi). Models for special applications, such as downhole pumps or viscous fluid applications, have pressure capabilities up to 200 bar (2900 psi) but are usually limited to less than 25 m^3/h (110 gpm) flow rates.



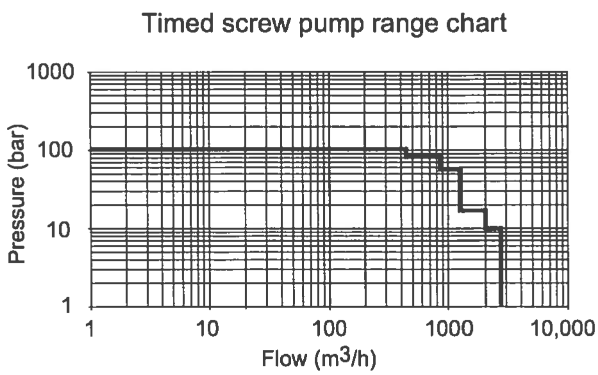
Although some of the smaller progressing cavity pump models operate at speeds up to 1800 rpm, most industrial pump models operate at speeds from 150 to 600 rpm. The low operating speeds and rotor and stator design enable progressing cavity pumps to handle delicate and shear-sensitive fluids without damaging the fluid.

3.1.8.2 Multiple-screw pumps

Multiple-screw pumps have multiple external screw threads. Such pumps may be timed or untimed. Figure 3.1.8.2a illustrates a timed screw pump. Figures 3.1.8.2b and 3.1.8.2c illustrate untimed screw pumps.

3.1.8.2.1 Timed screw pump

Within this family there are a broad range of mechanical configurations, from standard designs to highly customized special units. Unique designs deal with elements as fundamental as screw forms, casing configuration, and timing gear types. Common among the family, however, are two rotating screws positioned by bearing locations and with synchronizing oil-lubricated timing gear elements on both rotors.



Illustrated in Figure 3.1.8.2a is a timed screw pump. Fluid enters at the center inlet, splits axially into two end suction sections, and, as the rotating screws intermesh, chambers are formed trapping and conveying fluid axially to the center discharge of the pump.

Products of this design handle a wide range of viscosity from <2 to 1,000,000 cSt (33 to 4,500,000 SSU). They also have excellent multiphase capabilities and handle typical contaminants such as found in oil production/pipeline applications. They can be manufactured in a broad range of materials, including those for corrosive applications, making them suitable for chemical industry services. Because of the axial movement of the fluid through the pump and the compact diameter of the rotors, timed screw pumps typically operate at motor speeds (two-, four-, and six-pole).

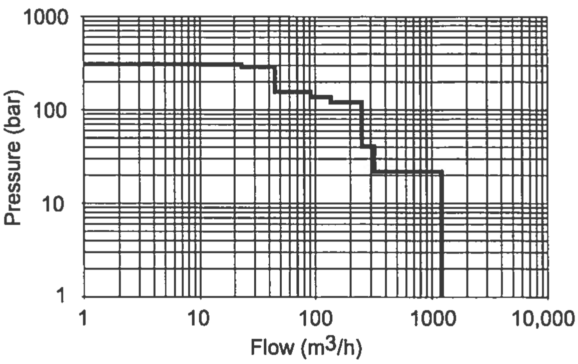
Pressure capabilities to 104 bar (1500 psi) are available for fuel injection and crude oil pipeline services. Flow ranges to 2700 m^3/h (12,000 gpm) are available for marine cargo handling and transfer pump applications.

They pump with a minimum of fluid shear also making them suitable for handling non-Newtonian fluids. Temperature capabilities to 315°C (600°F) qualify them for selected refinery process applications where meeting API Standards is also a requirement. They have extremely low net positive inlet pressure required (NPIPR) capabilities for difficult vapor pressure fluid applications and are frequently found to be the pump technology for difficult service applications.

3.1.8.2.2 Untimed screw pump

The untimed rotary screw pump is an axial-flow, multirotor, positive displacement design used in a wide range of applications in pumping clean to mildly abrasive viscous liquids. It is often a more efficient alternative than centrifugal pumps. The design may use two, three, four, or five screws. The most common configuration is the three-screw pump, which consists of a power rotor (drive screw) and two symmetrically opposed idler rotors (driven screws) that mesh within a close-fitting housing forming a succession of cavities to continuously convey fluid to the pump discharge.

Untimed screw pump range chart



Untimed screw pumps are available with a double-ended flow path as illustrated in Figure 3.1.8.2c or with a single-ended flow path as shown in Figure 3.1.8.2b. Timing is accomplished through rotor geometry. In a properly applied three-screw pump, there is no rotor contact because screws are supported radially in their bores and are hydraulically balanced or free to float on a hydrodynamic film created by the pumped liquid. In other untimed screw pump configurations, the screws may be supported in product-lubricated bushings.

Units are commercially available in product families with flows to 1200 m^3/h (5300 gpm) and discharge pressures to 310 bar (4500 psi). Applications cover a wide viscosity range from 2 to 220,000 cSt (33 to 1,000,000 SSU) and temperatures from below zero to 274°C (500°F). Because of the axial movement of the fluid and the compact diameter of the rotors, untimed screw pumps typically operate at motor speeds (two-, four-, and six-pole). Screw pumps operate with a minimum of noise, vibration, and fluid pulsation. Other characteristics important in many applications are their good suction capability and low shear rate. Untimed screw pumps are frequently found in installations where extended uninterrupted service life is required.

Materials of construction are dictated by the application and product family with options available in aluminum, cast iron, ductile iron, carbon steel, alloy steel, bronze, and corrosion-resistant materials. Hardened components may be offered for mildly abrasive applications in which internal wear is a function of the amount and nature of particulate present in the pumped liquid, materials of construction, and operating conditions.

Screw pumps are used in oil field, pipeline, refinery, marine, power generation, chemical, hydraulic systems, and general industrial applications for transfer, lubrication, injection, and hydraulics handling a wide range of fluids, such as fuel oils, lube oils and greases, asphalts, noncorrosive viscous chemicals, and high-pressure coolants. Because of their broad application scope, numerous standard option packages are available, such as machinery attached, close-coupled designs, magnetically driven, and API compliant versions.

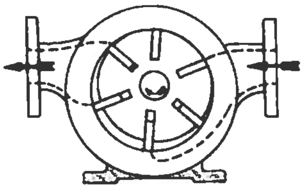


Figure 3.1.2a — Sliding vane pump (vane in rotor)

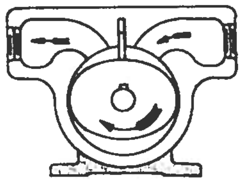


Figure 3.1.2b — External vane pump (vane in stator)

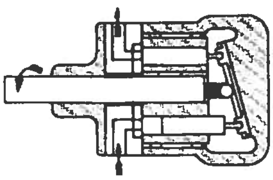


Figure 3.1.3 — Axial piston pump

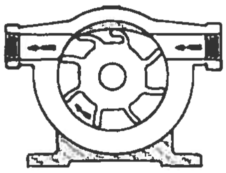


Figure 3.1.4.1 — Flexible vane pump

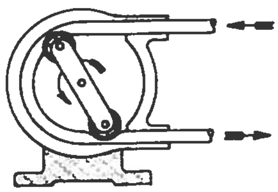


Figure 3.1.4.2 — Flexible tube pump (peristaltic)

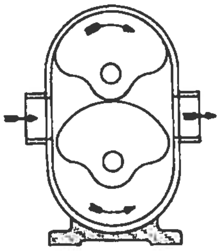


Figure 3.1.5a — Single-lobe pump

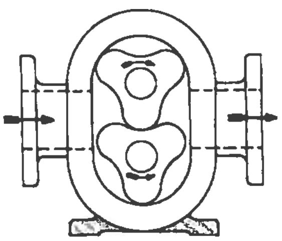


Figure 3.1.5b — Three-lobe pump

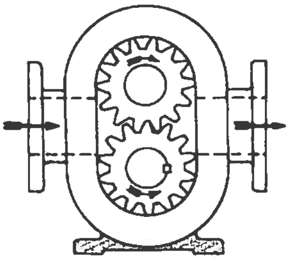


Figure 3.1.6.1 — External gear pump

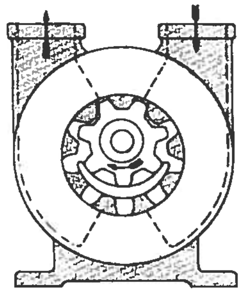


Figure 3.1.6.2a — Internal gear pump (with crescent)

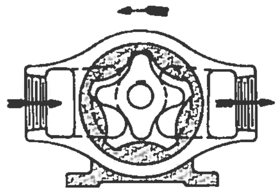


Figure 3.1.6.2b — Internal gear pump (without crescent)

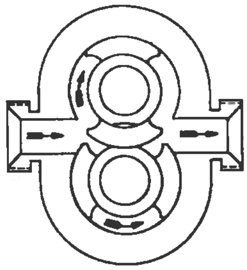


Figure 3.1.7 — Circumferential piston pump

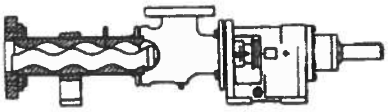


Figure 3.1.8.1 — Single-screw pump (progressing cavity)

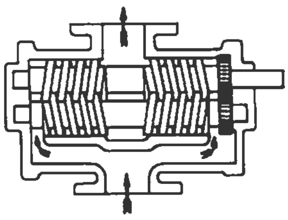


Figure 3.1.8.2a — Two-screw pump (timed)

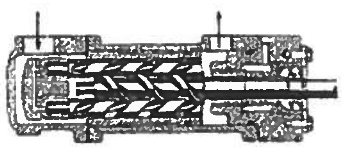


Figure 3.1.8.2b — Three-screw single end pump (untimed)

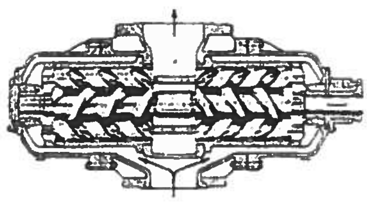


Figure 3.1.8.2c — Three-screw double end pump (untimed)

3.2 Definitions

The nomenclature and definitions in these standards identify the various pump components and provide terminology that will be mutually understandable to the purchaser, manufacturer, and anyone writing specifications for pumps and pumping equipment.

Throughout this Standard, standard terms and units, applicable to the quantities discussed, are defined in the following paragraphs. A summary of the terms, subscripts, symbols, and units used throughout this Standard is shown in Tables 3.2a and 3.2b.

Table 3.2a — Symbols and terminology

Symbol	Term	Metric Unit	Abbr.	US Customary Unit	Abbr.	Conversion Factor*
A	Area	square millimeter	mm^2	square inches	in^2	645.2
D	Displacement	cubic centimeters/revolution	cc/rev	cubic inches/revolution	in^3/rev	16.39
d	Diameter	millimeter	mm	inches	in	25.4
Δ (delta)	Difference	dimensionless	--	dimensionless	--	1
η (eta)	Efficiency	percent	%	percent	%	1
F	Force	Newton	N	pound-force	lbf	4.448
g	Gravitational acceleration	meters/second squared	m/s^2	feet/second squared	ft/s^2	0.3048
γ (gamma)	Specific weight	kilonewtons/cubic meter	kN/m^3	pounds/cubic foot	lb/ft^3	6.37
h	Head	meter	m	feet	ft	0.3048
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPIPA	Net positive inlet pressure avail.	bara	bara	pounds/square inch	psia	0.06897
NPIPR	Net positive inlet pressure required	bara	bara	pounds/square inch	psia	0.06897
ν (nu)	Kinematic viscosity	millimeter squared/sec	mm^2/s	Seconds Saybolt Universal	SSU	0.216 @ 100° F and > 325 SSU†
ν (nu)	Kinematic viscosity	centistokes	cSt	Seconds Saybolt Universal	SSU	0.216 @ 100° F and > 325 SSU†
Π	pi = 3.1416	dimensionless	--	dimensionless	--	1
p	Pressure	bar	bar	pounds/square inch	psi	0.06897
P	Power	kilowatt	kW	horsepower	hp	0.7457
Q	Rate of flow (capacity)	cubic meter/hour	m^3/h	US gallons/minute	gpm	0.2271
s	Specific gravity	dimensionless	--	dimensionless	--	1
t	Temperature	degrees Celsius	°C	degrees Fahrenheit	°F	$(°F-32) \times \frac{5}{9}$
τ (tau)	Torque	newton-meter	N•m	pound-feet	lb-ft	1.356
v	Velocity	meters/second	m/s	feet/second	ft/s	0.3048
x	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	meter	m	feet	ft	0.3048

* Conversion factor × US units = metric units

† Refer to ASTM 1261 for < 70 cSt

Table 3.2b — Subscripts

Subscript	Term	Subscript	Term
a	Absolute	mot	Motor
b	Barometric	OA	Overall unit
d	Outlet (discharge)	p	Pump
g	Gauge	s	Inlet (suction)
im	Intermediate mechanism	t	Theoretical
m	Metering	v	Velocity, volumetric
max	Maximum	vp	Vapor pressure
min	Minimum	w	Water

3.2.1 Fluids and liquids

In these standards, the term *fluid* covers liquids, gases, vapors, and mixtures thereof. The word *liquid* is used only to describe true liquids that are free of vapors and solids. The word *fluid* is more general and is used to describe liquids that may contain, or be mixed with, matter in other than the liquid phase.

3.2.2 Pumping chamber

The pumping chamber or cavity is the space formed by the body and end plate(s), into which fluid is drawn and from which fluid is discharged by the action of the rotor(s).

3.2.3 Inlet or suction port

One or more openings in the pump through which the pumped fluid may enter the pumping chamber.

3.2.4 Outlet or discharge port

One or more openings in the pump through which the pumped fluid may leave the pumping chamber.

3.2.5 Body

The body is an external part which surrounds the periphery of the pumping chamber and which also may form one end plate. It is sometimes called a *casing* or a *housing*.

3.2.6 End plate

An end plate is a part that closes an end of the body to form the pumping chamber. One or more are used, depending on the construction of the pump. It is sometimes called a *head* or *cover*.

3.2.7 Stator

The stationary parts of the pump which surround the pumping chamber.

3.2.8 Rotor

A rotor is a part that rotates in the pumping chamber. One or more are used per pump. It is sometimes referred to by a specific name, such as *gear, screw, impeller*, etc.

3.2.9 Bearing

A bearing is a part that supports or positions the shafts on which a rotor is mounted. A bearing may be internal (wetted by the liquid being pumped) or external and may be either a rolling element bearing (ball or roller) or fluid film type (sleeve and journal).

3.2.10 Timing gear

A timing gear is a part used to transmit torque from one rotor shaft to another and to maintain the proper angular relationship of the rotors. It may be outside the pumping chamber and is sometimes called a *pilot gear*.

3.2.11 Rotating assembly

The rotating assembly generally consists of all rotating parts essential to the pumping action but also may include other parts specified by the manufacturer.

3.2.12 Relief valve

A relief valve is a mechanism designed to control or to limit pressure by the opening of an auxiliary passage at a predetermined pressure.

A relief valve may be either integral with the body or end plate or attachable. It may be adjustable through a predetermined range of pressures or have a fixed setting. It may be designed to bypass the liquid internally from the pump outlet to the pump inlet or externally through an auxiliary port. Bypass of liquid internally is not recommended for continuous operation.

Terms commonly used in specifying relief valve performance are shown in Sections 3.2.12.1 through 3.2.12.4.

3.2.12.1 Cracking pressure

Sometimes called *set pressure, start-to-discharge pressure,* or *popping pressure*—the pressure at which the relief valve just starts to open. This pressure cannot be determined readily in a valve that bypasses the liquid within the pump.

3.2.12.2 Full-flow bypass pressure

The pressure at which the full output of the pump flows through the valve and the auxiliary passage.

3.2.12.3 Reseating pressure

The pressure at which the relief valve is closed completely. This pressure is usually below the cracking pressure and is difficult to measure accurately when the liquid is bypassed within the pump.

3.2.12.4 Percent overpressure

Sometimes called *percent accumulation* or *percent regulation*—the difference between full bypass pressure and cracking pressure, expressed as a percent of cracking pressure.

3.2.13 Stuffing box

A stuffing box is a cylindrical cavity through which a shaft extends and in which leakage at the shaft is controlled by means of packing and a gland or a mechanical seal. (See 3.2.17 Seal chamber.)

3.2.14 Gland

A gland is a part that may be adjusted to compress packing in a stuffing box. It is sometimes called a *gland follower.* A gland is also used to hold the stationary element of a mechanical seal.

3.2.15 Packing

A pliable lubricated material used to provide a seal around that portion of the shaft located in the stuffing box (see Figure 3.2.39k).

3.2.16 Lantern ring

A lantern ring is an annular ring located in a stuffing box to provide space between or adjacent to packing rings for the introduction of a lubricant or a barrier fluid, the circulation of a cooling medium, or the relief of pressure against the packing. It is sometimes called a *seal cage*.

3.2.17 Seal chamber

A seal chamber is a cavity through which a shaft extends and in which pumped liquid is contained by means of a mechanical seal or radial seal. A seal chamber is designed specifically for a mechanical seal or radial seal and usually cannot be fitted with packing. (See 3.2.13 Stuffing box.)

3.2.18 Mechanical seal

A mechanical seal is a device located in a seal chamber or stuffing box and consists of rotating and stationary elements with opposed seal faces. A rotating element is fastened and sealed to the shaft. A stationary element is mounted and sealed to the gland or body. At least one element is loaded in an axial direction, so that the seal faces of the elements are maintained in close proximity to each other at all times. In a typical seal design, the seal faces are flat, highly lapped surfaces on materials selected for low friction and resistance to corrosion by the fluids to be pumped. Mechanical seals are sometimes called *face-type seals*. Detailed information on mechanical seals is available in *Mechanical Seals for Pumps: Application Guideline* published by the Hydraulic Institute.

3.2.19 Radial seal

A radial seal is a device located in a seal chamber that seals on its outside diameter through an interference fit with its mating bore and on the rotating shaft with a flexible, radially loaded surface. Radial seals include: lip-type seals, O-rings, V-cups, U-cups, etc., and may or may not be spring-loaded.

3.2.20 Direction of rotation

Driveshaft rotation is designated as "clockwise" (CW) or "counterclockwise" (CCW) as determined when facing the shaft end of the pump.

3.2.21 Jacketed pump

A jacketed pump is one in which the body and/or end plates incorporate passageways through which steam, oil, water, or other fluid can be circulated to control the temperature of the pump or the fluid in the pump.

3.2.22 Rate of flow (*Q*)

The rate of flow of a rotary pump is the volumetric quantity of fluid actually delivered per unit of time, including both the liquid and any dissolved or entrained gases, under stated operating conditions.

In the absence of any vapor entering or forming within the pump, rate of flow is equal to the volume displaced per unit of time, minus slip (see Section 3.2.26).

3.2.23 Displacement (*D*)

The displacement of a rotary pump is the volume displaced per revolution of the rotor(s). It may be calculated from the physical dimensions of the pumping elements, or it may be determined empirically as the volume of liquid pumped per revolution at zero differential pressure. In pumps incorporating one or more rotors operating at different speeds, it is the volume displaced per revolution of the driving rotor. The standard unit of displacement is the cubic centimeter (cubic inch) per revolution. A variable displacement pump shall be rated at its maximum displacement.

3.2.24 Speed (*n*)

Speed is the number of revolutions per minute of the pump shaft. The proper rotating speed of a rotary pump in rpm is dependent on: (1) the type and the size of the pump, (2) the characteristics of the fluid pumped, and (3) the inlet and outlet pressures. Operation of rotary pumps at speeds higher than those approved for the particular application by the pump manufacturer is not recommended.

3.2.25 Pump volumetric efficiency (η_v)

The pump volumetric efficiency is a ratio of the actual pump rate of flow to the volume displaced per unit of time. The formula for computing volumetric efficiency in percent is:

$$\text{(Metric) } \eta_v = \frac{10^6 Q}{60 Dn} \times 100$$

$$\text{(US units) } \eta_v = \frac{231 Q}{Dn} \times 100$$

3.2.26 Slip (*S*)

Slip is the quantity of fluid that leaks through internal clearances of a pump per unit of time. It is dependent upon the internal clearances, the differential pressure, the characteristics of the fluid handled, and, in some cases, the speed.

3.2.27 Pressure (*p*)

Pressure is the compressive stress in a liquid at a given point. It has the units of force per unit area.

3.2.27.1 Gauge pressure

Gauge pressure is the pressure of a fluid relative to the local atmospheric or ambient pressure, as determined by a pressure gauge.

3.2.27.2 Absolute pressure

Absolute pressure (p_a) is the algebraic sum of gauge pressure (p_g) and barometric pressure (p_b).

3.2.27.3 Datum

The pump's datum is the reference plane or centerline of the pump inlet from which the elevations of gauges are measured. The datum serves as the reference of pressure measurements taken during test.

3.2.27.4 Elevation pressure (p_z)

The potential energy of the fluid due to the elevation of the gauge above or below datum, expressed as equivalent pressure. It will be positive when the gauge is above the datum and negative when the gauge is below the datum.

(Metric) $p_z = 9.8sZ$

(US units) $p_z = 0.433sZ$

3.2.27.5 Velocity pressure (p_v)

The velocity pressure is the kinetic energy of the liquid flow expressed in equivalent pressure. It is defined by the expression:

$$\text{(Metric)}\quad p_v = \frac{9.8 \times 10^{-2} s v^2}{2g}$$

$$\text{(US units)}\quad p_v = \frac{0.433 s v^2}{2g}$$

Where:

s = specific gravity (ref. 3.3.8) of the fluid pumped

$v^2/2g$ units = meters of water (metric) or feet of water (US units)

$0.433s$ = the conversion factor for feet of water to psi of liquid pumped

The mean velocity (v) is calculated for incompressible flow in the pipe where the gauge is connected by:

$$\text{(Metric)}\quad v = \frac{278\,Q}{A} = \frac{354\,Q}{d^2}$$

$$\text{(US units)}\quad v = \frac{0.321\,Q}{A} = \frac{0.4085\,Q}{d^2}$$

Where:

d = diameter of the pipe

A = the inside cross-sectional area of the pipe

3.2.28 Pump pressures

3.2.28.1 Maximum allowable casing pressure

This is the highest pressure at the maximum pumping temperature for which the pump casing components are designed. The maximum allowable casing pressure shall be greater than the maximum allowable working pressure and rated pressure by a suitable factor of safety. The maximum allowable casing pressure for the suction and discharge sides of the pump may be different.

3.2.28.2 Outlet (discharge) pressure (p_d)

The discharge pressure at the pump outlet is the algebraic sum of the measured gauge pressure (p_g), the velocity pressure (p_v) at the point of gauge attachment, and the elevation pressure (p_z) from the discharge gauge centerline to the pump datum.

$$p_d = p_g + p_v + p_z$$

$$\text{(Metric)} \quad p_d = p_g + 9.8 \times 10^{-2} s\left[Z_d + \frac{v_d^2}{2g}\right]$$

$$\text{(US units)} \quad p_d = p_g + 0.433 s\left[Z_d + \frac{v_d^2}{2g}\right]$$

For tests, p_d is equal to gauge pressure at the pump outlet, which is p_g, providing the gauge is within ± 0.75 m (2.5 ft) elevation of the inlet gauge and pipe velocity is less than 4.5 m/s (15 ft/s).

The measuring section should be located in the outlet pipe immediately after the pump outlet connection.

3.2.28.3 Maximum allowable working (operating) pressure

The maximum allowable working pressure is the outlet pressure established by the manufacturer for satisfactory operation under specified application conditions.

3.2.28.4 Inlet pressure (p_s)

The inlet pressure is the algebraic sum of the gauge pressure, the velocity pressure, and elevation pressure as measured at the pump inlet:

$$\text{(Metric)} \quad p_s = p_g + 9.8 \times 10^{-2} s\left[Z_s + \frac{v_s^2}{2g}\right]$$

$$\text{(US units)} \quad p_s = p_g + 0.433 s\left[Z_s + \frac{v_s^2}{2g}\right]$$

For the tests, p_s is equal to gauge pressure at the pump inlet, which is p_g, providing the gauge is within ± 0.75 m (2.5 ft) elevation of the outlet gauge and pipe velocity is less than 4.5 m/s (15 ft/s).

The symbol (p_s) may be positive or negative with reference to atmospheric pressure and may, therefore, have positive or negative values. The symbol is called *inlet pressure* when positive and *inlet vacuum* when negative. The measuring section should be located in the inlet pipe immediately before the pump inlet connection.

3.2.28.5 Maximum allowable inlet working pressure

The maximum allowable working inlet pressure is established by the manufacturer for satisfactory operation of the pump. It is typically based on the pump's design limits.

3.2.29 Differential pressure (Δp)

The differential pressure is the algebraic difference of the outlet pressure and inlet pressure, with terms expressed in like units:

$$\Delta p = p_d - p_s$$

3.2.30 Maximum differential pressure (Δp_{max})

The maximum differential pressure is the maximum algebraic difference between the outlet pressure and the inlet pressure.

3.2.31 Net positive inlet pressure available (*NPIPA*)

Net positive inlet pressure available is the algebraic sum of the inlet and barometric pressure minus the vapor pressure (see Sections 3.3.9 and 3.3.10) of the liquid at the inlet temperature:

$$NPIPA = p_s + p_b - p_{vp}$$

$$NPIPA\ (\text{psi}) = \frac{[NPSHA\ (\text{ft}) \times \text{specific gravity}]}{2.31}$$

This value must be equal to or greater than the net positive inlet pressure required (NPIPR) as established by the pump manufacturer for the speed, pressure, and fluid characteristics that exist; otherwise, the rate of flow will be reduced, and operation may be noisy and rough due to incomplete filling of the pump. This condition may damage the pump and associated equipment.

It must be recognized that many rotary pumps can stably and quietly operate at rate of flow reductions of 20 to 35% due to low NPIPA with no ill effect. Many services employ this characteristic to operate high-vacuum systems for extraction of gas or light liquids.

3.2.32 Net positive inlet pressure required (*NPIPR*)

Net positive inlet pressure required is the pressure required, above liquid vapor pressure (see Section 3.3.9), to fill each pumping chamber or cavity while open to the inlet chamber. It is expressed in bar (psi). NPIPR is sometimes called *NPSH3* for rotodynamic pumps.

$$NPIPR\ (\text{psi}) = \frac{[NPSHR\ (\text{ft}) \times \text{specific gravity}]}{2.31}$$

Many liquids handled by rotary pumps have an unpredictable or very low vapor pressure. Most of these liquids have entrained and dissolved gas (frequently air) as well. The practical effect of dissolved and entrained gas is to increase the NPIPR in order to suppress the symptoms of cavitation. While true cavitation occurs if the liquid reaches its vapor pressure during filling of the pumping cavities, most of the cavitation symptoms will be exhibited before reaching liquid vapor pressure. This is largely due to the entrained and dissolved gas expanding when subjected to reduced pressure. Because the level of dissolved gas is a function of the liquid and its temperature, and the level of entrained gas is a function of system design and operation, NPIPR for a rotary pump is difficult to establish with precision.

NPIPR tests are normally conducted by the manufacturer in a test environment that minimizes entrained gas using a test liquid of negligible vapor pressure. NPIPR is established at the first indication of any of the following:

A 5% reduction in output rate of flow at constant differential pressure and speed;

A 5% reduction in power consumption at constant differential pressure and speed;

The inability to maintain a stable differential pressure and speed;

The onset of loud or erratic noise when this criterion is previously agreed upon by all parties.

3.2.33 Power (P)

The work requirement per unit of time to operate the pump.

3.2.34 Pump input power (P_p)

The mechanical power delivered to a pump input shaft.

3.2.35 Pump output power (P_w)

The power imparted to the liquid by the pump. It is also referred to as *water horsepower* or *liquid horsepower.*

(Metric) $P_W = \frac{(Q \times \Delta p)}{36}$

(US units) $P_W = \frac{(Q \times \Delta p)}{1714}$

3.2.36 Pump torque

There are two torque values associated with rotary pumps: breakaway or starting torque, and running torque. The breakaway torque is the turning force or moment that causes initial rotation of the pump shaft. The running torque is the turning force or moment that maintains rotation of the pump shaft at the speed, fluid, and pressure conditions required by the application.

3.2.37 Pump efficiency (η_p)

The pump efficiency is the ratio of the pump output power (P_w) to the pump input power (P_p).

The formula for computing pump efficiency as a percent is:

$$\eta_p = \frac{P_w}{P_p} \times 100$$

3.2.38 Multiphase

This is a term used to describe a fluid stream composed of a mixture of fluid states. Multiphase fluids are typically liquids and gas. In oil field applications, such as described in Section 3.3.18, the multiphase fluid is typically a mixture of gas, oil, and water. Multiphase pump systems are designed to handle fluid streams ranging from 100% liquid to 99% gas and 1% liquid.

3.2.38.1 Gas volume fraction (GVF)

GVF, also called *gas fraction*, is the relationship of gas content at pump inlet pressure and temperature to the total flow stated in percentage or as a fraction.

3.2.38.2 Gas oil ratio (GOR)

GOR is the volume of gas at standard conditions per barrel of oil.

3.2.38.3 Water cut (WC)

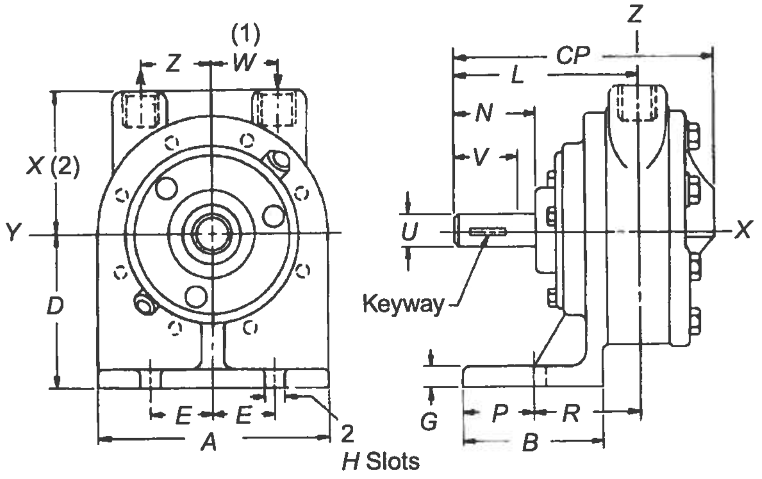
WC is the percentage of water in the combined oil and water mixture.

3.2.38.4 Standard conditions

Standard conditions of pressure and temperature are 1.01 bar and 15°C (14.7 psia and 60°F).

3.2.39 Letter (dimensional) designations

The letter designations used on the following drawings (Figures 3.2.39a–k) were prepared to provide a common means for identifying various pump dimensions and to serve as a common language that will be mutually understandable to the purchaser, manufacturer, and anyone writing specifications for pumps and pumping equipment.



1) If equal to "Z", use "Z". These do not change for horizontal nozzles.

2) If not equal for suction and discharge, use *Y* for suction and show separately.

Figure 3.2.39a — Internal gear pump (foot mounting)

A Width of base support or foot or width of pump.

AF Mounting hole bolt circle.

AJ Diameter, mounting pilot fit.

B Length of base support or foot.

CP Length of pump.

D Vertical height – centerline of pump shaft to bottom of base support or foot.

E Distance – centerline of pump shaft to centerline of hold-down bolt hole or slot.

G Thickness of pads on support, or height of base plate, depending on location of bolt holes or slots.

H Diameter of hold-down or mounting bolt holes or slots.

L Distance – centerline of outlet port to end of pump shaft.

LP Length of adapter piece or length of pilot fit.

N Distance – end of shaft to nearest obstruction.

P Length from edge of support or base plate to centerline or bolt hole.

R Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.

S Distance – end of pump shaft to face of mounting flange.

U Diameter of straight shaft – coupling end.

V Length of shaft available for coupling or pulley.

W Distance – centerline of shaft to centerline of inlet port.

X Distance – centerline of pump shaft to outlet port.

Z Distance – centerline of pump shaft to centerline of ports.

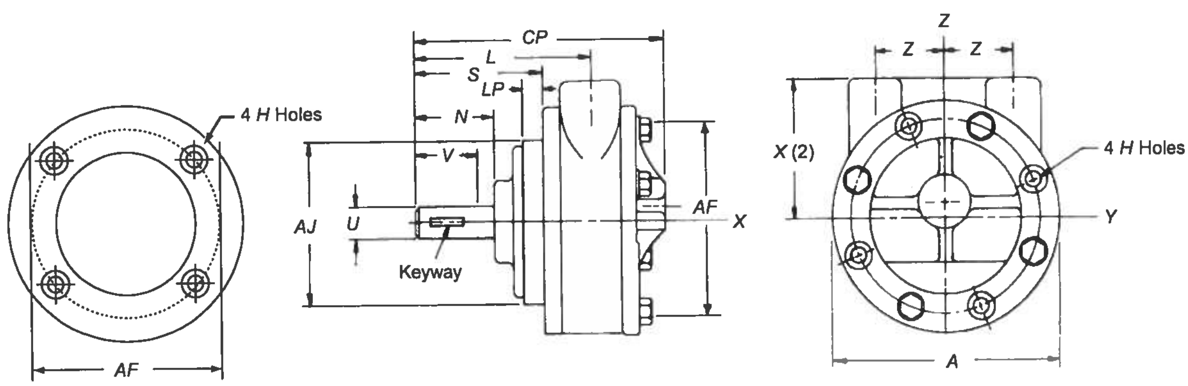


Figure 3.2.39b — Internal gear pump (flange mounting)

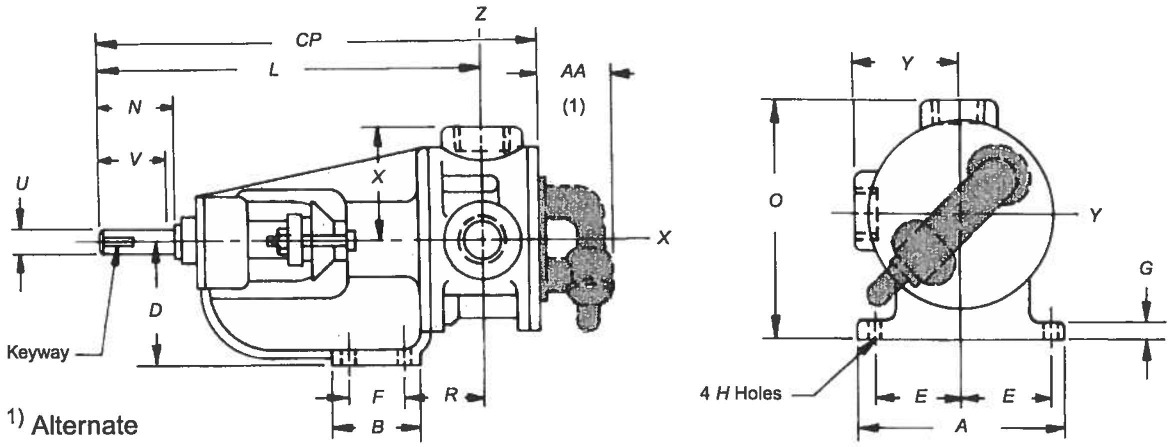
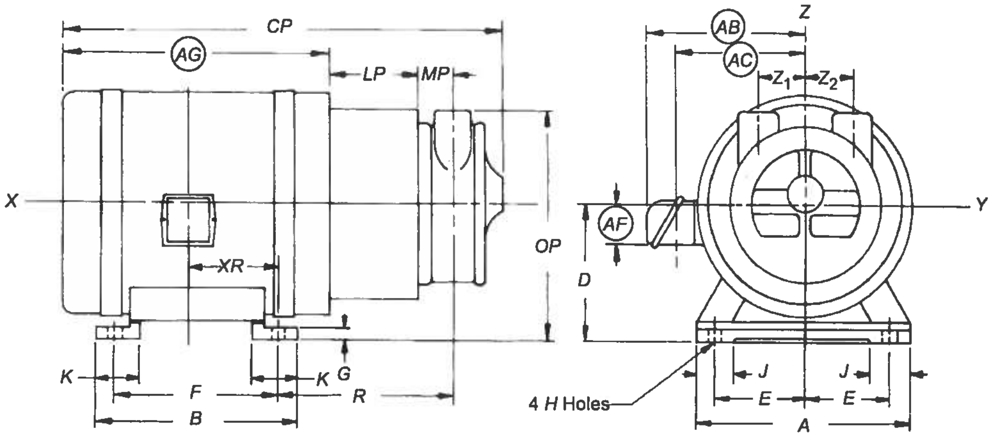


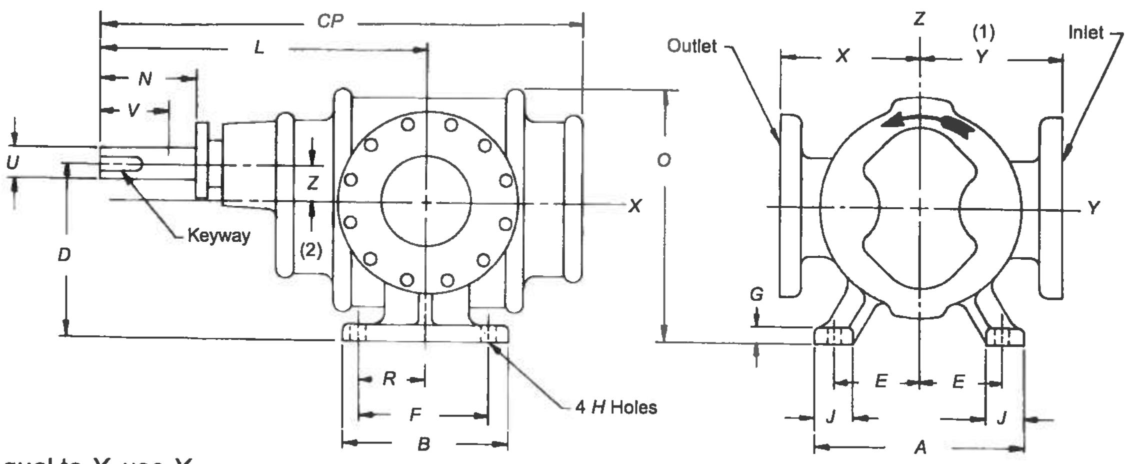
Figure 3.2.39c — Internal gear pump (frame mounting)

A Width of base support or foot or width of pump.
AA Added length for alternate feature.
B Length of base support or foot.
CP Length of pump.
D Vertical height – centerline of pump shaft to bottom of base support or foot.
E Distance – centerline of pump shaft to centerline of hold-down bolt hole or slot.
F Distance – centerline to centerline of hold-down bolt holes.
G Thickness of pads on support, or height of base plate, depending on location of bolt holes or slots.
H Diameter of hold-down or mounting bolt holes or slots.
J Width of pads for hold-down bolts.
K Length of pads for hold-down bolts.
L Distance – centerline of outlet port to end of pump shaft.
LP Length of adapter piece or length of pilot fit.
MP Distance – centerline of outlet port to mounting flange of adapter.
N Distance – end of shaft to nearest obstruction.
O Vertical distance – bottom of support to outlet port or top of pump.
R Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.
U Diameter of straight shaft – coupling end.
V Length of shaft available for coupling or pulley.
X Distance – centerline of pump shaft to outlet port.
XR Distance – centerline of motor hold-down bolt to centerline of conduit box.
Y Distance – centerline of pump shaft to inlet port.
Z Distance – centerline of pump shaft to centerline of ports.



NOTE: Circled letters are NEMA designations. If Z_1 and Z_2 are not equal, substitute W_1 for Z_1, the distance from the centerline of pump to centerline of inlet.

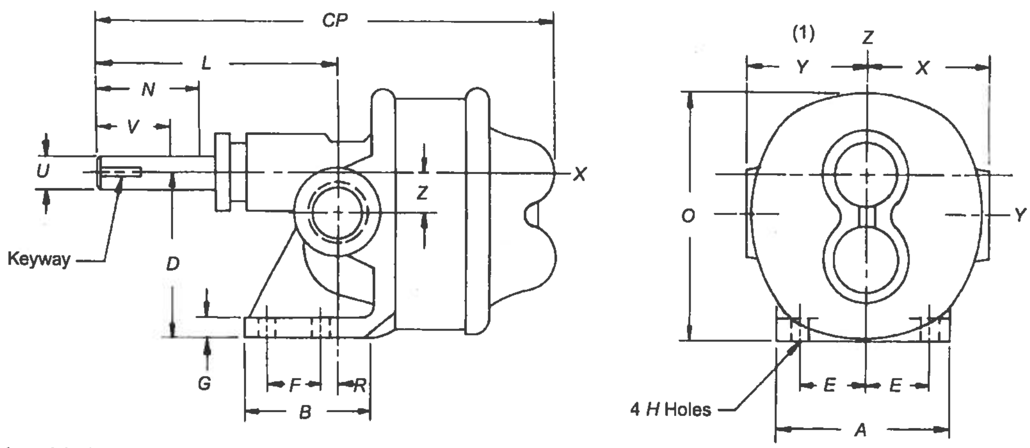
Figure 3.2.39d — Internal gear pump (close coupled)



1) If equal to *X*, use *X*.
2) If not equal for inlet and outlet, use *W* for inlet and show both in end view.

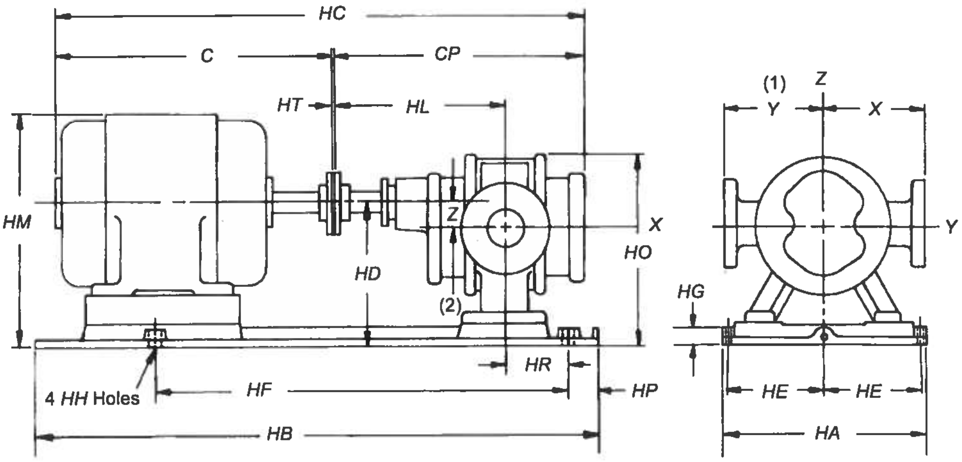
Figure 3.2.39e — External gear pump (flanged ports)

A Width of base support or foot or width of pump.
B Length of base support or foot.
CP Length of pump.
D Vertical height – centerline of pump shaft to bottom of base support or foot.
E Distance – centerline of pump shaft to centerline of hold-down bolt hole or slot.
F Distance – centerline to centerline of hold-down bolt holes.
G Thickness of pads on support, or height of base plate, depending on location of bolt holes or slots.
H Diameter of hold-down or mounting bolt holes or slots.
J Width of pads for hold-down bolts.
L Distance – centerline of outlet port to end of pump shaft.
N Distance – end of shaft to nearest obstruction.
O Vertical distance – bottom of support to outlet port or top of pump.
R Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.
U Diameter of straight shaft – coupling end.
V Length of shaft available for coupling or pulley.
X Distance – centerline of pump shaft to outlet port.
Y Distance – centerline of pump shaft to inlet port.
Z Distance – centerline of pump shaft to centerline of outlet ports.



1) If equal to *X*, then use *X*.

Figure 3.2.39f — External gear pump (threaded ports)



1) If equal to *X*, then use *X*.
2) If not equal for inlet and outlet, then use *W* for inlet and show both in end view.

Figure 3.2.39g — External gear pump on base plate

C Length of driver.

CP Length of pump.

HA Width of base support or foot or width of pump.

HB Length of base support or foot.

HC Overall length of combined pump and driver when on base.

HD Vertical height – centerline of pump shaft to bottom of base support or foot.

HE Distance – centerline of pump shaft to centerline of hold-down bolt hole.

HF Distance – centerline to centerline of hold-down bolt holes.

HG Thickness of pads on support, or height of base plate, depending on location of bolt holes.

HH Diameter of hold-down or mounting bolt holes.

HL Distance – centerline of outlet port to end of pump shaft.

HM Height of unit – bottom of base to top of driver.

HO Vertical distance – bottom of support base or foot to top of pump.

HP Distance from end of support or base plate to centerline of bolt hole.

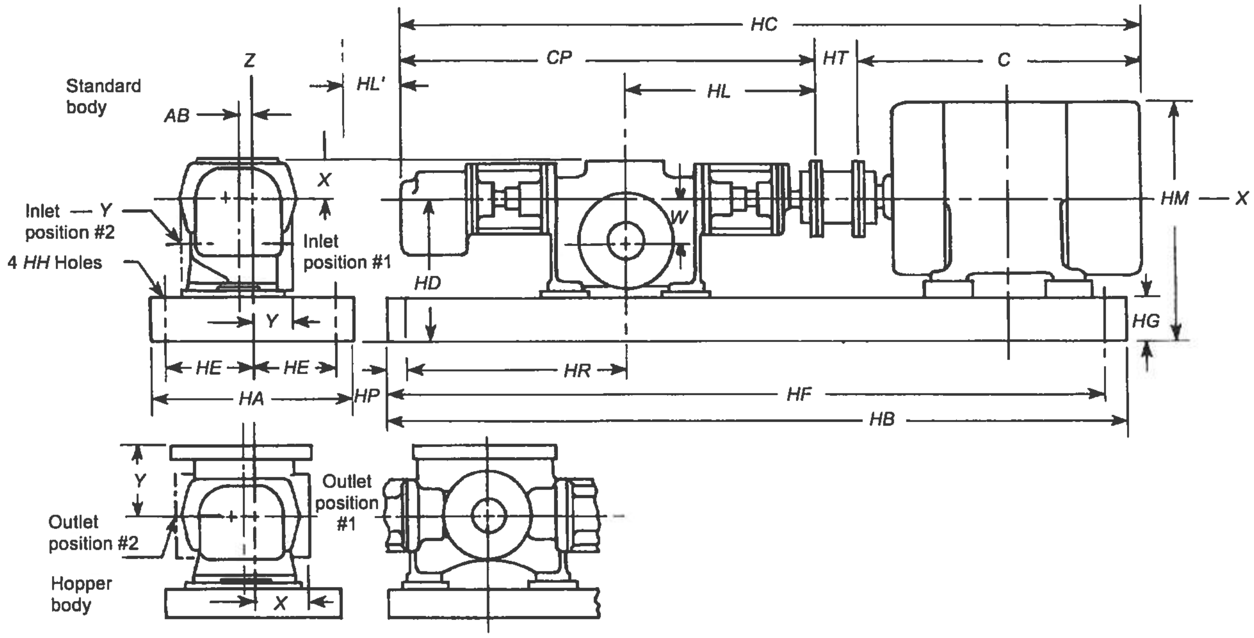
HR Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.

HT Horizontal distance – between pump and driving shaft.

X Distance – centerline of pump shaft to outlet port.

Y Distance – centerline of pump shaft to inlet port.

Z Distance – centerline of pump shaft to centerline of ports.



NOTE: Inlet or outlet positions = 2 are optional alternate locations

Figure 3.2.39h — External gear and bearing screw pump on base plate

AB Distance – pump centerline to pump shaft centerline.

C Length of driver.

CP Length of pump.

HA Width of base support or foot or width of pump.

HB Length of base support or foot.

HC Overall length of combined pump and driver when on base.

HD Vertical height – centerline of pump shaft to bottom of base support or foot.

HE Distance – centerline of pump shaft to centerline of hold-down bolt hole.

HF Distance – centerline to centerline of hold-down bolt holes.

HG Thickness of pads on support, or height of base plate, depending on location of bolt holes.

HH Diameter of hold-down or mounting bolt holes.

HL Distance – centerline of outlet port to end of pump shaft.

HL' Length required for removal of rotor (*HL* + ½ body length).

HM Height of unit – bottom of base to top of driver.

HP Distance from end of support, or base plate to centerline of bolt hole.

HR Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.

HT Horizontal distance – between pump and driving shaft.

W Distance – centerline of pump shaft to centerline of inlet port.

X Distance – centerline of pump shaft to outlet port.

Y Distance – centerline of pump shaft to inlet port.

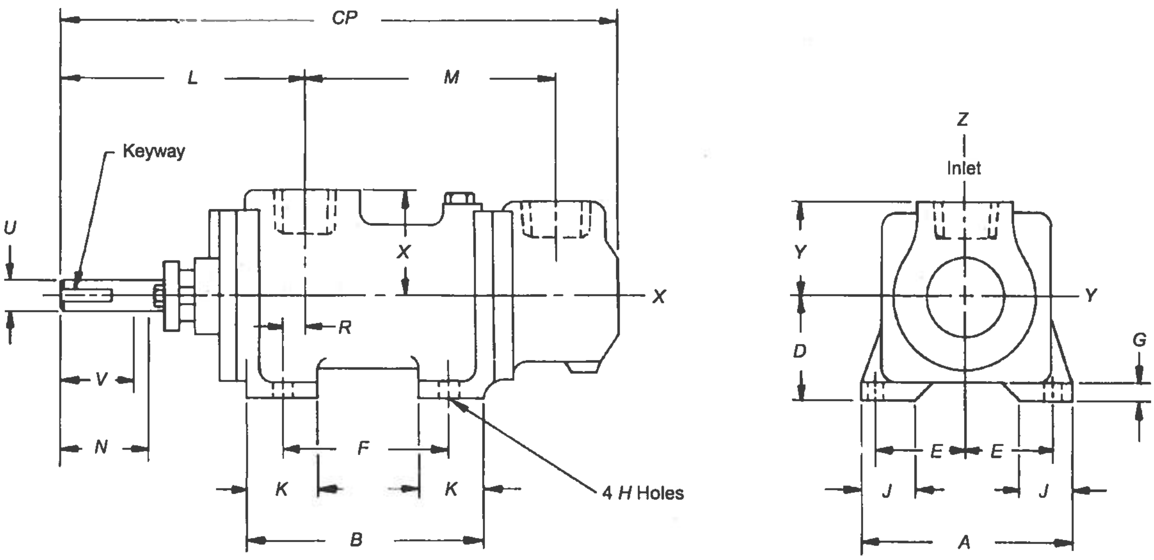
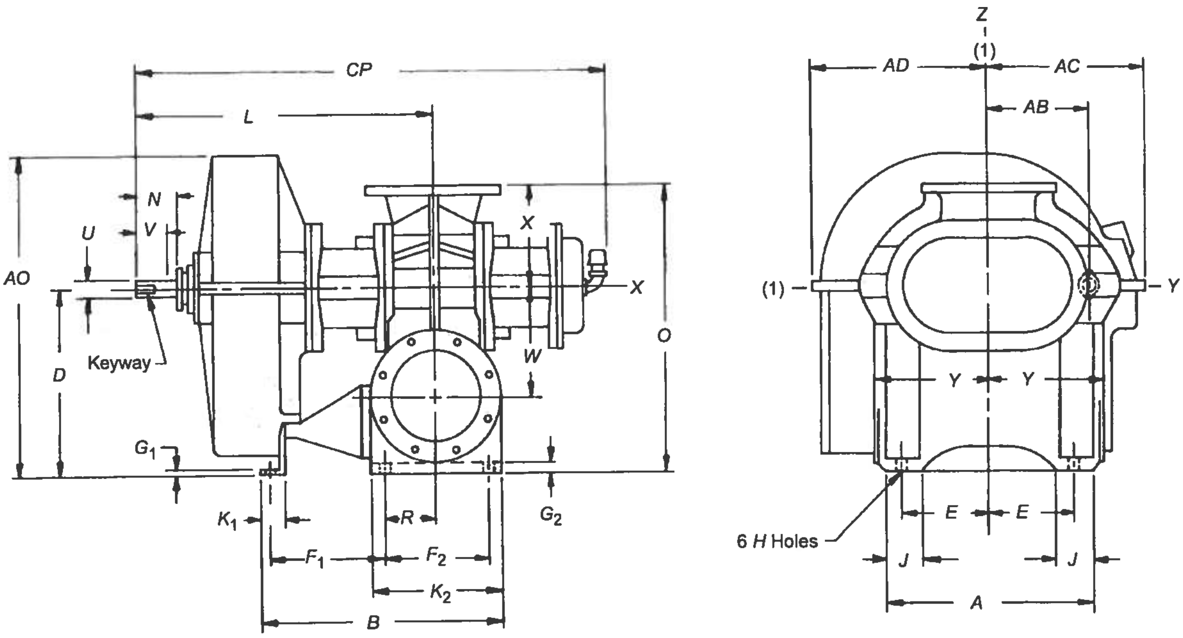


Figure 3.2.39i — Multiple screw pump

A	Width of base support or foot, or width of pump.
B	Length of base support or foot.
CP	Length of pump.
D	Vertical height – centerline of pump shaft to bottom of base support or foot.
E	Distance – centerline of pump shaft to centerline of hold-down bolt hole or slot.
F	Distance – centerline to centerline of hold-down bolt holes.
G	Thickness of pads on support, or height of base plate, depending on location of bolt holes or slots.
H	Diameter of hold-down or mounting bolt holes or slots.
J	Width of pads for hold-down bolts.
K	Length of pads for hold-down bolts.
L	Distance – centerline of outlet port to end of pump shaft.
M	Horizontal distance from centerline to centerline of ports.
N	Distance – end of shaft to nearest obstruction.
R	Horizontal distance—centerline outlet port to centerline of hold-down bolt hole.
U	Diameter of straight shaft – coupling end.
V	Length of shaft available for coupling or pulley.
X	Distance – centerline of pump shaft to outlet port.
Y	Distance – centerline of pump shaft to inlet port.



1) Use these centerlines instead of centerline of pump shaft because some models have alternate shaft positions.

Figure 3.2.39j — Lobe pump

A Width of base support or foot or width of pump.

AB Distance – pump centerline to shaft centerline.

AC Distance – pump centerline to left side of gear box or other structure.

AD Distance – pump centerline to right side of gear box or other structure.

AO Height, overall, to top of gear or other structure.

B Length of base support or foot.

CP Length of pump.

D Vertical height – centerline of pump shaft to bottom of base support or foot.

E Distance – centerline of pump shaft to centerline of hold-down bolt hole or slot.

F Distance – centerline to centerline of hold-down bolt holes.

G Thickness of pads on support, or height of base plate, depending on location of bolt holes or slots.

H Diameter of hold-down or mounting bolt holes or slots.

J Width of pads for hold-down bolts.

K Length of pads for hold-down bolts.

L Distance – centerline of outlet port to end of pump shaft.

N Distance – end of shaft to nearest obstruction.

O Vertical distance – bottom of support to outlet port or top of pump.

R Horizontal distance – centerline outlet port to centerline of hold-down bolt hole.

U Diameter of straight shaft – coupling end.

V Length of shaft available for coupling or pulley.

W Distance – centerline of pump to centerline of inlet port.

X Distance – centerline of pump shaft to outlet port.

Y Distance – centerline of pump shaft to inlet port.

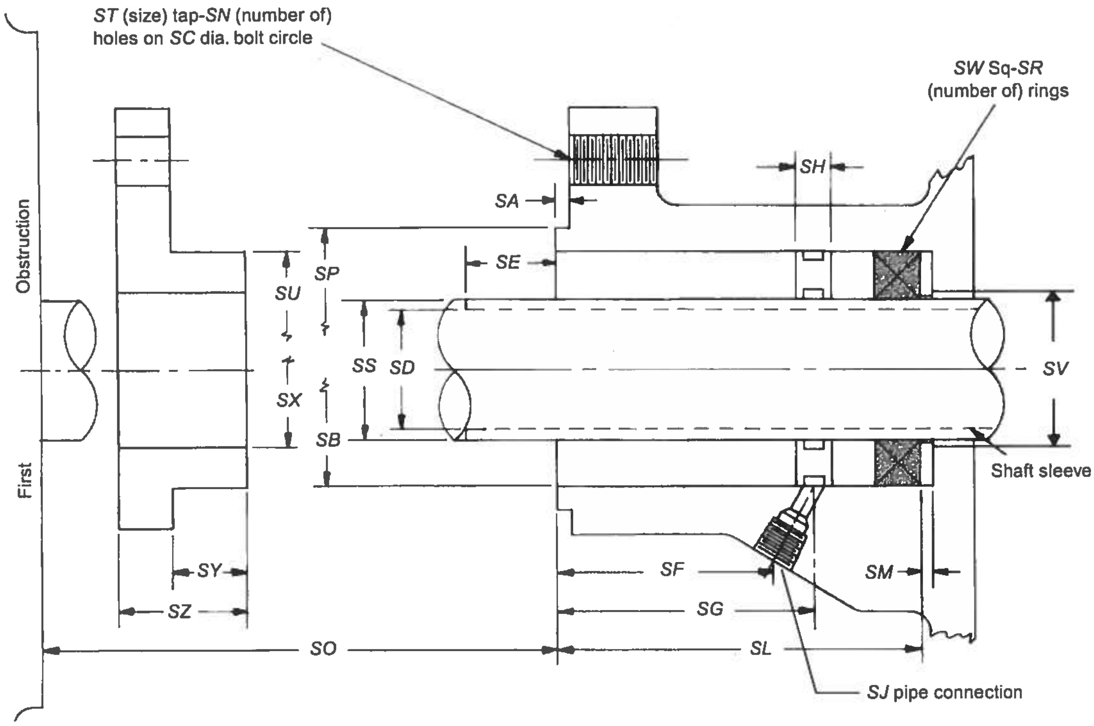


Figure 3.2.39k — Stuffing box or seal chamber

SA Depth of pilot register on face of stuffing box.

SB Inside diameter of stuffing box.

SC Diameter of bolt circle for gland studs or bolts.

*SD** Diameter of shaft under shaft sleeve.

*SE** Distance from face of stuffing box to end of shaft sleeve.

SF Distance from face of stuffing box to centerline of tapped hole for lubrication or coolant fitting outside of stuffing box.

SG Distance from face of stuffing box to centerline of tapped hole for lubrication or coolant fitting inside of stuffing box.

SH Thickness of lantern ring (sometimes called a *seal cage*).

SJ Size of tap and type thread of flush port.

SL Effective length of stuffing box.

SM Thickness of stuffing-box bushing.

SN Number of tapped holes for gland studs or bolts.

SO Distance from face of stuffing box to first obstruction.

SP Outside diameter of pilot register on face of stuffing box.

SR Number of packing rings.

SS Outside diameter of shaft sleeve or diameter of shaft if no sleeve is used.

ST Size tap and type thread for gland studs or bolts.

SU Outside diameter of insertable portion of gland.

SV Throat bore.

SW Size of packing.

SX Inside diameter of gland.

SY Length of insertable portion of gland.

SZ Overall length of gland.

* Applicable only when shaft sleeve is used.

3.3 Design and application

This section provides a basic guide for the application of rotary pumps to fluid handling systems. It includes information to ensure adequate communication of essential facts between the pump purchaser and the pump supplier that will result in a pump-system combination that performs in the manner required.

Because of the wide variety of types and designs of rotary pumps, it must be understood that any statement of general characteristics may be subject to exceptions. Nevertheless, it may be stated that most rotary pumps are self-priming, although some types require the pumping elements be wetted with an appropriate fluid in order to self-prime. Most rotary pumps can be operated dry for only short periods of time. Typically rotary pumps can handle fluids consisting of liquids with entrained vapor. Their ability to do so is, however, highly variable as functions of the particular pump, system, or operating conditions.

Rotary pumps are also available in types suitable for pumping liquids of extremely high viscosity. When handling high-viscosity liquids, the rate of flow of any particular rotary pump varies directly with rotating speed and is relatively independent of system pressure within the operating limits. When handling low-viscosity liquids, the rate of flow varies with speed but may be affected by pressure because of the effect of pressure on slip, the internal circulation of pumped liquid. Consequently, rotary pumps will handle higher rates of flow with increased viscosity at constant speed and pressure.

Rotary pumps are usually designed to operate with close running clearances and with wetted internal surfaces. Therefore, they are not always suited to being run dry. As outlined in Section 3.1, however, there are rotary designs suitable for handling fluids with high abrasive solids contents, but not all types are suitable for such applications.

Rotary pumps are positive displacement machines that deliver a specific volume of fluid to the system with every revolution of the shaft regardless of system conditions. Rotary pumps must, therefore, be suitably protected against excess pressure and temperature rise.

These pumps offer the user an opportunity for power savings over other pumping technologies on viscous fluid applications. Without special design features, however, rotary pumps are constant flow devices; therefore use of variable-speed drives may be recommended to provide effective control where multiple operating points are required.

A rotary pump may be considered for any pump requirement that falls within the scope of the above capabilities and limitations. Investigation of rotary pumps as an alternative to other pumping technologies can be beneficial when application criteria call for the following: viscous liquids, low shear, increased efficiency, low flows, self-priming, multiphase flow, moderate pressures, and solids handling capability. Numerous applications may be found in the fields of chemical processing, refining and transfer of petroleum products, food handling, hydraulic power transmission, and in scores of other industries. Refer to specific standards, such as the 3-A Standards for pumps used to handle food products and API Standards for pumps used in the petrochemical refinery and oilfield services.

3.3.1 Temperature (t)

The ambient temperature range and the temperature of the liquid to be pumped in the application must be specified by the purchaser.

Variations in temperature may cause changes in the specific weight, vapor pressure, and viscosity of the liquid, resulting in variations in pump performance.

Applications involving ambient or liquid temperatures above or below the pump's rated (normal) operating range may require alternate materials of construction, modified internal operating clearances, use of heating or cooling devices, and special attention to features such as shaft sealing and pressure relief valves.

In addition, extreme temperatures may affect the selection and application of drivers and drive equipment, and modification of installation, maintenance, inspection, and lubrication recommendations.

3.3.2 Liquid identification and properties

Many commonly pumped liquids are familiar enough that identification of the liquid by name specifies its properties. For other liquids, however, properties at the conditions of the application should be determined and specified by the purchaser to the pump manufacturer. The liquid properties of possible importance in rotary pump applications are discussed in the following paragraphs.

3.3.3 Fluid type

Rotary pump ratings and performance data are usually determined for liquids that are clean and vapor-free. The type of fluid should be specified in descriptive words, such as liquid, liquid with entrained gas or aerated liquid, liquid with suspended solids, fluid slurry, and any fluid property that could affect pump selection. This allows both the pump supplier and purchaser to select a pump suited to the fluid types. When gases or solids, or both, are used with a pumped fluid, additional information about percent by volume of entrained or dissolved gas, percent by weight of entrained or suspended solids, particle shape and size distribution, among other factors, should be determined and specified by the purchaser. In particular, it is very important to note and specify the presence of solid particles in a liquid that could be abrasive to the materials of construction of the pump.

Liquids are further classified by the viscous response to shear forces.

3.3.4 Entrained or dissolved gases in liquids

Most liquids are susceptible to air or gas entrainment. Air entrainment is prevalent in systems where the liquid is circulated repeatedly and is exposed to air or mechanically agitated in air. It is also found in systems where the pump handles a quantity of air, either intentionally or unintentionally.

Many liquids are known to contain dissolved air or gas. The solubility of air or gas in liquids varies with the type of liquid, time and conditions of exposure, and so forth. Lubricating oils at atmospheric pressure and temperature may contain up to 10% dissolved air by volume. Under similar conditions, gasoline may contain as much as 20% air in solution.

Entrained and dissolved air or gas in liquids handled by rotary pumps are important factors affecting pump performance, both mechanically and hydraulically, especially where negative inlet pressure exists. When the inlet pressure is below atmospheric, entrained gas in the fluid expands, or dissolved gas is freed, and takes up a larger part of the pump displacement, thereby reducing its liquid flow rate (see Figures 3.3.4a, b, c, and d). A saturated solution is one that contains as much dissolved gas as it can at a given temperature.

3.3.5 Viscosity

The viscosity of a fluid is a measure of its tendency to resist an internal shearing force. Hence, for the same given internal shear rate, a fluid that requires a high shear stress has a high viscosity and a fluid that requires a low shear stress has a low viscosity.

3.3.5.1 Shear stress

The shear stress of a fluid is the force per unit area required to cause flow.

3.3.5.2 Shear rate

The shear rate is the velocity gradient normal to the direction of flow or the rate at which adjoining layers of fluid move relative to one another. The fluid shear rate within a pump depends on the pump speed, pump size, and pump clearances.

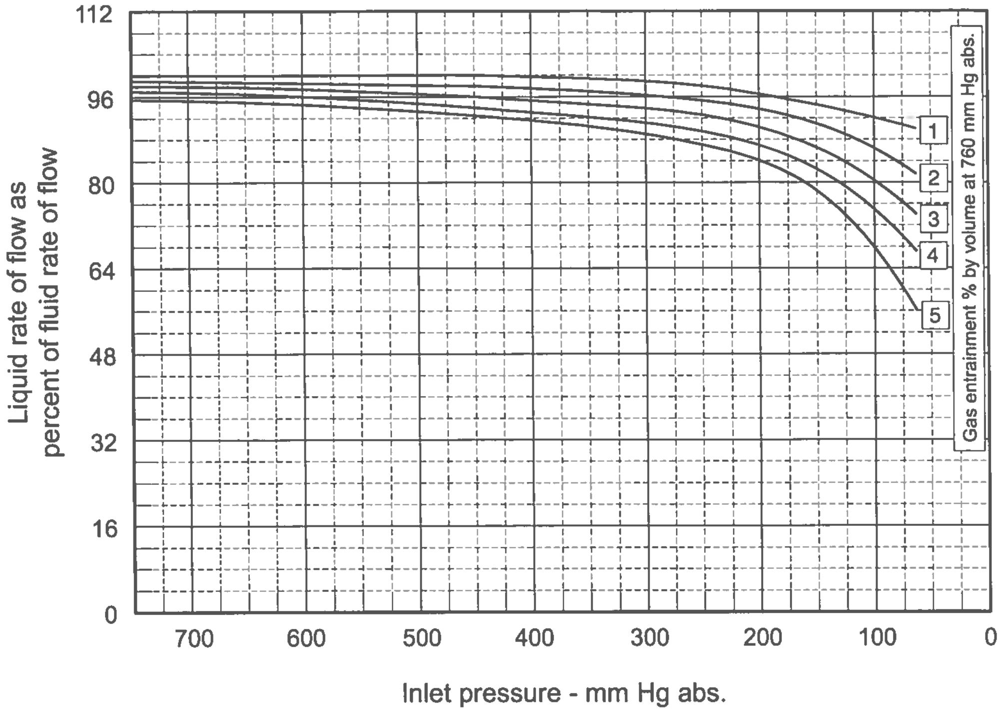


Figure 3.3.4a — Effect of entrained gas only on liquid rate of flow of rotary pumps (metric)

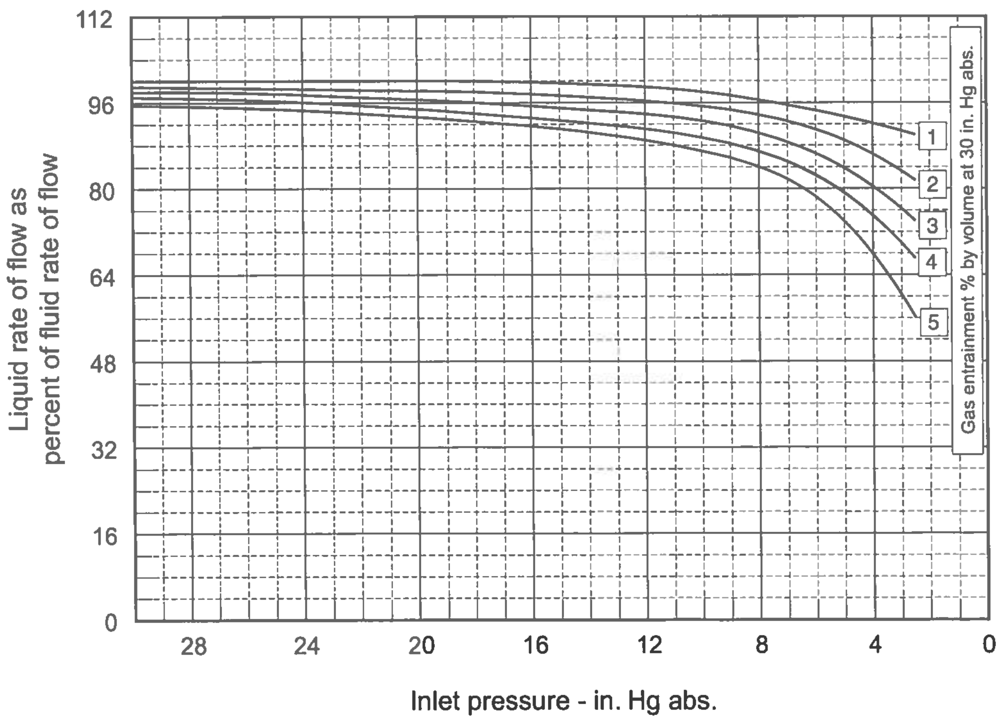


Figure 3.3.4b — Effect of entrained gas only on liquid rate of flow of rotary pumps (US units)

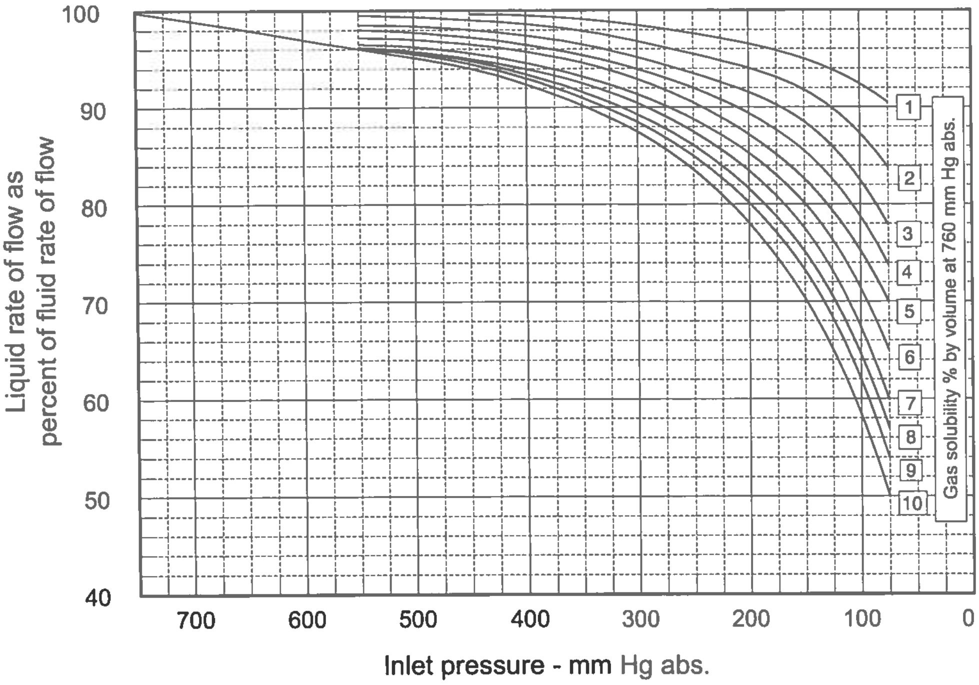


Figure 3.3.4c — Effect of dissolved gas only in saturated solution on liquid rate of flow of rotary pumps (metric)

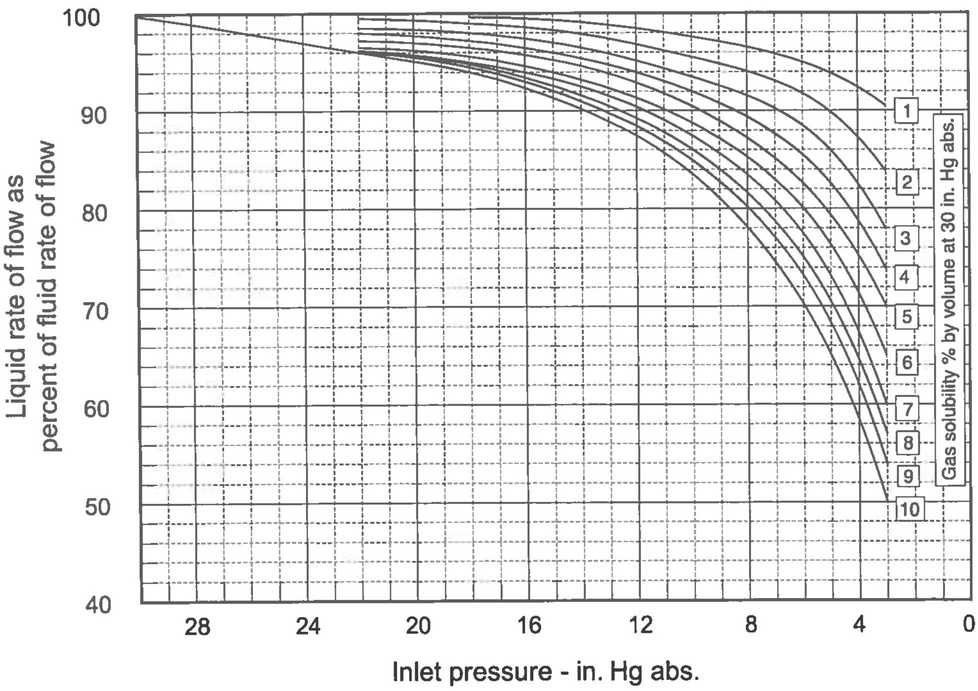


Figure 3.3.4d — Effect of dissolved gas only in saturated solution on liquid rate of flow of rotary pumps (US units)

3.3.5.3 Apparent viscosity

The apparent viscosity of a non-Newtonian fluid is a measure of the resistance to flow at a given shear rate. Therefore, for a specific shear rate, the apparent viscosity can be used in standard pump calculations.

3.3.5.4 Viscosity units

Viscosity units are the poise (dynamic viscosity) and the stoke (kinematic viscosity). These are defined and discussed in the *Hydraulic Institute Engineering Data Book*. They are more commonly expressed as *centipoises* and *centistokes*, each 1/100th of the respective basic unit. One centistoke equals 1 mm^2/s.

A variety of viscometers (viscosimeters) are used in the measurement of the viscosity of liquids. Many of these have generated "units" of viscosity of liquids usually named for the viscometer (*Saybolt Universal, Saybolt Furol, Ostwald, Bingham, Ubbeiohde, Redwood, Redwood Admiralty, Engler, Brookfield*, etc.).

Conversion tables for viscosity units are given in the *Hydraulic Institute Engineering Data Book*, in *ASTM Standards D-446* and *D-2161*, and in other publications.

Typical viscosities for a number of common fluids are shown in the following table. Rotary pumps are capable of pumping many more fluids than those shown in the table.

Viscosity of common fluids

The viscosity values shown in Table 3.4.5.4 are typical viscosities for typical fluids at 68 to 70°F (unless otherwise indicated). Rotary pumps are not limited to pumping the fluids listed.

Table 3.4.5.4 — Viscosity of common fluids

Fluid	Viscosity, SSU	Viscosity, cSt
Alcohol, ethyl	32	1.3
Asphalt @ 250°F	4500	1000
Blood	60	10
Caulking compound	1,000,000+	220,000+
Corn syrup	2000 – 500,000	440 – 110,000
Cream	20 – 100	1 – 20
Crude oil	40 – 5000	4.2 – 1100
Fuel oil #1 #4	 40 80	 4.2 15
Glycerine	5000	1100
Glycol ethylene propylene	 90 240	 17.5 50
Honey	6000	1320
ISO 32 lube oil	350	75
ISO 68 lube oil	750	160

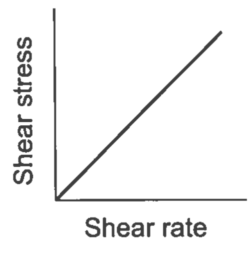
Table 3.4.5.4 — Viscosity of common fluids (*continued*)

Fluid	Viscosity, SSU	Viscosity, cSt
Ketchup	200,000	44,000
Mayonnaise	25,000	5500
Milk	34	2.1
Molasses B	50,000	11,000
Peanut butter	800,000	17,500
SAE 10 oil	250	53
SAE 30 oil	650	143
SAE 50 oil	2000	440
Sour cream	85,000	18,700
Tomato juice	200	41
Varnish	1400	307
Vegetable oil	200	41
Water	31	1

3.3.6 Viscous response types

The viscosity of the fluid to be pumped should be determined and specified for the proposed range of operating conditions. In particular, the operating temperature or range of temperatures and the fluid viscous response type should be specified. Viscous response types are typically categorized as follows:

Newtonian. When the ratio of shear stress to shear rate is a constant for all shear rates, is independent of time, and zero shear rate exists only at zero shear stress, a fluid is termed *Newtonian*. Most mineral oils at temperatures above the cloud point (the temperature at which the oil begins to appear cloudy), solvents, and water approximate this condition and are considered Newtonian fluids. The viscosity of these fluids is independent of rate of shear.



Non-Newtonian. A non-Newtonian fluid will change viscosity with changes in the rate of shear applied to the fluid and/or the length of time at shear.

Several types of non-Newtonian fluids are defined below. When the ratio of shear stress to shear rate increases as shear rate increases, reversibly and independent of time, a fluid is said to be *dilatant*. Highly concentrated pigment-vehicle suspensions, such as paints, printing inks, and some starches, are dilatant fluids. The apparent viscosity of these fluids increases as the rate of shear increases. Some dilatant fluids solidify at very high rates of shear.

Dilatant

When the shear stress to shear rate ratio is constant for shear rates above zero, it is independent of time, but when shear occurs only for shear stress above a fixed minimum greater than zero, a fluid is termed *plastic.* As illustrated, a plastic fluid, such as putty or molding clay, is characterized by a yield point. This means that a definite minimum stress or force must be applied to the fluid before any flow takes place.

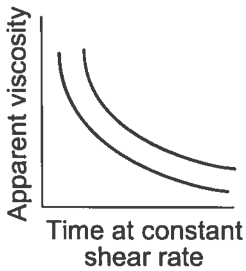
Plastic

When the ratio of shear stress to shear rate decreases as shear rate increases, reversibly and independent of time, and zero shear rate occurs only at zero shear stress, a fluid is *pseudo-plastic.* Many emulsions, such as water-base fluids and resinous materials, are pseudo-plastic fluids. Their apparent viscosity decreases with increasing shear rates but tends to stabilize at very high rates of shear.

Pseudo-plastic

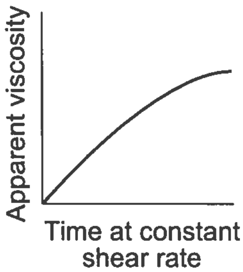
The three types of fluids described above — dilatant, plastic and pseudo-plastic — are also known as *time-independent non-Newtonian* fluids because their flow properties are independent of time. On the other hand, the flow properties of two other types of non-Newtonian fluids are dependent on time. The apparent viscosity of these more complex fluids depends on both the rate of shear and the length of time during which shear has been applied.

A fluid is *thixotropic* when the ratio of shear stress to shear rate decreases and is time-dependent in that this ratio increases back to its "rest" value gradually with lapse of time at zero shear rate and stress, and decreases to a limit value gradually with lapse of time at constant shear rate. Most greases, drilling mud, gels, and quicksand are thixotropic fluids when the apparent viscosity of these materials decreases for an increasing rate of shear and for an increasing length of time at shear.



Thixotropic

When the ratio of shear stress to shear rate is constant for all shear rates at any given instant of time, but increases with time, a fluid is *rheopectic*. Some greases are intentionally manufactured to have partial rheopectic properties that facilitate pumping in a stable condition; but, upon shearing in a bearing, the grease builds up to a higher apparent viscosity.

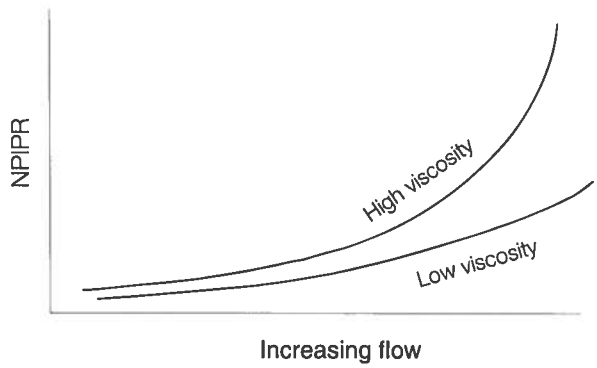


Rheopectic

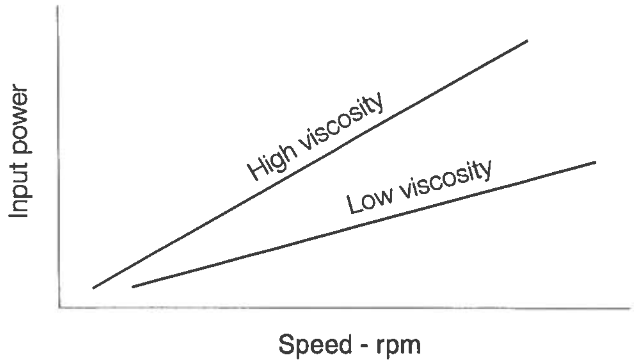
3.3.7 Effect of viscosity on pump and system performance

The viscosity of a pumped fluid typically affects pump ratings as follows:

The net positive inlet pressure required (NPIPR) increases with increasing viscosity, as shown here.

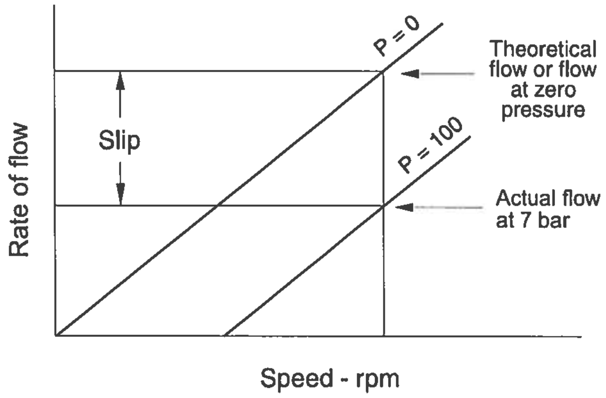


The required pump input power (P_p) increases with increasing viscosity, as shown here.



The maximum allowable pump speed (n) decreases with increasing viscosity:

The pump slip (S), as shown here, decreases with increasing viscosity.



Care must be exercised in applying these generalities to non-Newtonian fluids, as the viscosity may change within the pump due to shear. When the apparent viscosity of a non-Newtonian fluid can be determined, then these generalities can be applied.

Because the exact relationship between viscosity and pump ratings depends on the pump design and on conditions of the application, reference should be made to the pump manufacturer's published data for a particular pump, or the manufacturer should be consulted when considering viscous fluid pump applications.

Energy put into a fluid to overcome resistance to shear causes a finite temperature rise of the fluid.

Manufacturers should be consulted for recommendations on rotary pump applications involving fluids that are shear- or temperature-sensitive.

3.3.8 Specific gravity (s)

Specific gravity is a dimensionless unit defined as the ratio of the density of the fluid to the density of water at a specified temperature. Density is defined as the mass per unit volume. The specific gravity of the fluid, referred to 20°C (68°F) water, is used in many conversions of unit relationships and in formulas or equations used in the computation of pressures from heads, etc. Hence, either the specific gravity or the density of the fluid at the pumping temperature should be stated.

3.3.9 Vapor pressure

The vapor pressure of a liquid is the absolute pressure at which the liquid changes to vapor at a given temperature. It is usually expressed in units of bar absolute or pounds per square inch absolute (psia). For example, the vapor pressure of water at 100°C is 1.01 bar absolute (212°F is 14.7 psia). The vapor pressure of a liquid increases with increasing temperature. Hence, the temperature must always be specified for a stated vapor pressure.

3.3.10 Effect of vapor pressure on pump performance

A liquid moving through a pump vaporizes rapidly whenever the local absolute pressure falls to, or attempts to fall below, the liquid vapor pressure. This kind of rapid vapor bubble formation and subsequent collapse is called *cavitation*. The occurrence of cavitation in a pump may reduce rate of flow. The subsequent collapse of vapor pockets as the fluid is swept into the higher pressure regions of the pump may cause material damage, generate sound and vibration, and produce flow and pressure pulsations in the outlet. The net positive inlet pressure required (NPIPR) rating is based on the prevention of these objectionable phenomena, and the computation of both the NPIPR (required) and the NPIPA (available) depends on knowledge of the liquid vapor pressure in absolute units.

3.3.11 Other fluid properties

The materials of construction of a pump that are wetted by the fluid pumped must resist corrosion by the fluid. Hence, the corrosive nature of the fluid should be specified, or a list of materials known to resist corrosion by the fluid should be specified.

The operation of some rotary pumps is affected by the lubricating qualities of the fluid to be pumped. Hence, the lubricity of uncommon fluids should be described. Lubricity is the ability of the liquid to reduce friction between moving parts.

The purchaser should advise the pump manufacturer if the fluid to be pumped is hazardous to personnel or property. Fluids that are radioactive, toxic, or explosive fall into this category.

3.3.12 Drive specifications

If a specific type of pump drive and/or control is required by the user, then the pump manufacturer should be given this information so that the proper size, type, speed, and rotation of the pump, and size of the base plate can be determined.

3.3.13 Efficiency and energy conservation

The Hydraulic Institute through its Standards and user guides as referenced here (*Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems* – joint Europump and Hydraulic Institute publication) provide the basis for the general recommendation that energy conservation efforts be directed toward pump systems. Additionally in both centrifugal and rotary positive displacement applications, variable-speed pumping offers the potential for lower energy consumption and reduced life cycle costs (LCC). The variable-speed guide details this application and reviews the impact of motor-drive efficiencies (*Variable Speed Pumping: A Guide to Successful Applications* – joint Europump and Hydraulic Institute publication).

Energy conservation is a primary factor in reducing LCC; however, individual users are in the position to decide how important LCC will be in purchasing decisions. Section 3.1 and the Capability Table provide an overview of rotary applications and limitations.

For services that meet product application criteria, users will find opportunities for energy savings in well-designed systems using rotary pumps. This technology has historically found its greatest application at increased viscosities; however, specific designs also handle very thin fluids. Rotary pumps therefore will cover a wide range of services; however, areas of specific attractiveness may be found in applications involving:

- Fluids with viscosities above 10.3 cSt (60 SSU)
- Differential pressures above 50 psi

Rotary pumps have slight performance variations with a change in viscosity, while even a small change can have a large impact on a centrifugal pump. (When considering use of a centrifugal pump for viscous liquids it is recommended the pump performance be rated using ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*.

Energy conservation reviews require examination of the full range of pump system application parameters, therefore, specific study is recommended over generalization. Such services are typically offered by manufacturers of rotary pumps. To assist in that examination, however, comparative efficiency data have been extracted from the *Pump Life Cycle Costs* guide, a joint publication of Europump and the Hydraulic Institute. Figure 2.7a, for rotary pumps, and Appendix C, Figure C2, for rotodynamic pumps, have been consolidated and plotted on Figure 3.3.13. These curves are based on 100 psi (for rotary pumps) and 460 SSU liquid. The data from Figure C2 have been corrected for viscosity using ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*. It is also important to note that this chart is based on centrifugal pumps operating at their best efficiency point (BEP).

3.3.14 Duty cycle

Continuous, cyclic, or intermittent duty requirements can affect the size, speed, bearing design, or life expectancy of the pump and driver, and should therefore be indicated or described.

3.3.15 Other user requirements

Requirements imposed by industrial or governmental codes, such as safety codes, material codes, sanitary codes, and codes for radioactive applications should be specified by reference to the code in question or by detailed description of the portions of the code that apply to the pump.

Requirements imposed by user preferences or rules, such as a particular make of seal or a particular type of port fitting should also be specified.

Requirements imposed by the application, such as jacketed pump, size or weight limitations, metering, low shear action, and unusual mountings should further be specified for like reasons.

The above requirements should be described in enough detail to permit a pump manufacturer to recommend and supply a pump that will meet the application needs. It is useful to include as part of the user specifications the name, address, and telephone number of one or more individuals who can supply additional application information.

3.3.16 Slurry applications

Rotary pumps may be used for in-plant process and pipeline transfer of slurries when metered flow or medium-to-high discharge pressures are required. Because volumetric efficiency and therefore mechanical efficiency are normally dependent on the clearances between the pumping elements of a rotary pump, care must be taken in the selection and application of the pump in slurry service. Slurries containing hard particles can cause abrasive wear in rotary pumps.

3.3.16.1 Slurry characteristics

Slurries vary widely in composition and character, thus it is important to define as closely as possible all the characteristics and components of the mixture. Slurries may exhibit non-Newtonian viscosity characteristics.

3.3.16.1.1 Carrier liquid for slurry

The liquid part of a slurry, which is primarily a vehicle to move the solids, may also be reacting, washing, catalyzing, or performing another function as well. Though the carrier liquid viscosity, temperature, and corrosivity may be known, the effect of solids addition must be evaluated.

3.3.16.1.2 Concentration of solids (by weight) in slurries

The concentration of solids usually has the greatest effect on fluid characteristics of the mixture. A trace to 3% by weight of fine sand content in crude oil, for example, would not be considered a slurry. A range of 5 to 15% by weight slurry of paper stock goes through three different stages of fluidity or shear characteristics. High concentrations of finer solids can change watery carrier liquid to viscous, even thixotropic, type fluid.

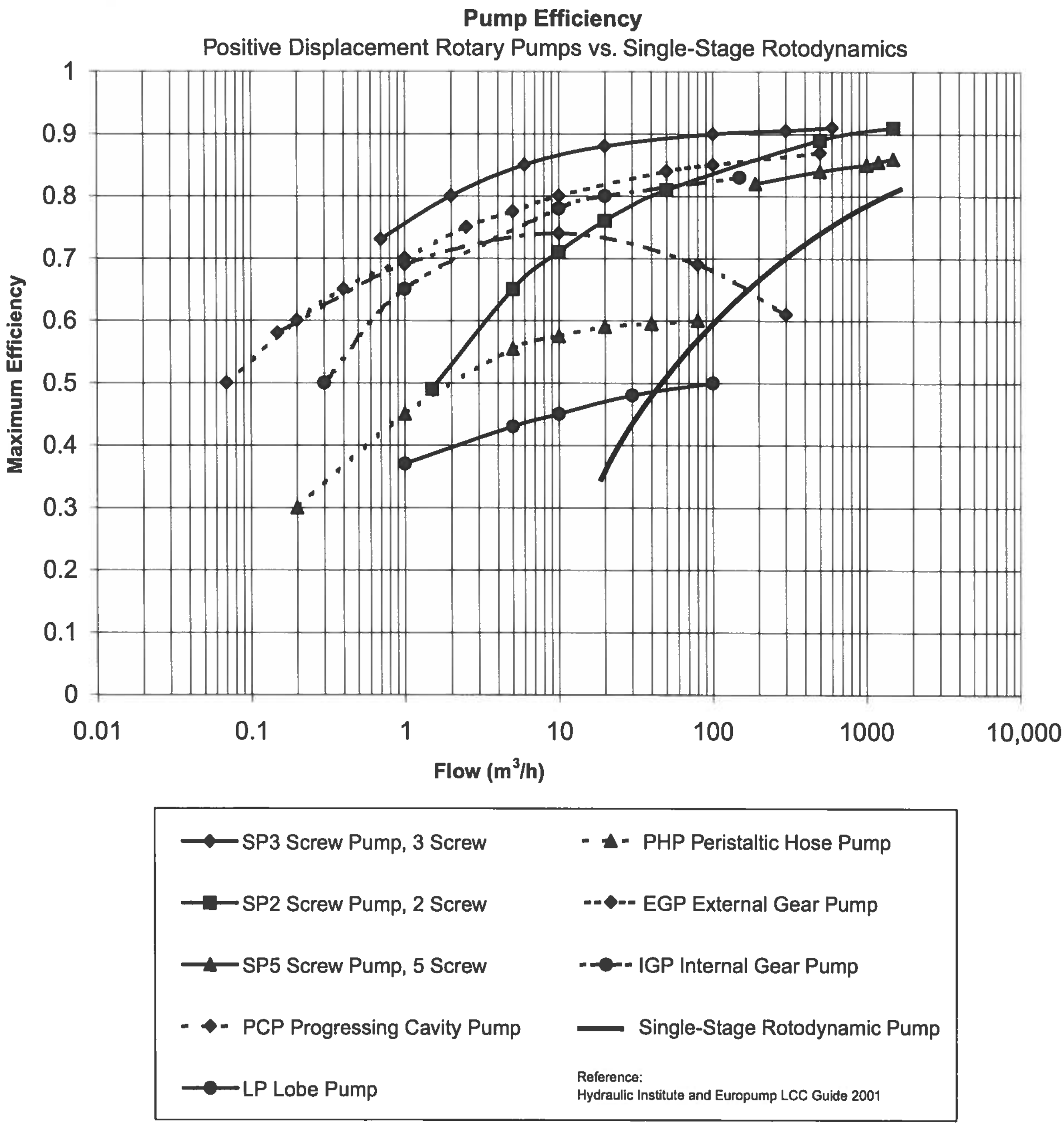


Figure 3.3.13 — Specified conditions: constant speed, constant pressure

3.3.16.1.3 Size of solids in slurries

The size and general shape of the solids in a slurry normally govern whether or not a particular rotary pump configuration and/or cavity or chamber size can be used for the slurry in question. Usually the cavity must be greater than particle size. In some pumps, the rotor configuration will accept large particles as long as the size, distribution, and shape are controlled. Thus, the size must be related to porting, rotor cavity size, configuration and interaction, and operating clearance.

3.3.16.1.4 Hardness of solids in slurries

The hardness of solids greatly influences the wear rate of the rotary pump because there is close relative motion of surfaces at rotor tip or flank velocity. The range of hardness is from soft solids, such as polymer chains of lubricating quality; to fibrous products that, though usually soft, may carry hard particles; to very hard irregular particles, such as sand and carbon. Carbon is mainly soft; however, the aggregate carries lumps approaching silicon carbide in hardness. A qualitative scale listing several material hardnesses is often useful to compare solids and select pump materials (see Figure 3.3.16.1.4).

3.3.16.1.5 Settling characteristics in slurries

The critical factor governing a system handling a watery slurry, in which the solids have a much higher specific gravity than the carrier liquid, is the settling rate and characteristics. The settling rate is a measure of how quickly solids will fall out of suspension in a slurry. Coarse solids with high settling rates are carried in a rotary pump with many precautions to prevent bridging, draining, and squeeze out.

3.3.16.1.6 Apparent viscosity of slurry versus shear rate

The apparent viscosity of a slurry for a pump and system can be obtained through use of viscometer data on a log-log curve of viscosity versus shear rate. Average shear rates for rotary pumps will vary widely with the design, but even at the reduced speeds required for slurry application, they will greatly exceed the values for the flow in the piping.

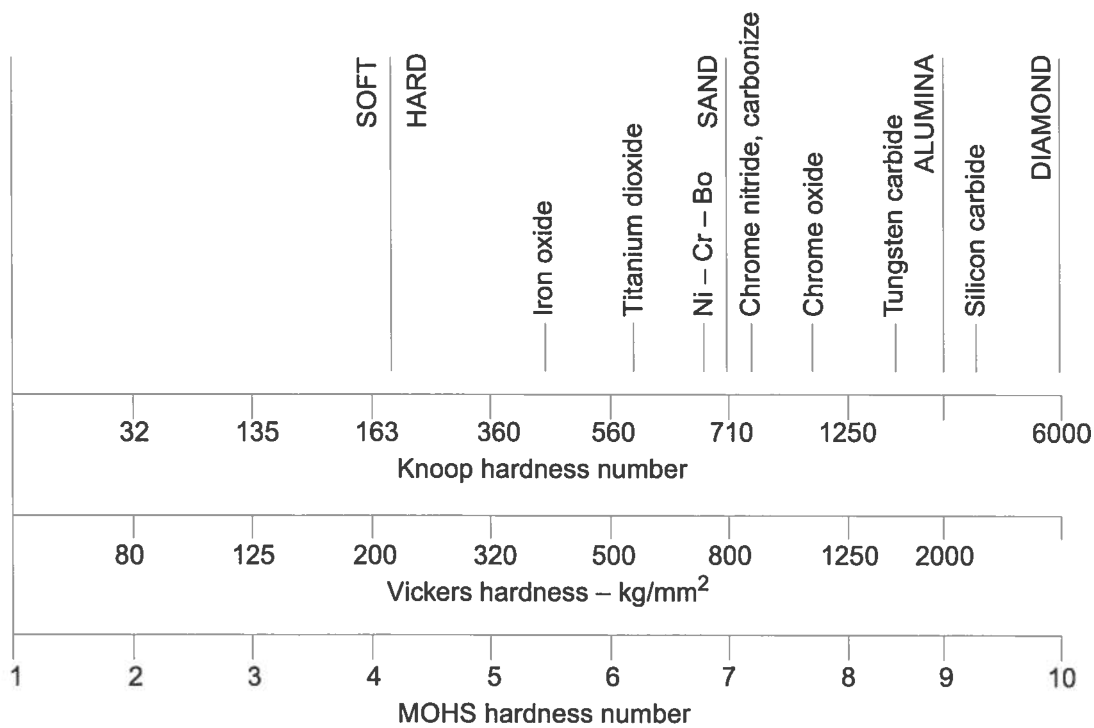


Figure 3.3.16.1.4 — Materials hardness

The power law, originally proposed as the Ostwald–de Waele power law, states that the shear rate curve should be a straight line on log-log plots. Note that most slurries follow the power law over a small range of shear rate and extreme extrapolation should be avoided (see Figure 3.3.16.1.6).

3.3.16.2 Performance changes with slurries

The effect on rotary pump performance can vary widely as slurries change with time, control, character, and so forth. Low concentrations of fine nonsettling solids in a Newtonian fluid carrier may have no appreciable effect on either the power requirements or the pump rate of flow. In general, as the percentage and size of the solids increase at given conditions of operation, speed, and pressure, the pump input power curve increases (see Figure 3.3.16.2).

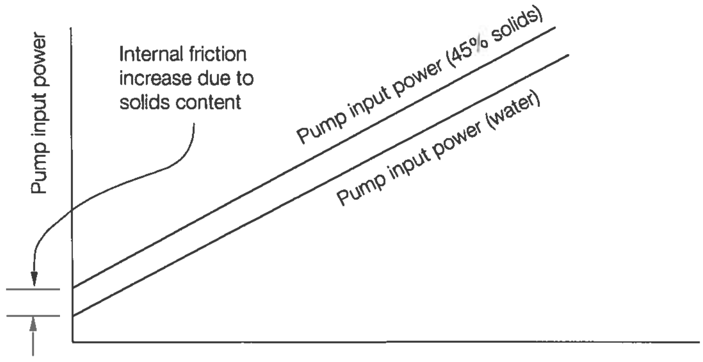


Figure 3.3.16.2 — Differential pressure versus pump input power

3.3.16.2.1 Speed effects with slurries

Regardless of the type of rotary pump used for abrasive slurries, pump speeds should always be reduced well below those for nonabrasive fluid applications. Speed directly affects product shear and the relative velocity of solids to the pump housing and rotor. The apparent viscosity of shear-sensitive mixtures can vary considerably from an apparent viscosity extrapolation as shear rates go beyond 1000 and 10,000 sec^{-1}.

3.3.16.2.2 Flow velocity of slurries

Although pump speeds should be minimized to reduce abrasive wear and maintenance, care should be taken to keep all velocities within the pump and system above the critical carrying velocity of the slurry. The magnitude of this velocity is a function of size, shape, density, and concentration of the slurry particles, as well as the density and viscosity of the carrying fluid and various other factors.

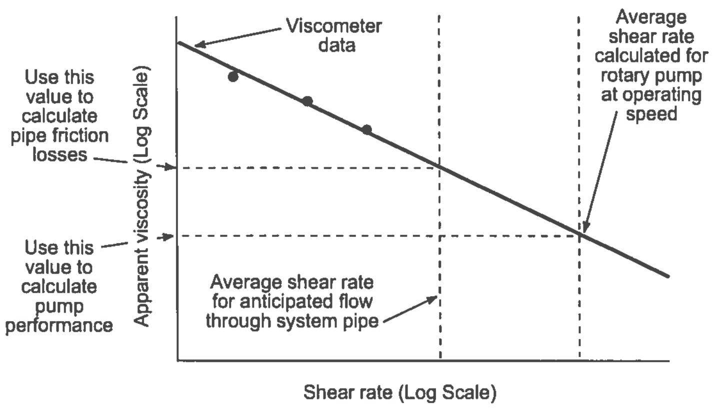


Figure 3.3.16.1.6 — Typical slurry system conversion curve

3.3.16.2.3 Slurry effect on friction power

Many factors other than the size and percentage of solids in the slurry affect the magnitude of the increase in friction power. These factors include the particle, the carrying fluid viscosity, the particle size distribution, the pump design, and many others. Although its potential shortcomings should be recognized, one convenient and reasonably accurate method of estimating these effects is to assume the pump performance to be equal to the known pump performance for a true fluid of Newtonian viscosity equal to the apparent viscosity of the slurry at the operating conditions and at the shear rate at the rotor.

3.3.16.2.4 Slurry effect on slip

Other performance characteristics of the rotary pump, such as rate of flow, may be affected to varying degrees by a slurry. Because slurries are two or more components mixed together, the rotary pump slip tends to approach that of the least viscous product at lower concentrations or with freer type solids.

3.3.16.2.5 Clearance provision for particle size

A clearance provision of greater than the maximum particle size for handling hard or fibrous particles is necessary in many rotary pump configurations to prevent binding, wedging, and severe wear. This clearance, if greater than normal design clearance, could permit greater slip within the pump and thus should be evaluated for the range of slurry handled.

3.3.16.2.6 Operating sequences with slurries: stop, flush

The pump and system should be flushed of all settleable solids before the pump is shut down and the velocity goes to zero. Nonsettling slurries, such as putties, thick pastes, and filter cakes, may be pumped at lower velocities to minimize friction losses in the system.

3.3.16.2.7 Pressure relief provision for slurries

Because rotary pumps are positive displacement and slurries have an inherent tendency to settle and clog piping, overpressure protection should be a part of the system. Slurry service, with its tendency to abrade or foul, precludes the use of many conventional safety relief valves, but rupture discs, pressure-controlled diaphragm and pinch valves, or safety relief valves designed specifically for slurry service should be used.

3.3.16.2.8 Testing and modeling for slurries

Testing and modeling of pumps on representative slurries should incorporate design velocities, shear rates, and speeds to get a true picture of apparent viscosity for both friction horsepower and slip. Reference testing at known viscosities in the same setup then confirms the working data for the application.

3.3.16.3 Wear consideration for slurries

3.3.16.3.1 Type of wear

The wear mode in pumps handling hard or fibrous solids is abrasive or erosive unless the materials of the pump wear surfaces are harder than the solids and/or resilient enough to not be scored by the solids.

3.3.16.3.2 Corrosion effect on wear

Special attention should be given to the corrosive aspects of the slurry when selecting rotary pumps. It should be noted that if the metal selected for a pumping element depends on an oxide film for corrosion protection from the slurry, then the abrading effect of the slurry will remove this protection, thus causing deterioration from corrosion at a much more rapid rate than would be expected in tanks and piping.

3.3.16.3.3 Speed effect on wear

Speed is one of several contributors to wear rate. With abrasive solids, wear rate is usually proportional to speed and affects the high relative motion/lower hardness material, except where a soft embeddable material is adjacent to a hard material.

3.3.16.3.4 Materials of construction for slurries

Some pump design techniques to minimize wear include the following:

Use pumping elements that are harder than the hardest slurry particles. The slurry particles must be fine enough to prevent jamming of the pump, rapid settling, or injurious high friction on pumps that employ hard pumping elements.

Use pumping elements that combine soft and hard materials in such a fashion as to reduce abrasion and provide embeddability.

3.3.16.3.5 Pump design for slurries

Although rotary pumps are capable of limited slurry handling, the particular pumping elements and chamber configuration make the various rotary designs more or less adaptable to specific types of slurry. Many slurries similar to paper stock require open porting and clearances and definite minimum velocities of flow. Clay slurries require low shear rate.

3.3.16.3.6 Sealing against slurries

Sealing rotary pumps handling slurries usually requires flushing and/or other means to exclude the solids from the area of the sealing surfaces. Another measure to reduce sealing surface wear is to apply material(s) harder than the solids being pumped. In some instances, special abrasive seals are designed to run directly into the slurry without an external flush.

3.3.17 Rotary pump noise levels

Noise generated by pumping equipment can be categorized into the following three basic types:

- Fluidborne noise – associated with pressure variations within the fluid (also called *pressure pulsation*);
- Structureborne noise – associated with vibration in a structure;
- Airborne noise – audible sound caused by vibrating surfaces exciting the air surrounding them or by windage from rotating surfaces.

Analysis of pumping system noise can be very complex. During operation, pump systems generate all three types of noise. They are all interrelated, being caused by the energy input to the system from the pump and its drive components. For example, pressure pulsation can cause structureborne noise by exciting mechanical vibrations and vibration can become audible noise. The amplitude of noise from a pumping system can also be greatly affected by the associated piping and structural supports, which may be in resonance with the pump-initiated vibrations and pulsations.

3.3.17.1 Fluidborne noise

Fluidborne noise is caused by pressure pulsations created when the individual pump cavities are discharged into the outlet chamber of the pump, by turbulence within the pump or system at high velocities or flow rates, by cavitation, or by excessive amounts of dissolved or entrained gas in the pumped liquid. The magnitude of the pressure pulsation is a function of system operating conditions. The magnitude of the pressure pulsation at a particular operating condition will vary with the type of pump applied and its mechanical condition.

The pressure pulsations generated by the pumping action may couple directly to the structure, such as the piping, and radiate throughout the system.

Turbulent flow produces broadband energy and can also contribute to the excitation of the system structure. It may occur within the pump itself or in valves and piping. System design must take fluidborne noise into consideration.

Cavitation produces violent, high-intensity energy whenever localized system pressure is insufficient to completely fill the pump suction cavity. When this occurs, vapor pockets or bubbles are formed in the fluid to complete the filling of the pumping voids. These vapor pockets then collapse or implode when they encounter fluid pressure above vapor pressure as the liquid passes through the pump. The collapse produces local shock waves of sufficient energy to excite vibration in the structure, causing airborne noise and sometimes physically damaging components within the pump. Cavitation noise usually has a predominant structureborne and fluidborne component at vane or tooth frequency and its multiples.

Dissolved and entrained gas produces noise in much the same manner as cavitation, except gas comes in and out of solution and expands instead of changing between the liquid and vapor states, and the energy levels are less, avoiding damage to pump components.

3.3.17.2 Structureborne noise

Structureborne noise may result from mechanical contact within the pump, pressure pulsations in the pump or piping, or mechanically from the pump, driver, or coupling due to misalignment or wear. Rotary pumps normally operate at slow enough speeds with small diameter rotors, so that mechanical imbalance is usually of little significance.

Within a rotary pump, physical contact between the rotating elements, such as gear teeth (especially when the contact surfaces are irregular as may be caused by wear), result in vibration. Certain types of rotary pumps, such as spur gear pumps, also have a tendency to trap and attempt to compress part of the liquid being pumped during the pumping process. This leads to resultant shock loads on the rotor(s) and bearings of the pump, causing vibration and structureborne noise.

Mechanical design of the bedplate grouting and piping supports, and proper alignment of pump and driver also impact structureborne noise level in a system.

Vibration measurements made with appropriate sensors are often used to monitor the condition of pump units.

3.3.17.3 Airborne noise

Airborne noise is usually the result of the interaction of fluidborne noise on the structure and structureborne noise on the surrounding air space and is evident by the presence of sound in the space. The apparent magnitude of sound from a particular source is affected by the physical characteristics of its environment.

Airborne noise measurements are typically made in terms of the sound pressure level using a sound level meter. The unit of measurement is the decibel (dB). This type of measurement is a broadband or overall noise measurement. For demonstrating compliance with most requirements, the meter must have a frequency weighting network, called the *"A" scale*, which electrically approximates the frequency response of the human ear. The measurements thus made are "A-weighted" sound levels in units of dbA, with "A" denoting only that the measurement was made using "A-weighting".

Presently, OSHA (US Occupational Safety and Health Administration) requirements limit the sound level to 90 dBA for eight hours of exposure. The allowable dBA value increases 5 dBA for each halving of the exposure time.

To define the frequency characteristics of the overall airborne noise, octave band sound pressure levels are measured with an instrument called an *octave-band analyzer.* An octave is a range of frequencies in which the ratio of the highest to lowest frequencies is 2. Octave bands are measured as sound pressure levels in terms of decibels. No frequency weighting is used for these measurements. The preferred series of octave bands cover the audible range in 10 bands and extend from the center frequencies of 31.5 to 16,000 hertz.

Additional information on broadband and octave-band airborne measurements can be found in the Hydraulic Institute Standards for the *Measurement of Airborne Sound from Pumping Equipment* found in HI 9.1– 9.5 *Pumps – General Guidelines.*

For performing a diagnostic analysis, an analyzer with a much narrower bandwidth is required to accurately identify each individual frequency component.

3.3.18 Rotary multiphase pumps in oil and gas application

Rotary pumps historically have been applied to pumping viscous fluids that are not readily handled by centrifugal and other types of rotodynamic pumps. A further advantage of certain rotary designs is their ability to add full pressure rise to various kinds of liquid-and-gas mixtures with widely varying density. This advantage was found to be useful for low viscosity fluids, such as oil and gas applications, and has resulted in a much wider application of progressing cavity and two-screw rotary pumps.

Rotodynamic pumps deployed in such applications have to substantially vary the rotation speed to accommodate typical ranges of gas void fraction (GVF). Additionally in oil and gas applications the flow of such multiphase fluids has often proved to be unsteady to the point that slugs of gas or liquid can develop that last for significant periods of time. Associated variations in pressure rise also occur. All of these are readily accommodated by rotary multiphase pumps – even without a change in speed. Speed variability for such rotary machines is intended to accommodate the longer-term changes in the production rates of the field.

Integrated multiphase pump (MPP) packages using these versatile pumps are often used as booster pumps downstream of downhole well pumps. Two situations are typical of the variations in conditions encountered in the field: 1) in normal, steady operation, as the production decreases, the pumping action can depress the inlet pressure, increasing the GVF at the MPP; and 2) during startup, the sequence of starting the downhole pumps and the MPP can result in very high inlet pressure that must be accommodated by the mechanical seals in the pump.

Progressing cavity, timed screw pumps, and timed rotor lobe pumps present specific advantages in their application on this service. A distinguishing constructional feature of the timed screw MPP is that it is axially balanced, having double-section rotors. The internal leakage or slip through the resulting clearances profoundly affects the performance of this machine. When handling incompressible fluids, the slip is the same through each location of trapped volume (lock) defined by the meshing screws and the pressure rise is linear across the screw length. In MPPs this slip is not the same in every lock and the pressure rise across the screws is nonlinear. The pressure loading produces deflection of the rotors, which can be successfully limited by the proper choices of the rotor geometry.

Timed rotor lobe pumps are a compact alternative for single wells with GVF up to 93%.

As long as the GVF does not exceed approximately 80%, a standard screw pump can be used for multiphase applications. In many multiphase applications, the average GVF may be acceptable but it can instantaneously vary from 0 to 100%, resulting in slugs of liquid or gas. Multiphase designs are available that separate and recirculate small amounts of liquid or supply liquid from another source when pumping multiphase product with GVF greater than 95% or when slugs of 100% gas are expected.

Progressing cavity pumps can operate at gas fractions of 99% without recirculating fluid, and can handle sand cuts of 3 or 4% without any filtering or preseparation. The excellent suction capability of progressing cavity pumps enables them to be used in vapor recovery systems.

3.3.19 Data sheet

The information required for proper selection or design of rotary pumps is summarized in the data sheet shown in Figure 3.3.19.

3.4 Installation, operation, and maintenance

Rotary positive displacement pumps are manufactured in a wide variety of designs or types per Figure 3.1, Types of rotary pumps. Proper installation, operation, and maintenance of rotary pumps vary widely over the complete range of services to which the pumps may be applied. Satisfactory results can only be fully achieved by following the manufacturer's instructions for the size and type of pump or unit involved.

General instructions that follow normally apply to all pumps and units, but they are only intended to supplement the manufacturer's manual or recommendations. Should questions arise that are not answered by material contained herein or in the manufacturer's manual, then the manufacturer or his representative should be contacted directly.

The term *units* is construed to mean pumps complete with mounting base and with or without driver. In actual practice, the terms *suction, discharge*, and *liquid* are often substituted for the terms *inlet, outlet*, and *fluid*.

3.4.1 Shipment inspection

Pumps and units are shipped suitably protected to prevent damage in transit from normal handling. When received, the shipment should be inspected immediately. Damages to the packaging or crating that may reveal content damages when unpacked should be noted on the carrier's bill of lading. Shipment shortages, checked against the bill of lading, should be reported to the carrier and likewise noted on the bill of lading.

The manufacturer or its representative should be advised of damage to contents not a fault of the carrier, or, in the event of shortages, checked against the packing list.

3.4.2 Storage

After receipt and inspection, a pump not installed immediately should be repackaged and placed in suitable storage. Protective coatings on unpainted surfaces should be inspected and left intact. Unpainted surfaces, not factory-treated with a rust-inhibiting coating, should have a protective coating applied. Plastic or gasket-type port covers should be left in place. Pumps received wrapped with corrosion-inhibiting treated material should be rewrapped. Select a clean, dry storage location. When storage must be in areas of moist, dusty atmosphere, further protect the pump or unit with a moisture-repellent cover until it is to be installed.

3.4.2.1 Handling equipment and tools for installation

Overhead handling equipment with proper slings or chains for setting the pump and equipment may be required. Lifting equipment should be carefully selected with consideration for load carrying ability and compatibility with the pump manufacturer's installation recommendations. Safety slings, wire rope, or chain should be placed only at specific lift points and should not contact other points of the unit.

3.4.2.2 Manufacturer's instructions

The manufacturer's service manual should be read thoroughly before installing or operating the equipment. This manual should be retained for reference.

3.4.3 Installation

All pumps and units must be installed in compliance with regulatory body codes (national, state, and local) in effect at the time of installation. In cases where a code conflicts with the following instructions, the code shall prevail.

General

1) Name of user ______________________________

2) Pump site location ______________________________

3) Elevation ____________ 4) Ambient temperature range _____ to _____

5) Type of service: ❑ once through ❑ recirculating

6) Duty cycle: ____________ hours/day ______ days/week ______

7) Area electrical classification ______________________________

Fluid to be handled

NOTE: Provide data called for in items 8 – 17, inclusive, for each fluid to be pumped.

8) Type or specification: ❑ Newtonian ❑ Non-Newtonian

NOTE: See Section 3.3.6, Viscous response types. If non-Newtonian, submit all available apparent viscosity vs. shear rate data.

9) Viscosity ________________ at rated (normal) temperature

________________ at minimum temperature

________________ at maximum temperature

10) Hazardous characteristics: ❑ corrosive nature ___ pH ❑ Toxic ❑ Flammable

Other ______________________________

11) Pumping temperatures: Rated (normal) ______ Minimum ______ Maximum ______

12) Vapor pressure: at rated temperature ____________

at minimum temperature ____________

at maximum temperature ____________

13) Specific gravity ______ at ______ (temperature)

14) Pour point ______ Flash point ______

15) Solids present:

Type ______________________________

Amount ______________________ % by weight

Abrasive character ______ Particle shape ________

Size distr. ________ Density ______________

16) Gas(es) present in fluid at atmospheric conditions:

Entrained _____ % by volume at 760 mm Hg (30 in. Hg) abs.

Dissolved _____ % by volume at 760 mm Hg (30 in. Hg) abs.

(Continued)

Figure 3.3.19 — Suggested rotary pump application data sheet

Solubility ______ % by volume at 760 mm Hg (30 in. Hg) abs.

17) Flushing liquid available ____________________

System conditions

18) Rate of flow desired: Min. ________ Max. ________

19) Required pump outlet pressure: Min.______ Max.______ Continuous ______

20) Available pump inlet pressure: Min. ______ Max. ______

NOTE: For determination of values in Items 19 and 20, refer to Section 3.2.27, Pressure (p), and 3.2.31, Net positive inlet pressure available (NPIPA).

Driver

NOTE: Direction of driver rotation is determined as viewed facing the outboard end of the driver and shall be the same as the direction of the pump rotation when viewed facing the shaft end of the pump.

21) To be supplied by pump manufacturer?: ❑ Yes ❑ No

22) Electric motor: mfr. ____________ frame __________ enclosure __________

motor type __________ power __________ rpm __________

phases ________ volts ________ amperes __________

cycles ________ direction of rotation ❑ CW ❑ CCW

23) Steam turbine: mfr.____________ type ________ rated power ________

rated speed ________ rpm direction of rotation ❑ CW ❑ CCW

rated initial steam pressure ____________

rated initial steam temperature ____________

rated exhaust steam pressure ____________

24) Engine: mfr. ____________ type ________ power (continuous) ________

speeds (continuous) ________ rpm direction of rotation ❑ CW ❑ CCW

25) Power take-off: mfr.____________ model____________

speed ________ rpm direction of rotation ❑ CW ❑ CCW

26) Other types of driver ____________________

____

____

(Continued)

27) Auxiliary equipment, such as coupling type, clutch, brake, speed control, starter, drive guards, etc.

____

____

____

Other user specifications

Performance of any required pressure control devices. (Refer to Section 3.4.3.12.3, Pressure relief valves.)

Integral relief valve ______________________________

Attached relief valve ______________________________

Pressure-regulating valve, pressure switch, rupture diaphragm, etc. ______________

____

Pump materials: seal types, mounting arrangement, port size, type and arrangement, space limitations, cleaning or flushing cycle, heating or cooling requirements, coupling guard, compliance with applicable standards or regulations, etc.

____

____

____

____

____

____

____

Name, address, telephone, FAX, and e-mail of person to contact for additional application data:

____

____

____

____

____

____

____

____

____

3.4.3.1 Cleaning

To ensure that test fluids and/or preservatives will not contaminate the fluid to be pumped, the pump shall be thoroughly flushed before it is placed in service. The flushing media must be compatible with the pump and seal materials. Rust-inhibiting coatings on unpainted exterior surfaces should be removed with a safe solvent (gasoline or toxic solvents are not recommended). Ports and other openings in the pump should be kept covered until ready for piping.

3.4.3.2 Location

Rotary pumps are self-priming, but to prevent inlet condition problems, place the pump near the liquid source and preferably below it. For some pumps handling solids in suspension, the manufacturer requires a vertical inlet port location directly under the source of supply. Short, straight inlet lines will reduce cavitation noise and subsequent pump damage.

A dry, clean, well-lighted and ventilated site should be selected for installing the pump unit. The site should also provide accessibility for visual observation and routine inspection. The driver shall be compatible with the environment (e.g., an electric motor may be open, dripproof, totally enclosed, or explosionproof).

Aboveground locations are recommended. Where a pit installation is necessary, provision must be made to prevent flooding. When pump units are located on any floor not supported by the ground, the floor and beam loads must be checked to determine their ability to safely carry the mass of the unit and its foundation.

For larger pump units, ample headroom shall be provided for lifting devices during installation and servicing. Sufficient open area around the unit shall be provided for maintenance or service work. Axial space to permit removal of the main pumping elements is extremely important.

3.4.3.2.1 Access for maintenance

Pumps should have adequate access and working room for maintenance operations. Adequate axial and overhead space for lifting devices and working clearances must be provided.

3.4.3.3 Foundation

The foundation is a structure designed to absorb any vibration, strain, or shock while at the same time providing a permanent, rigid support for securing the pump unit.

Many small pumps or units that are part of a larger machine package may be suitably mounted directly on the metal base of the package. Lightweight pumps or units may be fastened directly to an existing concrete floor, with the floor meeting the criteria of a foundation. In both cases, these small units must be treated the same (shimming, alignment, etc.) as larger pumps or units mounted on a separate foundation.

A concrete mix of 1:2:4 ratio by volume (cement, fine aggregate, coarse aggregate) provides an excellent foundation for the pump unit. Concrete is low in cost, yet it has suitable mass to amply support the unit. For epoxy-base mixes, follow the manufacturer's recommendations for mix and cure time.

The foundation should be 100 to 150 mm (4 to 6 in) longer and wider than the mounting base of the unit. Height of the foundation should run about twenty times the diameter of the foundation bolts. After the foundation form has been constructed, a template should be made to position and hold the foundation bolts in place while pouring the concrete. The manufacturer's certified print lists size and location of the bolt holes in the base. Figure 3.4.3.3 illustrates two types of foundation bolts commonly used.

Each foundation bolt is installed in a pipe sleeve with an inside diameter three times the bolt diameter. The pipe sleeve allows for slight lateral movement of the bolts after the foundation is poured. Stuff rag or paper waste between the bolt and inside of the sleeve to prevent concrete from entering the sleeve while the foundation is poured. The pipe sleeve length should be approximately ten times the foundation bolt diameter. Length of the foundation bolt should allow for 20 to 40 mm (0.75 to 1.5 in) of grout, the height of the base, and 6 to 12 mm (0.25 to 0.5 in) extension of bolt above the nut after the pump is leveled and fastened down. The concrete should be left rough at the top of the foundation for anchoring the grout. Cure time of the concrete, prior to operation of the unit, should be a minimum of 14 days. A cure time of 28 days is preferred.

Large units with pump and driver mounted on separate bases may be mounted on separate foundations; however, a common foundation is recommended. The same foundation, leveling, and grouting recommendations apply as when pump and driver are mounted on a common base.

3.4.3.4 Mounting the driver

Many pump units are shipped complete with driver. For units shipped without the driver, the driver should be mounted to the base before the unit is placed on the foundation. Final alignment of pump and driver should take place after the unit is secured to the foundation and grouted.

Units shipped without the driver usually do not have drilled and/or tapped holes in the base to position and secure the drive. Depending on the unit size, mounting pads on the base to accept the driver may or may not be included. On small units, the driver is often secured directly to the formed steel base.

For direct-driven pumps, locate and secure the coupling halves on the driver and pump shafts. Chalk the base or pads at the approximate location where the cap screws or bolts will secure the driver to the base. Place the driver on the base and set the proper distance between the coupling halves. Locate the driver so that pump and driver shafts are in axial alignment, particularly as viewed from the top of the unit. (See Section 3.4.3.8, Alignment.) Scribe the location of the driver-mounting holes on the base. Remove the driver, then drill or drill and tap the holes to accept the proper size bolts or screws. Replace the driver on the base, insert the screws or bolts, and align the driver before tightening.

For pumps driven through a separate speed reducer (or increaser), first mount the reducer relative to the pump, then mount the driver relative to the reducer. Follow the same procedure as for direct-driven pumps.

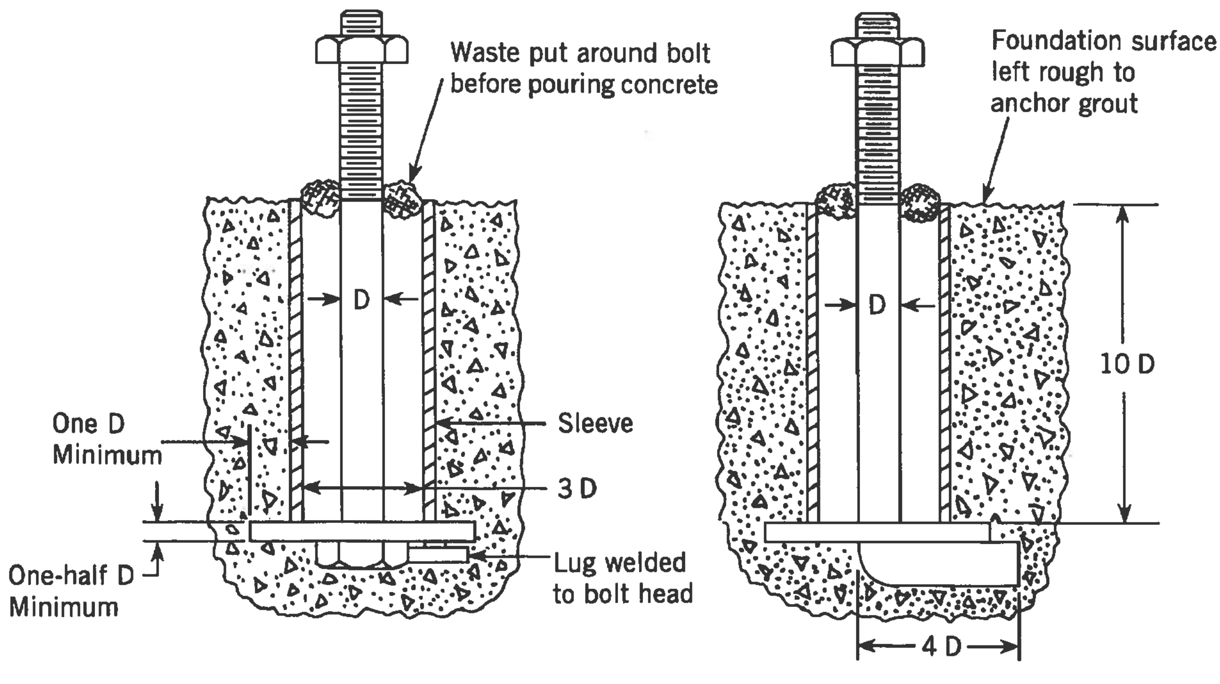


Figure 3.4.3.3 — Typical foundation bolts

For V-belt driven pumps, the belt sheaves must be aligned before scribing the base. Adjustable slide rails mounted under the driver should be installed for proper belt tensioning. Where adjustable slide rails are not used, the driver base should have slotted feet to permit both belt removal and provision for tightening the belts. In this case, the mounting screws or bolts should be located at the center of the slot with the belts drawn up to approximate proper tension.

When not supplied by the manufacturer, coupling, shaft, and belt guards conforming to ANSI B15.1 shall be installed to protect personnel while the unit is in operation.

3.4.3.5 Leveling

Before the unit is set on the foundation, clean the underside of the unit base and the top of the foundation. Remove the waste from the pipe sleeves around the foundation bolts. Set leveling pads, either tapered wedge or flat shim plate, adjacent to the foundation bolts. Pads should be 12 to 20 mm (0.5 to 0.75 in) total height for grouting. Lower the unit over the foundation bolts to rest on the leveling pads.

A small spirit level should be used in leveling the unit. Surfaces for determining the level in order of preference are the pump flange faces, the pump shaft, projections of machined surfaces on the base, and, finally, the surface of the base. Adjust the height of the leveling pads until the base is level and supported at all leveling pad locations. The foundation bolt nuts are then snugged up but not tightened. Double-check the level of the unit.

3.4.3.6 Grouting

The purpose of grouting is to prevent lateral shifting of the base, and to add mass and stiffness to it, not to correct for irregularities in the foundation.

Construct a wood frame around the foundation as shown in Figure 3.4.3.6, with the height of the frame set by the desired thickness of finished grout. A typical grout mix of 1:2 ratio by volume (portland cement and fine sand) is prepared with just enough water to obtain a creamy consistency and allow free flow under the base. Wet the top of the foundation prior to grouting. Pour grout between the frame and the base, and also through grout holes in the base when this provision is made. Puddle the grout as poured, working as much as possible under the base and into the sleeves around the foundation bolts. Ideally, the complete space under the base is filled to the height of the grout around the base. After the grout is poured, keep covered with wet burlap for 48 hours to effect slow drying and prevent cracking. When the grout is set sufficiently, the frame may be removed and the grout finished as desired. Tighten foundation bolt nuts 72 hours after grouting. For epoxy-base grout, follow the manufacturer's recommendations for mix and cure time.

3.4.3.7 Rotation check

The direction of rotation of the pump is either clockwise (CW) or counterclockwise (CCW) when viewed facing the shaft end of the pump.

Most pump drivers are induction motors, and direction of rotation is dependent on the connection of the three-phase wire connection. Some drivers, such as unidirectional motors, engines, and turbines must be carefully checked to ensure that their direction of rotation matches the pump.

It is imperative that induction motors be correctly wired to provide proper rotation of the pump. Pump rotation is specified by a directional arrow, nameplate, or other means of designation on the pump.

Because alignment follows rotation check, and most pumps should not be run dry, it is usually best to separate pump and driver for the rotation check.

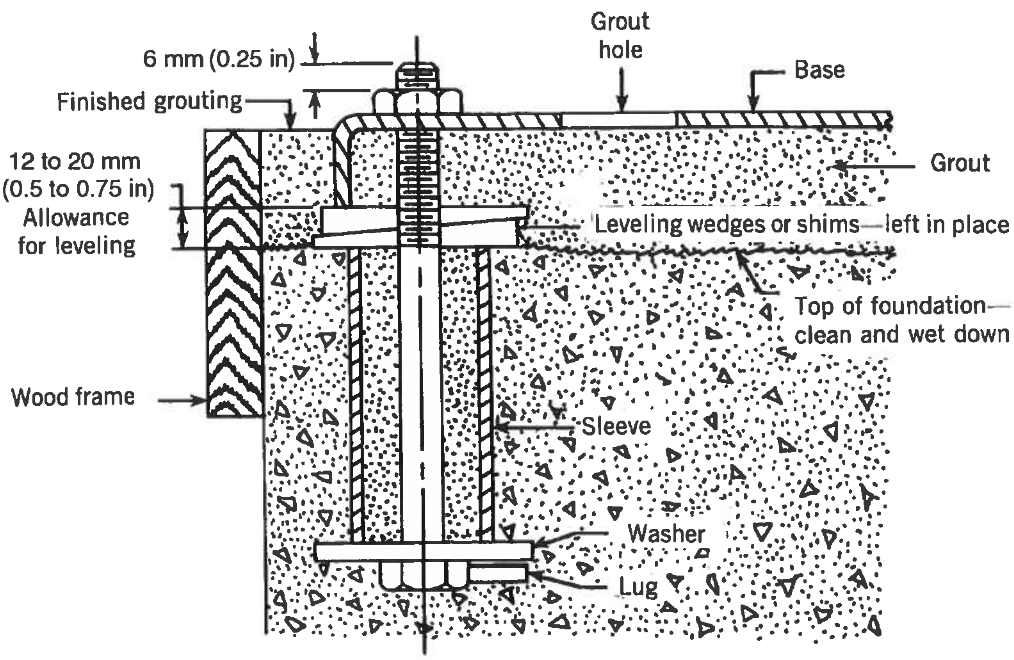


Figure 3.4.3.6 — Leveling and grouting

Remove coupling or belt safety guards. For belt-driven units, remove the belts. For couplings, remove the covers, then remove the chain, gear, or steel grid connecting coupling halves. Flex member couplings will require loosening of the coupling set screw on one half, sliding the coupling half back on the shaft, and removing the flex member.

The driver rotation is then checked against the required pump rotation. When a source of power is not yet available, the rotation check may be delayed until later. When the rotation check is delayed, the couplings or belts will probably be installed. In this case, the check will probably be made without liquid in the piping system and the check must be of short duration (a few turns of the pump shaft).

3.4.3.8 Alignment

All pump units are aligned at the factory during assembly, and in accordance with standard manufacturing practice, shims are used under the driver to provide alignment between pump and driver. Because alignment is critical to good pump performance and is frequently disturbed during shipment, handling, or installation, field alignment is usually necessary before the pump is put into operation.

Misalignment of the pump and driver shafts may be angular (shaft axes concentric but not parallel), parallel (shaft axes parallel but not concentric), or a combination of the above, as shown in Figure 3.4.3.8.

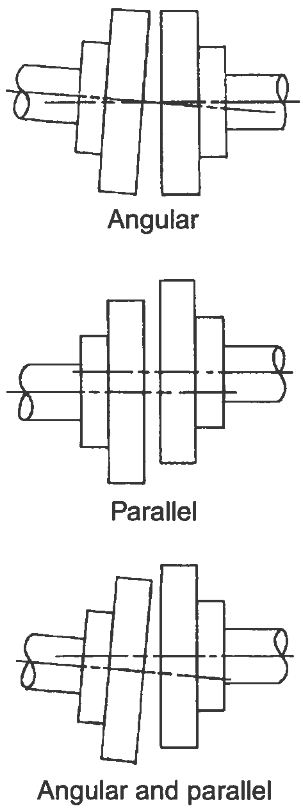


Figure 3.4.3.8 — Types of misalignment

3.4.3.9 Couplings

Couplings are intended to provide a mechanically flexible connection for two shaft ends in line. Additionally, they provide limited shaft end float (for mechanical movement or thermal expansion) and, within prescribed limits, angular and parallel misalignment of shafts. Couplings are not intended to compensate for major angular or parallel misalignment. The allowable misalignment varies with the type of coupling, and reference should be made to the manufacturer's literature enclosed with the shipment.

Flexible coupling types in general use are chain, gear, steel grid, and flex member. For aligning the pump shafts, remove the coupling cover, then remove the chain, gear, steel grid, or flex member connecting the coupling halves.

To check angular misalignment, as shown in Figure 3.4.3.9a, insert a feeler gauge between the coupling halves to check the spacing. Rotate the complete coupling one-quarter turn, one-half turn, and three-quarters turn, checking the spacing between coupling halves at the same location on the coupling as for the original spacing check. Checking the difference in spacing between coupling halves without rotating the complete coupling may result in an error because coupling faces are sometimes not machined nor are they square with the centerline of the shaft. The variation in spacing should not exceed the manufacturer's recommendations. To correct angular misalignment, adjust the amount of shims under the driver or adjust driver location in the horizontal plane.

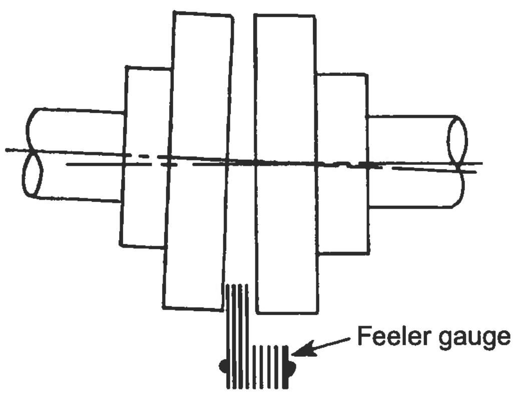


Figure 3.4.3.9a — Checking angular alignment

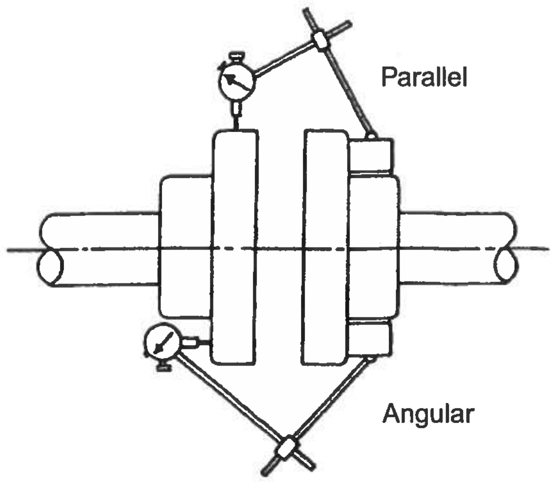


Figure 3.4.3.9b — Dial indicator method of alignment

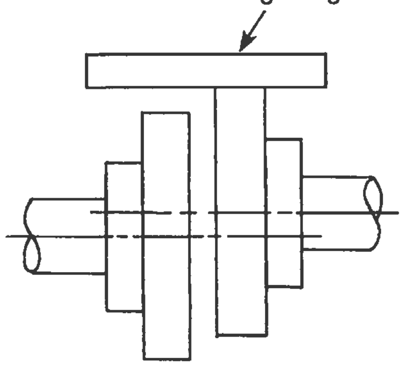


Figure 3.4.3.9c — Checking parallel alignment

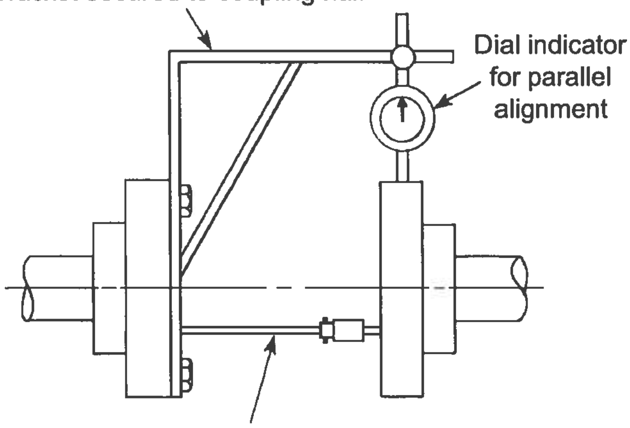


Figure 3.4.3.9d — Checking spacer coupling alignment

To check parallel misalignment, the dial indicator check as shown in Figure 3.4.3.9b is preferred. With the dial indicator secured to pump or driver shaft, rotate both shafts together, noting dial indicator readings through one complete revolution. As with angular misalignment, correct the parallel misalignment by adjusting the shims under the driver. Only when absolutely necessary should shims be adjusted or added under the pump. When a dial indicator is not available, an alternate check for a parallelism may be made by use of a straightedge, as shown in Figure 3.4.3.9c.

On certain large units, limited end float couplings are used, and the instruction manual furnished with such units should be consulted for special alignment instructions that apply to such couplings.

Spacer-type couplings may be checked for angular and parallel misalignment by the methods previously described after the spacer has been removed. Because of the distance between coupling halves, minor changes in the procedure are required. For the angular misalignment check, an inside micrometer replaces the feeler gauge. For the parallel misalignment check, a bracket should be attached to one coupling half for mounting of the dial indicator as shown in Figure 3.4.3.9d.

Following the check for parallel misalignment, the angular alignment must again be checked since angular misalignment may have resulted when correcting for parallel misalignment.

For applications where pumps are operated at elevated temperatures, as specified by the manufacturer, final alignment may not be possible at operating temperature. In this instance, proper allowance should be made for the increase in pump shaft height due to thermal expansion. As a rule of thumb (for cast-iron or steel pumps), a vertical allowance of 0.001 mm/mm (0.001 in/in) of pump shaft height above the base per 65°C (150°F) should be added to the height of the driver shaft.

Laser detector systems are used to determine the extent of shaft misalignment by measuring the movement of a laser beam across the surface of a detector plate as the shafts are rotated. Several different systems of lasers and detectors are used, and the procedure for alignment is provided by the laser system's producer.

3.4.3.9.1 Coupling guards

Before proceeding, after alignment is complete, make sure the coupling guards are properly installed.

3.4.3.10 V-belts and sheaves

V-belts and sheaves must be aligned and properly tensioned to eliminate the possibility of turn-over in the grooves, reduce belt wear, and transmit the motor power to the pump. Alignment should be checked, as shown in Figure 3.4.3.10, by placing a straightedge (or string) along the rim of both sheaves. If the sheaves are not of equal width, alignment can be checked by resting the straightedge across the rim of the widest sheave and measuring the distance from the straightedge to the nearest belt groove with a scale. Sheave adjustment on the shaft or driver adjustment on the base may be necessary.

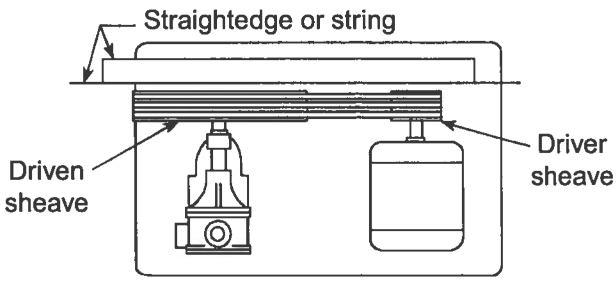


Figure 3.4.3.10 — V-belt sheave alignment

The driver should be moved on the slide rails toward the pump before the belts are installed if they have been removed. With the belts in place, move the driver away from the pump until proper belt tension is obtained. Good belt wear is dependent on proper tension. Proper tension is just beyond the point of slippage when the drive is operating at full load and speed. Slipping belts will squeal and cause overheating of the small sheave. Excessively tightened belts will result in reduced belt life and possible failure of bearings. Remount belt guards. It may be necessary to readjust belt tension after the first few days of operation due to initial belt stretch.

3.4.3.10.1 Belt guards

Before proceeding, after alignment is complete, make sure that the belt guards are properly installed.

3.4.3.11 Piping

Several precautions apply to all piping, either inlet or outlet. The user is referred to ANSI/HI 9.6.6 for additional recommendations on piping.

Because the piping system may require installation of accessory equipment, reference should be made to Section 3.4.3.12 for review of the subject before commencing with the piping installation.

Figure 3.4.3.11 illustrates pipe-to-pump alignment considerations.

Because rotary pumps are designed with close running clearances, clean piping is a must. Dirt, grit, weld bead, or scale, later flushed from an unclean piping system, will damage and may stall the pump.

Piping should be installed on supports independent of the pump. Supports must be capable of carrying the mass of the pipe, insulation, and the fluid carried. Supports may be hangers or stands, which, respectively, carry the mass from above or below. Clamps or brackets may be used to secure piping to existing columns. Supports must allow for free movement of the piping caused by thermal expansion or contraction. Supports should be installed at intervals such that piping load is uniformly and amply supported, precluding contact with adjacent piping and equipment. Pipe strains or stresses transmitted to the pump by improper piping support systems may cause pump distortion, wear, or binding of the rotary members and excessive power requirements. See Section 3.4.3.11.4, Nozzle loads.

Piping systems containing expansion joints must be designed so that the expansion joint is not exposed to motion greater than that for which the joint is designed. Expansion joints or flexible connectors should not be used to compensate for misaligned piping.

Threaded joints should be coated with compounds compatible with but not soluble in the liquid handled. Care must be taken with Teflon-taped joints to prevent shredded pieces of Teflon from entering the piping system. Piping should start at the pump, and work toward the source of supply and the point of discharge. Shutoff valves and unions (for pumps with tapped ports) are recommended to facilitate future inspection and repair. Reducers are preferred to bushings when a change in pipe size is necessary. Avoid unnecessary restrictions in the pipeline, such as elbows, sharp bends, globe or angle valves, and restricted-type plug valves.

Pipe size must be predetermined by taking into account the required rate of flow; minimum or maximum velocities; the fluid viscosity at the lowest pumping temperature; the length of the piping system, including valves, strainers, and other restrictions; and the elevation of the pump with reference to supply and discharge points. Friction or line losses may be calculated by reference to the manufacturer's engineering manual or the Hydraulic Institute's *Engineering Data Book*.

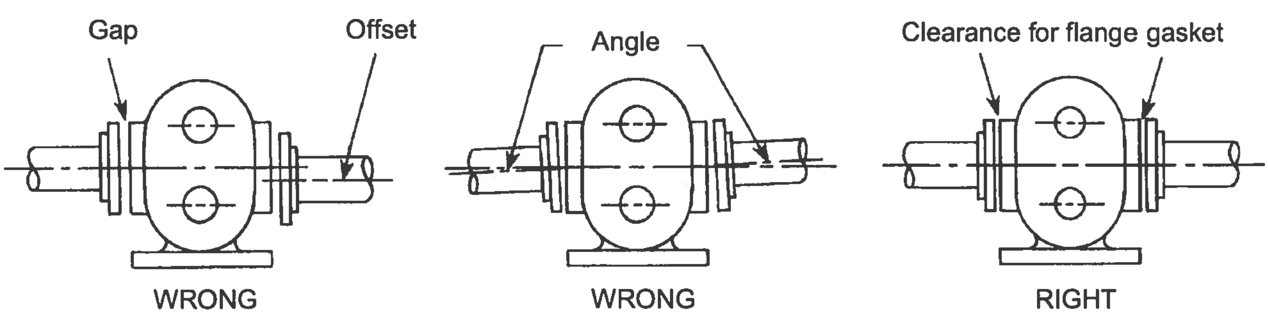


Figure 3.4.3.11 — Pipe-to-pump alignment

3.4.3.11.1 Inlet piping

Based on preceding paragraph factors, inlet piping should normally be equal to or one size larger than the inlet port connection. Viscous liquids may require larger inlet piping. Absolute airtightness of the inlet line is a must. Pumps operating with a static suction lift should have the inlet piping sloping gradually upward to the pump. On a long, horizontal run, keep the horizontal piping below the liquid level when possible. Piping should go around obstacles that may be encountered rather than over them, because piping a loop over an obstacle creates an air pocket, thereby making priming difficult.

3.4.3.11.2 Outlet piping

Outlet piping should be sized (by calculation) to accommodate the pump discharge pressure and flow.

3.4.3.11.3 Jacket piping

Jacket piping should take into account the heating media. For steam heating, the inlet is located at the top and the outlet on the bottom. For water and hot oil, the inlet is at the bottom and the outlet at the top. In accordance with the manufacturer's recommendation, valves may be installed in the jacket piping to regulate media flow.

3.4.3.11.4 Nozzle loads

The forces and moments applied to rotary pump inlet and outlet connections can affect operating clearances, casing stresses, mechanical seals, and alignment. Ideally, both suction and discharge piping should be independently supported near the pump so that when the flange bolts are tightened, no loading will be transmitted to the pump casing.

Due to the variations in rotary pump designs, it is not practical to provide recommendations for allowable nozzle loads that will apply to the different designs. Therefore, the pump manufacturer must be contacted for specific recommendations.

3.4.3.12 Accessory equipment

3.4.3.12.1 Foot valves

When pumping volatile low-viscosity liquids such as gasoline, a foot valve located near or at the end of the inlet line can prevent the inlet line from draining while the pump is idle, thus assisting in maintaining the pump prime. The size of the foot valve should be such that the net area of the valve opening exceeds the area of the inlet pipe.

When a foot valve is used in an installation where the inlet line is exposed to sunlight or artificial heat when the pump is idle, a small pressure relief valve set at 0.69 to 1.03 bar (10 to 15 psig) should be connected to the inlet line to permit the expanding fluid to return to the source.

3.4.3.12.2 Strainers

With a few exceptions, as noted in the manufacturer's literature, a pump should not be installed without strainer protection. Foreign matter may be large enough or of sufficient volume to jam a pump, resulting in probable damage to both pump and drive equipment. Smaller matter passing through the pump will cause rapid pump wear and premature pump failure.

Strainers should be of ample size to prevent an excessive vacuum condition at the pump. In general, the net area of the strainer screen should be three to five times the area of the inlet pipe. The required net area is dependent on the flow rate and the liquid viscosity. Strainer basket design must be capable of operation under high vacuum without collapsing.

Install the strainer in the inlet piping near the pump, making certain that it is located where it may be readily serviced for cleaning. The strainer must be installed according to arrows or notation designating direction of flow. Provide a vacuum or compound gauge before and after the strainer to determine when cleaning is required.

For continuous pumping, a bypass (including valving) should be piped around the strainer to permit cleaning. Alternately, two strainers in parallel or a duplex strainer may be used.

Check the strainer frequently when the pump is first started, since it is at this time that foreign material in the system is most likely to collect in the strainer.

3.4.3.12.3 Pressure relief valves

Because rotary pumps are positive displacement machines, the use of suitable pressure relief valves in the pump discharge piping is recommended to prevent damage to the pump or system from excessive pressures that might occur if the discharge piping becomes obstructed or a valve in the discharge piping is closed.

Various manufacturers refer to these as *relief valves, safety relief valves, pressure control valves,* or *pressure limiting valves.* Each is intended as a safety device to protect against excessive discharge pressure in the pump or outlet piping when the outlet line is restricted or closed. Unexpected high fluid viscosity may also cause a discharge pressure increase. A relief valve external to the casing or built-in is required for safe operation.

Relief valves are typically the spring-loaded, poppet-type design and may be either adjustable or nonadjustable. As discharge pressure is increased, a level of pressure is reached where the fluid pressure under the poppet overcomes the spring pressure and the poppet lifts off the seat (cracking pressure). When the discharge line is closed, all of the liquid passes through the relief valve. The pressure measured at the pump discharge with all liquid passing through the relief valve is known as the *relief valve full-flow* or *complete bypass setting.*

Relief valves for rotary pumps may be categorized in one of three styles.

- *Integral*: Secured to or designed directly into the pump. This style allows for liquid flow from the outlet to pass through the valve directly back to the pump inlet port.
- *Return-to-source*: Secured to or designed into the pump. This style allows for liquid flow from the outlet through the valve and, through separate piping, back to the source or to some point in the inlet line.
- *External*: Piped into the outlet line between the pump and any shutoff valve. This style allows for fluid flow from the outlet through the valve and, through separate piping, back to the source or to some point in the inlet line. External relief valves are often referred to as *in-line.*

Only the return-to-source and external styles should be used when the outlet line may be closed for more than a few minutes. Operation of a pump with an integral relief valve and a closed outlet line will cause overheating of the pump and a foamy discharge of fluid after the outlet line is reopened.

The difference in pressure between cracking and complete bypass will vary with valve design, valve capacity, fluid viscosity, and valve setting, thus the manufacturer should be consulted for this information. Operation of the pump at or near the cracking pressure may result in noisy operation due to valve chatter.

Most relief valve settings are differential pressure and, in applications involving high inlet pressures, the actual pressure in the outlet piping will be equal to the inlet pressure plus the bypass pressure setting. When integral relief valves are used on high-pressure systems, other pressure relief devices, such as external relief valves set to relieve at a predetermined gauge pressure, must be employed to protect against exceeding the manufacturer's maximum pressure limit for the pump or the piping system.

Most manufacturers provide pumps with their standard relief valve settings unless otherwise requested by the user. This standard setting is only correct for the test conditions at the time of manufacture.

3.4.3.12.4 Protective devices

Protective devices provide a pressure relief in any portion of a pump that can be valved off and thus completely isolated. This is particularly important when handling very cold fluids, such as refrigeration ammonia, that can warm up to ambient temperatures when the pump is shut down or when handling high-temperature fluids, such as asphalt, that must be heated before being pumped. Without provision for pressure relief, the expanding fluid may rupture the pump or piping.

Other forms of protective devices (automatic shutdowns, rupture discs, etc.) are considered part of the pumping system and are usually beyond the scope of the pump manufacturer's supply. These must be safely designed into, and supplied by, the system designer and/or by the user.

3.4.4 Operation

Instructions for operation vary widely among rotary pump manufacturers, primarily because of the wide variety of designs for the several types of pumps. Operating instructions published by the manufacturer should be followed. In lieu thereof, the following may be used as a general guide.

3.4.4.1 Prestart-up

Inspection checks are essential to avoid operational difficulties and ensure trouble-free start-up. Listed below are several items that need to be checked before the pump is started.

- Inspect all piping for undue stress and strain on the pump.
- Include a leak test on the piping. This may be done in conjunction with a line flush to remove foreign material.
- Flexible pipe connections should be a part of the piping system; they are imperative if the pump is to handle high-temperature fluids.
- Fully open inlet and outlet valves. Pump must not be started with throttled or closed inlet or outlet valves.
- Close all drain valves.
- Check shaft and gland clearance on packed pumps by removing the packing gland nuts and sliding the packing gland in and out of the stuffing box. It should move freely along the shaft and without interference between the gland and stuffing box. Reinstall packing gland nuts and snug them up, but do not tighten.
- Check wiring diagrams for proper connections for voltage and rotation. Proper rotation is indicated by an arrow on many pumps. Disconnect driver coupling and start driver momentarily to ensure correct rotation.
- Before reconnecting driver coupling, turn the pump shaft to be sure that it rotates freely.
  (Note: Free rotation may be difficult on certain pump types and sizes due to internal construction.)
- Reconnect driver coupling and replace guard. Make certain that all other guards are in place.
- For relief valves that can be installed for either direction of rotation, make certain that the valve is installed properly for the rotation desired.
- Install pressure and vacuum gauges to be used in checking start-up conditions.

3.4.4.2 Lubrication

For some pumps, timing rotation of the pumping elements is effected by means of a gear set contained in a gear box that is an integral part of the pump. Additionally, intermediate speed reducers are often used between the pump and driver. In both cases, oil lubrication is required. It is imperative that the manufacturer's literature be consulted to ensure that proper lubrication is used. Tags are often attached to the pump or unit to reflect the manufacturer's recommendations.

CAUTION: Do not assume that the pump or reducer gear box was shipped complete with lubricant. Most are not, and lubricant must be provided before start-up. Bearings, rolling element or journal, are usually factory-lubricated and usually do not require additional lubricant before start-up.

3.4.4.3 Start-up

Although rotary pumps are normally self-priming, most should never be operated dry. **CAUTION: Starting or running some rotary designs dry may cause galling, seizing, or destructive wear between the pumping elements and the casing.** For a pump with a negative inlet pressure (suction lift), fill the pump with liquid to seal the clearances and lubricate the pump during initial starting. Pumps started with only a small amount of fluid in the pump may require an air vent at the high point in the discharge piping. This vent should be kept open until the air is exhausted and fluid emerges.

The pump may now be started. Jogging the driver initially before continuous operation helps to determine if the pump unit has the proper rotation and is functioning properly. Normally a noise change is evident when the fluid first enters the pump. While the pump is running, check the unit for unusual noise or vibration. Investigate any abnormality, as it may indicate impending problems. Check vacuum and pressure gauges to see if the pump is operating within its prescribed limits or planned application conditions.

Controlled but measurable leakage along the shaft(s) is to be expected if the pump has packing as a shaft seal. Initially, allow for liberal leakage, in excess of 60 drops per minute, until the packing wears in. Then gradually adjust the packing gland until just enough leakage occurs to lubricate the packing, about 10 drops per minute. During adjustment, check for abnormal heating of the stuffing box, which may indicate that the gland has been overtightened. Check various locations, such as journal bearing areas and around the casing, for overheating.

Check the fluid flow rate to be sure full flow is being delivered. If not, check gauge readings against data calculated prior to installation and refer to Table 3.4.6, Malfunctions: Cause and remedy. If full flow is delivered, check power consumption to see if the driver is overloaded.

Start-up and operation of pumps used for elevated-temperature service requires some additional precautions. Sudden introduction of hot fluid into a cold pump causes uneven expansion of internal parts with resultant pump wear or failure. If the system in which the pump is to be used does not allow a gradual increase in temperature, an auxiliary means of heating is required. Pumps may include an integral jacket in the pump body. This can be used to preheat the pump and melt any material such as tars and asphalt that might solidify in the pump during shutdown. Consult the manufacturer's literature for limits of pressure and temperature when steam or heat transfer oils are the heating media.

Jackets are normally used for preheating the pump at start-up only, because once the pump is in operation, fluid moves through so rapidly that only an insignificant amount of heat is transferred to the fluid pumped. Where pump jackets are not provided, electrical heat tape wrapped around the pump body will effect a preheat condition. Insulation to cover the tape prevents escape of heat to the atmosphere.

3.4.4.4 Shutdown

The "after pumping" manufacturer's instructions vary with pump type with respect to kind of flush permitted or recommended (water, solvent, etc.), reverse rotation for drain and flush, and auxiliary pump flush. These instructions are important, especially for fluids that set up on contact with air or solidify with drop in temperature.

3.4.5 Maintenance

Reference to the pump manufacturer's literature for maintenance of the pump is imperative. Instructions may vary for pumps of the same type. The following is intended only as a guide or check for proper pump maintenance.

CAUTION: Never turn the shaft or work on a pump unless it has been isolated from the system, both electrically and hydraulically.

3.4.5.1 Preventive maintenance

A regular, documented, preventive maintenance program pays dividends in the form of efficient and dependable pump operation, as well as reducing the possibility of a breakdown in the pumping system. The recommended intervals for preventive maintenance checks may be listed in the owner's manual.

The importance of periodic lubrication of the externally lubricated bearings cannot be overemphasized. Oil-lubricated bearings should be serviced with the grade of lubricant recommended by the manufacturer. For grease lubrication, selection of the type of lubricant must be based on factors such as sub-zero operation and the handling of elevated-temperature fluids. Excessive bearing temperatures are often a result of overgreasing, and it is best to determine the cause of overheating before more grease is added. Some pumps are fitted with pressure relief fittings to prevent excessive lubrication and to allow for lubricant expansion as the bearings warm up during operation. Bearing vibration levels should be monitored regularly. A change in vibration levels should be evaluated to determine the root cause.

Instructions for proper lubrication of intermediate speed reducer (or increaser) gearing are usually affixed to the gear housing. As with pump lubrication, the correct lubricant depends on ambient operating temperatures. Extreme variations of temperature may necessitate use of more than one grade of lubricant during the year.

Filter elements should be cleaned and/or replaced at regular intervals. The clean or change interval can best be determined by monitoring the pressure drop across the filter.

Stuffing boxes should be checked regularly. Overheating is a result of overtightening of the packing gland in an attempt to minimize or eliminate leakage. If excessive leakage exists, the packing gland may be tightened (see Section 3.4.4.3) until the gland travel is exhausted, at which time the packing should be replaced. No single packing is good for all applications, and, for a pump not purchased to a specific set of application specifications, it may be necessary to consult the manufacturer for a packing recommendation for the actual operating conditions.

At less frequent intervals, foundation and hold-down bolts should be checked for tightness. Also, alignment of pump and driver coupling hubs should be checked, preferably just after a period of operation when the unit is at operating temperature. Couplings should be lubricated per manufacturer's recommendation.

3.4.5.2 Packing installation

Instructions for repacking a pump can only be general in nature due to the wide variety of stuffing-box designs and/or types of packings used. A pump should be repacked when all the packing gland travel is exhausted or when the condition of the packing requires replacement. To repack, remove the adjusting nuts and remove or back off the gland as far as possible. Remove all old packing rings with commercially available packing hooks. Clean out the stuffing box thoroughly. Lubricate the stuffing box and shaft. Use only clean packing. Packing rings may be furnished precut to size. If packing rings are cut from a coil, wrap the packing around the shaft and cut individual rings (angle-cut recommended) so that the ends of the rings lap or butt properly when installed in the stuffing box. Install one ring at a time, seating each ring firmly before installing another ring. Stagger joints 120° apart for most effective sealing. Where a lantern ring is used, place sufficient packing ahead of the lantern ring, so that it lines up with the hole for lubrication or sealing. The stuffing box should be filled until only 3 to 6 mm (0.125 to 0.25 in) is left for packing gland entry. Reinstall the packing gland and adjusting nuts. For proper gland adjustment, refer to Section 3.4.4.3. Worn or badly scored shafts should be replaced prior to installing new packing.

3.4.5.3 Mechanical seals

Mechanical seals are available in many different designs and used in a wide variety of applications, thus instructions for their use and replacement shall be carefully studied and followed exactly.

Mechanical seals are precision devices that do not require adjustment or maintenance under rated (normal) operating conditions. Except for possible slight initial leakage, the seal should operate with negligible leakage. Depending on the severity of the application, mechanical seals are subject to wear with time in operation and subsequent leakage. They may require repair or replacement periodically.

All mechanical seal designs are limited in application, variables of speed, type of fluid, fluid viscosity, differential pressure, and temperature. When pumps are used on applications where any of these variables are different from those for which the pump was originally furnished, consult the manufacturer for a seal recommendation for the new application.

For additional information see *Mechanical Seals for Pumps: Application Guidelines.*

3.4.5.4 Spare parts

When downtime for repair is of vital concern and must be minimized, a set of recommended spare parts or parts kits should be retained for emergencies. When ordering spare parts, the manufacturer's literature should be consulted to determine what information (serial number, model number, etc.) the manufacturer needs to furnish the proper replacements.

3.4.6 Malfunctions: Cause and remedy

The cause of a majority of pump or system malfunctions can be detected by determination of inlet and outlet conditions. For this purpose, install a vacuum or compound gauge near the pump inlet and a pressure gauge near the pump outlet. Predominant malfunctions, their causes and remedies are as follows:

Table 3.4.6 — Malfunctions: Cause and remedy

Malfunction	Probable cause	Remedy
No liquid discharged	Pump not primed	Prime from outlet side. Keep outlet air vent open until liquid begins to discharge.
	Wrong direction of rotation	Reverse wiring at junction box or motor.
	Valves closed or obstruction in inlet or outlet line	Open valves. See that flange gaskets have the center cut out.
	End of inlet pipe not submerged	Increase length of inlet pipe or raise liquid level in supply tank.
	Foot valve stuck	Inlet pipe may be screwed in too far, blocking valve from opening.
	Net inlet pressure too low	Check reading on vacuum gauge. Liquid viscosity may be high due to cold start-up.
	Bypass valve open	Check all valves for sticking or foreign material under seat.
	Air leak in inlet line	Treat and tighten threaded joints, check flanged joint gaskets and tighten.
	Strainer clogged	Remove basket, clean, make certain it has ample open area, reinstall or replace.
	Pump badly worn	Replace parts (excessive clearances may cause slip equal to pump displacement).
	Loose coupling, broken shaft, failed pump	Correct or repair as required.
Low discharge rate	Net inlet pressure too low	Liquid not up to required temperature, causing high viscosity. Liquid more viscous than specified.
	Strainer partially clogged or of insufficient area	Clean and/or replace as required.
	Air leak in inlet line	Treat and tighten threaded joints, check flanged joint gaskets and tighten.
	Air leak through packing	Tighten gland or replace packing. Add water seal line to stuffing box.
	End of inlet line not sufficiently submerged, causing eddies and air entry to pump	Increase length of inlet pipe. Raise liquid level in supply tank. Install antivortex baffles.
	Starving or cavitating	Increase size and/or reduce length and simplify the inlet line.
	Bypass valve partially open	Remove foreign material from under poppet seat. Increase relief valve setting so valve does not open at desired discharge pressure for the system but rather protects the system from excessive pressure.
	Speed too low. Motor may be wired improperly or overloaded.	Check drive speed ratio.
	Pump worn	Replace parts and adjust clearances to eliminate excessive slip.

Table 3.4.6 — Malfunctions: Cause and remedy (*continued*)

Malfunction	Probable cause	Remedy
Loss of prime (after satisfactory operation)	Liquid supply exhausted	Replenish liquid supply.
	Substantial increase in liquid viscosity	Check malfunction of supply tank heating.
	Liquid vaporizes in inlet line. Liquid may be overheated.	Vaporization due to drop in net inlet pressure as a result of drop in liquid supply level requires change in inlet system.
	Air leaks developed in suction line	Treat and tighten threaded joints, check flanged joint gaskets and tighten.
	Relief valve stuck open	Check relief valve discharge.
Excessive power consumption	Pump running too fast	Check drive equipment including motor speed, reducer or increaser ratio and sheave sizes for proper speed.
	Higher liquid viscosity than specified	Reduce speed or heat liquid to reduce viscosity.
	Discharge pressure higher than calculated	Increase size of and simplify outlet line. Decrease liquid viscosity by heating. Reduce pump speed.
	Improperly adjusted packing gland (too tight) causing drag on shaft	Make adjustment to gland. Repack.
	Rotating element binding from misalignment	Correct pump-to-motor alignment. Check binding of pump due to improper pump-to-piping alignment.
	Where required, the extra clearances on rotating elements are insufficient for the liquid viscosity	Check manufacturer for recommendation.
Excessive noise-vibration	Pump cavitation due to vaporization in inlet line	Increase pipe inlet size. Reduce length. Increase net inlet pressure by raising liquid supply level. Change vertical position of pump.
	Pump starved on high-viscosity liquid	Make changes as for cavitation.
	Misalignment conditions	Correct, according to Sections 3.4.3.8 to 3.4.3.10.
	Relief valve chatter	Change pressure setting or check size and type of relief valve.
	Foundation and/or hold-down bolts loose	Tighten.
	Bearings failing	Replace.
	Piping inadequately supported	Add supports as required.
	Variable-speed drive	Speed setting may be on natural frequency of the rotodynamic system. Change speed setting.

Table 3.4.6 — Malfunctions: Cause and remedy (*continued*)

Malfunction	Probable cause	Remedy
Rapid pump wear	Abrasives in liquid	For closed system, check strainer size and mesh. For transfer system, consult manufacturer. Application may dictate expendable pump.
	Corrosion of pump elements	Consult manufacturer for proper materials of construction.
	Misalignment conditions	Correct, according to Sections 3.4.3.8 to 3.4.3.10.
	Pump runs dry. Pump stalls due to frictional heat generated.	Provide ample supply of liquid at all times. Provide automatic alarm or shut-off where fluid supply may fail.
	Lack of lubrication	Follow specified lubrication instructions to minimize wear.
	High discharge pressure	Consider changing pump materials or model.

3.5 Reference and source material

3.5.1 ASTM

ASTM D-446, *Specification for Operating Instructions for Glass Capillary Viscometer*

ASTM D-2161, *Practice for Conversion of Kinematic Viscosity to Sabolt Universal Viscosity or to Sabolt Furol Viscosity*

ASTM International, 100 Barr Harbor Drive, West Conshohocken, PA 19428-2959, www.astm.org.

3.5.2 Hydraulic Institute

Hydraulic Institute: *Engineering Data Book*

Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems

Variable Speed Pumping: A Guide to Successful Applications

ANSI/HI 9.1–9.5 *Pumps – General Guidelines*

ANSI/HI 9.6.6 *Rotodynamic Pumps for Pump Piping*

ANSI/HI 9.6.7 *Effects of Liquid Viscosity on Rotodynamic (Centrifugal and Vertical) Pump Performance*

Hydraulic Institute, 6 Campus Drive, First Floor North, Parsippany, NJ 07054-4406, www.Pumps.org.

3.5.3 3-A Sanitary Standards

3-A Sanitary Standards, Inc.

6888 Elm St. Suite 2D, McLean, Virginia 22101-3880

Phone: 703-790-0295, Fax: 703-761-6284, www.3-a.org

3.5.4 API

American Petroleum Institute (API)

1220 L. Street NW, Washington, DC 20005-4070

Phone: 202-682-8000, www.api.org

3.5.5 Igor J. Karassik, et al

Pump Handbook

McGraw Hill

3.5.6 Cameron Hydraulic Data

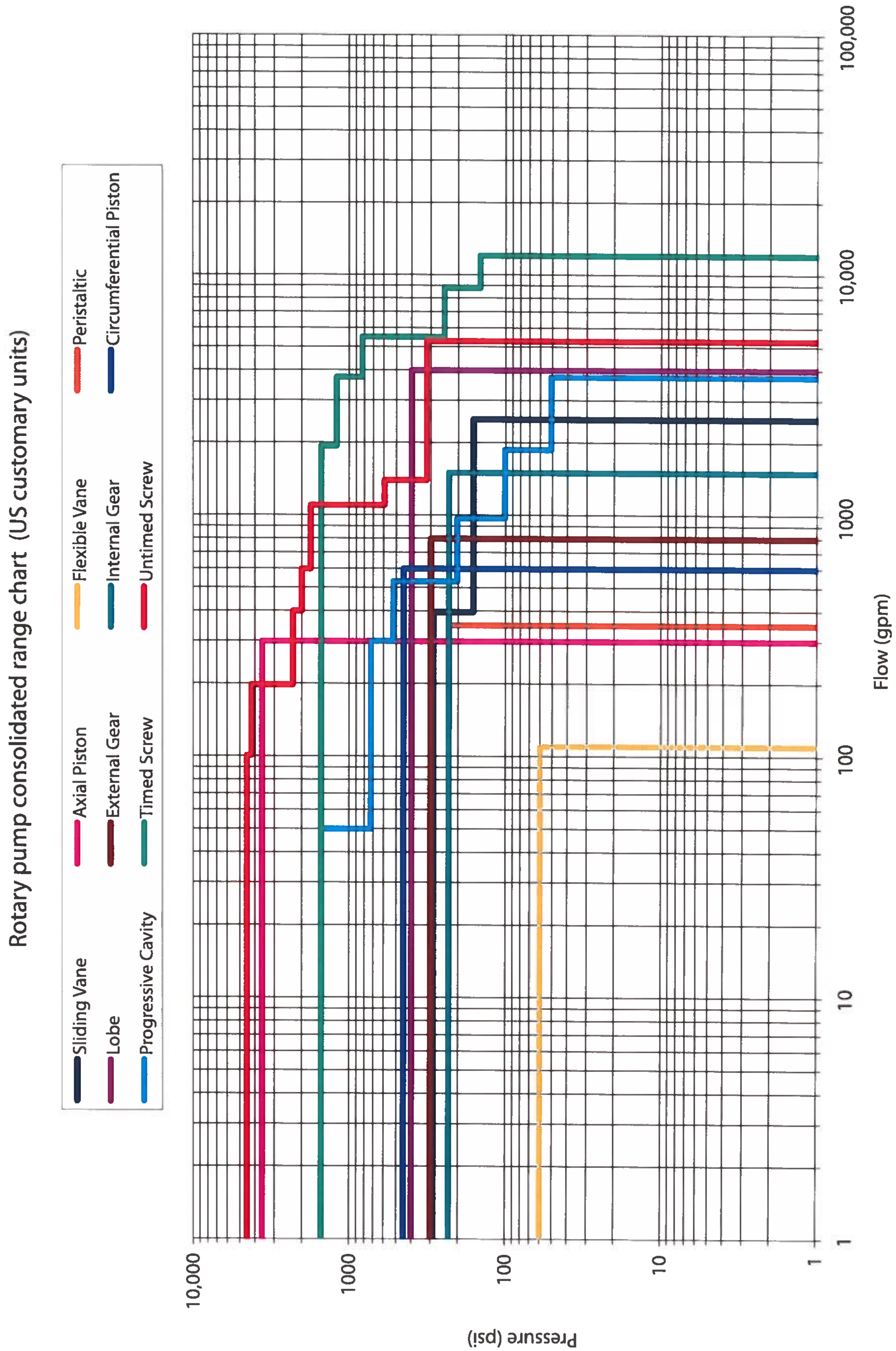
FLOWSERVE Pumps

Appendix A

Index

This appendix is not part of this standard, but is presented to help the user with factors referenced in the standard.

Note: an f. indicates a figure, and a t. indicates a table.



Rotary pump consolidated range chart (US customary units)
Sliding Vane
Axial Piston
Flexible Vane
Peristaltic
Lobe
External Gear
Internal Gear
Circumferential Piston
Progressive Cavity
Timed Screw
Untimed Screw
Pressure (psi)
10,000
1000
100
10
1
Flow (gpm)
1
10
100
1000
10,000
100,000

Hydraulic Institute
Creating Pump Standards Since 1917

Available at eStore.Pumps.org

Optimizing Pumping Systems: A Guide to Improved Efficiency, Reliability and Profitability

Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems

Variable Speed Pumping: A Guide to Successful Applications

Mechanical Seals for Pumps: Application Guidelines

ANSI/HI Pump Standards

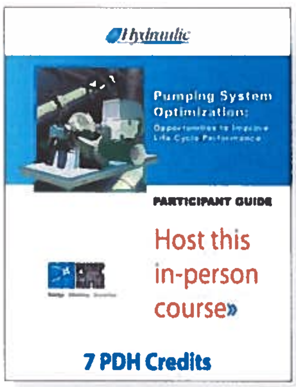
Individual Standards

- Hardcopy
- Downloadable

Complete Set — Hardcopy

ANSI/HI Pump Standards on CD — complete set

ANSI/HI Pump Standards by Subscription



Pumping System Optimization Course
Benefit from Major Energy Efficiency Improvements and Bottom-line Savings.

Rotodynamic (Centrifugal) Pumps: Fundamentals, Design and Applications Online Course

Positive Displacement Pumps: Fundamentals, Design and Applications Online Course

Click on the text or image to go to the eStore page for the product of your choice.

www.Pumps.org | www.PumpSystemsMatter.org | www. PumpLearning.org