ANSI/HI 6.1–6.5-2000

American National Standard for

Reciprocating Power Pumps

for Nomenclature, Definitions, Application, and Operation

ANSI/HI 6.1–6.5-2000

Hydraulic Institute

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

This page intentionally blank.

ANSI/HI 6.1–6.5-2000

American National Standard for

Reciprocating Power Pumps

for Nomenclature, Definitions, Application, and Operation

Secretariat
Hydraulic Institute
www.Pumps.org

Approved March 15, 2000
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CAUTION NOTICE: This American National Standard may be revised or withdrawn at any time.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 1-880952-39-4

Recycled paper

Contents

Page

Foreword vii

6 Reciprocating power pumps

6.0 Scope 1
6.1 Types and nomenclature 1
6.1.1 Horizontal pump 1
6.1.2 Vertical pump 1
6.1.3 Piston pump 1
6.1.4 Plunger pump 1
6.1.5 Single-acting pump 1
6.1.6 Double-acting pump 1
6.1.7 Simplex pump 2
6.1.8 Duplex pump 2
6.1.9 Multiplex pump 2
6.1.10 Description of components 2
6.1.11 Liquid end 5
6.1.12 Power end 13
6.2 Definitions 20
6.2.1 Flow rate 20
6.2.2 Pressures 20
6.2.3 Power (P) 23
6.2.4 Efficiencies (η) 23
6.2.5 Pistons, plungers and valves 23
6.2.6 Suction conditions 24
6.2.7 Slurry 27
6.3 Design and application 29
6.3.1 Typical services 29
6.3.2 Basic speeds 29
6.3.3 Discussion of speeds 33
6.3.4 Starting power pumps 34
6.3.5 Electric motor locked-rotor torques 38
6.3.6 Inlet system for power pumps 38
6.3.7 Discharge piping 45
6.3.8 Calculating volumetric efficiency for water (η_V) 45
6.3.9 Calculating volumetric efficiency for hydrocarbons (η_V) 47
6.3.10 Piston and plunger pumps for slurry service 53

6.4 Installation, operation and maintenance . . . 55
6.4.1 Safety . . . 55
6.4.2 Storage . . . 55
6.4.3 Location of pump . . . 55
6.4.4 Protection against seepage or flood . . . 55
6.4.5 Provision for servicing space . . . 55
6.4.6 Foundation . . . 55
6.4.7 Installation . . . 56
6.4.8 Plunger or piston rod packing installation . . . 60
6.4.9 Cup type pistons . . . 64
6.4.10 Installation . . . 64
6.4.11 Inspection . . . 65
6.4.12 Malfunctions, cause and remedy . . . 66
6.5 Reference and source material . . . 69
6.5.1 NEMA-MG1-1993, Motors and Generators . . . 69

Appendix A Index . . . 70

Figures

6.1 — Types of reciprocating power . . . 1
6.2 — Horizontal single-acting plunger power pump . . . 1
6.3 — Vertical single-acting plunger power pump . . . 2
6.4 — Horizontal double-acting piston power pump . . . 2
6.5 — Horizontal double-acting plunger power pump . . . 3
6.6 — Horizontal triplex plunger pump, on base, belt drive . . . 3
6.7 — Vertical triplex plunger pump, on base, gear reduction . . . 4
6.8 — Liquid cylinder . . . 5
6.9 — Manifold . . . 5
6.10 — Piston assembly . . . 6
6.11 — Bull and snap piston . . . 6
6.12 — Cup type piston . . . 6
6.13 — Slush piston . . . 6
6.14 — Individual ring piston . . . 6
6.15 — Plunger . . . 7
6.16 — Stuffing box . . . 7
6.17 — Packing . . . 7
6.18 — Gland . . . 7
6.19 — Lantern ring . . . 7
6.20 — Plate valve assembly . . . 8
6.21 — Disc valve assembly . . . 8
6.22 — Wing guided valve assembly . . . 8

6.23 — Ball valve assembly . . . 8
6.24 — Upper crosshead . . . 8
6.25 — Liquid end, horizontal plunger power pump . . . 9
6.26 — Liquid end, vertical plunger power pump . . . 10
6.27 — Liquid end, horizontal side pot piston pump . . . 11
6.28 — Liquid end, horizontal valve plate piston pump . . . 11
6.29 — Power frame (one piece) . . . 13
6.30 — Crankshaft . . . 13
6.31 — Sleeve bearing . . . 13
6.32 — Tapered roller bearing . . . 14
6.33 — Connecting rod . . . 14
6.34 — Crankpin bearing . . . 14
6.35 — Power crosshead . . . 14
6.36 — Wrist pin bearing . . . 15
6.37 — Crosshead extension . . . 15
6.38 — Frame extension . . . 15
6.39 — Power end, horizontal plunger power pump . . . 16
6.40 — Power end, vertical plunger power pump . . . 17
6.41 — Power end, horizontal duplex power pump with integral gears . . . 18
6.42 — Disc valve . . . 24
6.43 — Plate valve . . . 24
6.44 — Wing guided valve . . . 24
6.45 — Ball valve . . . 25
6.46A — Percent of basic pump speed as a function of average liquid velocity through suction valve (liquid velocity before derating) (Metric units) . . . 31
6.46B — Percent of basic pump speed as a function of average liquid velocity through suction valve (liquid velocity before derating) (US units) . . . 32
6.47 — Schematics of liquid bypass systems . . . 36
6.48 — Suction tanks . . . 38
6.49 — Recommended installation of multiple pumps to common manifolds . . 39
6.50 — Recommended connection of piping sections . . . 39
6.51 — Installation of eccentric reducers . . . 40
6.52 — Startup strainers . . . 40
6.53 — Suction system relationships – open supply . . . 42
6.54 — Suction system relationships – closed supply . . . 43
6.55 — Suggested piping system for power pumps . . . 46
6.56 — Plunger movement when calculating volumetric efficiency . . . 49
6.57 — Thermal expansion and compressibility of liquids . . . 51
6.58 — Foundation bolt data . . . 56
6.59 — Correct tension for V-belt drives . . . 60
6.60 — Piston packing joints . . . 62

6.61 — Hydraulic packing .63
6.62 — Bull ring packing .63
6.63 — Cup type packing .65
6.64 — Assembling cup piston .65
6.65 — Correct and incorrect piston rod nut tightening .65

Tables

6.1 — List of parts by key name – liquid end parts 12
6.2 — List of parts by key name – power end parts 19
6.3 — Symbols 21
6.4 — Subscripts 22
6.5 — Barometric pressure versus elevation 23
6.6 — Minimum locked-rotor torque ratings 39
6.7 — Water compressibility factor $\beta_t \times 10^{-6}$ (US units) 48
6.8 — Water compressibility $\beta_t \times 10^{-6}$ (US units) 49
6.9A — Physical properties of hydrocarbons (Metric) 50
6.9B — Physical properties of hydrocarbons (US Units) 52
6.10 — Suggested trial set pressures of pump relief valves 58
6.11A — Proper spring pull tension for new and used belts (Metric) 59
6.11B — Proper spring pull tension for new and used belts (US units) 60
6.12 — Maximum temperature for ring materials 63
6.13 — Maximum concentration of chemicals for phenolic type rings 64
6.14 — Malfunctions – cause and remedy 66

Foreword (Not part of Standard)

Purpose and aims of the Hydraulic Institute

The purpose and aims of the Institute are to promote the continued growth and well-being of pump manufacturers and 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 directed to the Hydraulic Institute. It will direct all such questions to the appropriate technical committee for provision of a suitable answer.

If a dispute arises regarding contents of an Institute publication or an answer provided by the Institute to a question such as indicated above, the point in question shall be referred to the Executive Committee of the Hydraulic Institute, which then shall act as a Board of Appeals.

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; corresponding US units appear in brackets. Charts, graphs and sample calculations are also shown in both metric and US units.

Since values given in metric units are not exact equivalents to values given in US 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 for this standard was achieved by use of the Canvass Method

The following organizations, recognized as having an interest in the standardization of centrifugal pumps 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.

Bal Seal Engineering
Black & Veatch LLP
Bran & Luebbe
Brown & Caldwell
Camp Dresser & McKee, Inc.
Cheng Fluid Systems, Inc.
David Brown Union Pumps
DeWante & Stowell
Equistar LP
Exeter Energy Limited Partnership
Fluid Sealing Association
Illinois Department of Transportation
Ingersoll-Dresser Pump Company
Krebs Consulting Service
Malcolm Pirnie, Inc.
Marine Machinery Association
McFarland Pump Company
Pacer Pumps
Pinellas County, Gen. Serv. Dept.
The Process Group, LLC
Raytheon Engineers & Constructors
Skidmore
South Florida Water Mgmt. Dist.
Stone & Webster Eng. Corp.
Summers Engineering, Inc.
Systecon, Inc.
Tuthill Pump Group

6 Reciprocating power pumps

6.0 Scope

This Standard applies to reciprocating power pumps. It includes types and nomenclature; definitions; design and application; and installation, operation and maintenance.

6.1 Types and nomenclature

A reciprocating power pump is one driven by power from an outside source applied to the crankshaft of the pump. It consists of a liquid end and a power end.

6.1.1 Horizontal pump

The axial centerline of the cylinder is horizontal (see Figure 6.2).

6.1.2 Vertical pump

The axial centerline of the cylinder is vertical (see Figure 6.3).

6.1.3 Piston pump

The liquid end contains pistons (see Figure 6.4).

6.1.4 Plunger pump

The liquid end contains plungers (see Figures 6.2, 6.3 and 6.5).

6.1.5 Single-acting pump

Liquid is discharged only during the forward stroke of the plunger or piston, that is, during one half of the revolution of the crankshaft (see Figures 6.2 and 6.3).

6.1.6 Double-acting pump

Liquid is discharged during both the forward and return strokes of the piston or pair of opposed plungers. That is, discharge takes place during the entire revolution of the crankshaft (see Figures 6.4 and 6.5).

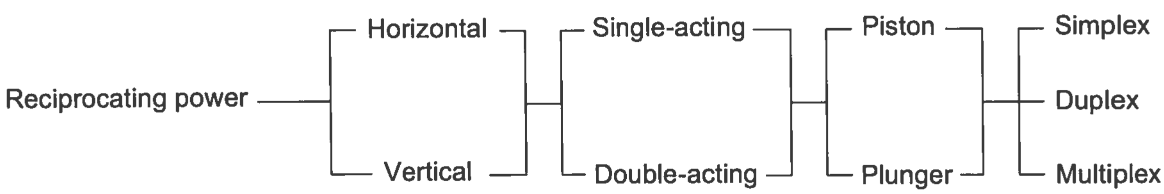


Figure 6.1 — Types of reciprocating power

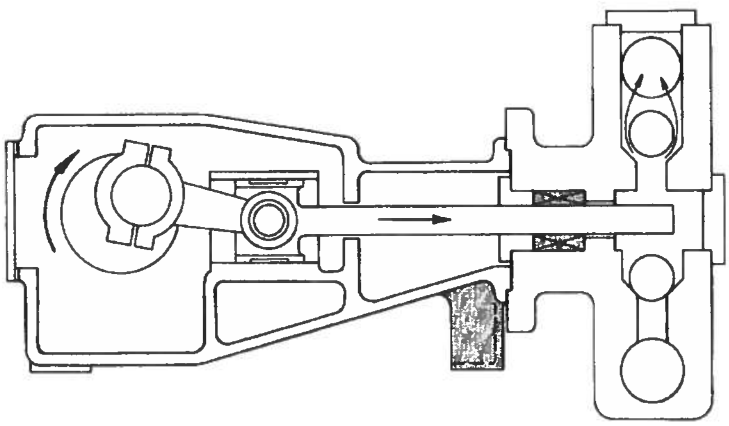


Figure 6.2 — Horizontal single-acting plunger power pump

6.1.7 Simplex pump

Contains one piston or one plunger or a pair of opposed plungers driven by one connecting rod (see Figures 6.4 and 6.5).

6.1.8 Duplex pump

Contains two pistons or two plungers or two pair of opposed plungers driven by two connecting rods.

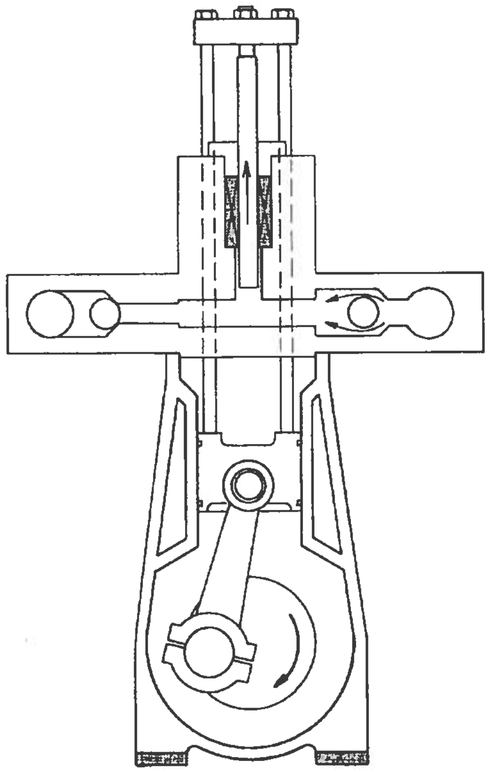


Figure 6.3 — Vertical single-acting plunger power pump

6.1.9 Multiplex pump

Contains more than two pistons or two single-acting or opposed plungers.

Number of Power Crossheads	Type pump
1	Simplex
2	Duplex
3	Triplex
5	Quintuplex
7	Septuplex
9	Nonuplex

6.1.10 Description of components

The nomenclature and definitions in these standards were prepared to provide a means for identifying the various pump components included in these Standards and also to serve as a common language for all who deal with this type of equipment.

The following definitions and drawings illustrate typical construction of reciprocating power pump components but do not necessarily represent recommended designs. Variations in design may exist without violating the intent of these Standards.

6.1.10.1 Right and left hand shaft extension of power pumps

"Right" or "left hand" designates the side of the power end (see Section 6.1.11) from which the crankshaft or

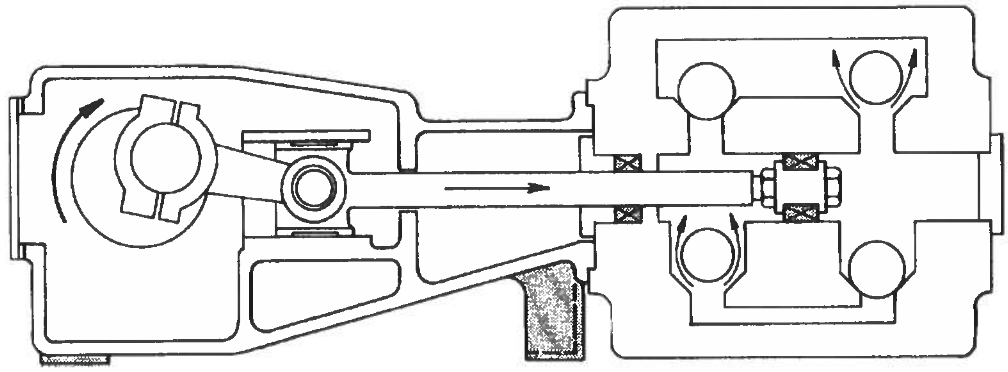


Figure 6.4 — Horizontal double-acting piston power pump

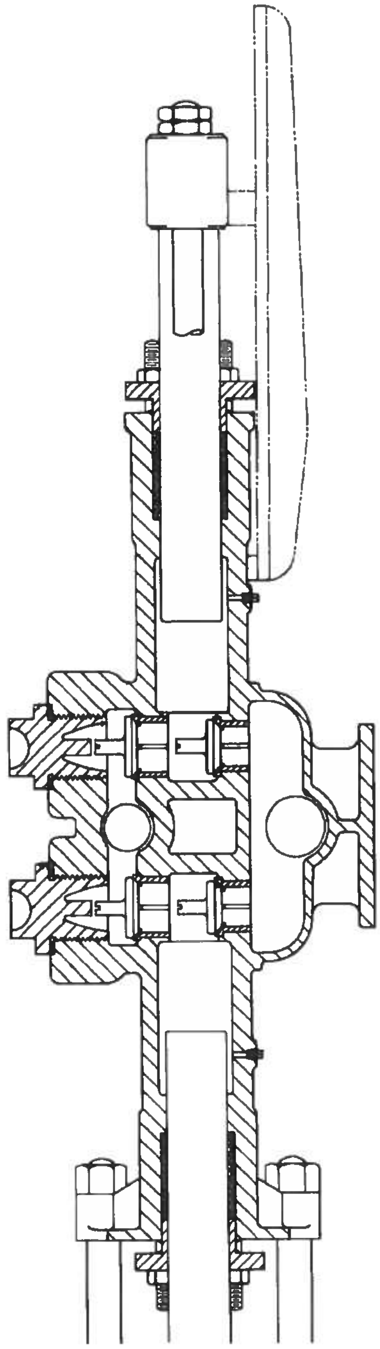


Figure 6.5 — Horizontal double-acting plunger power pump

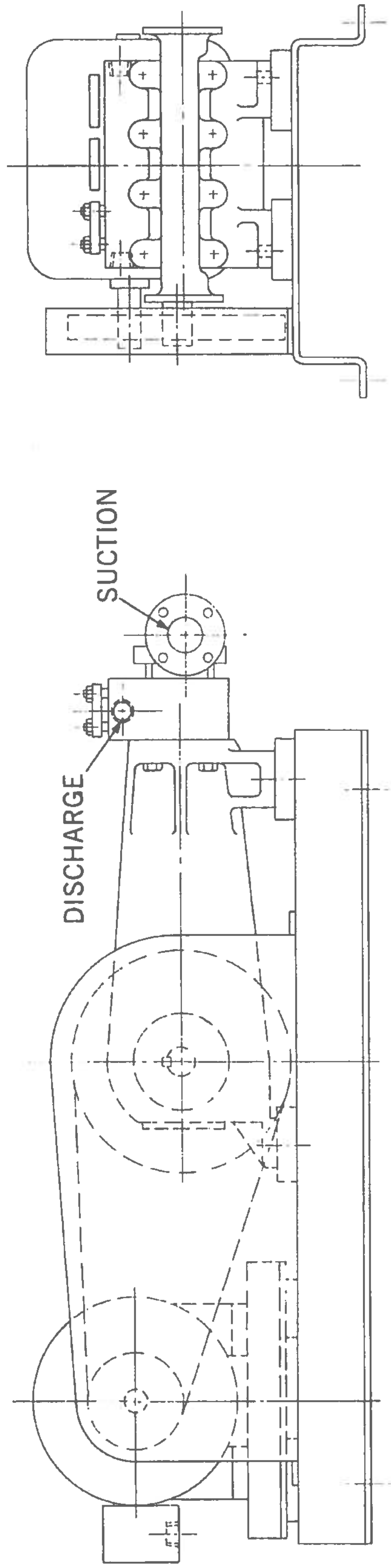


Figure 6.6 — Horizontal triplex plunger pump, on base, belt drive

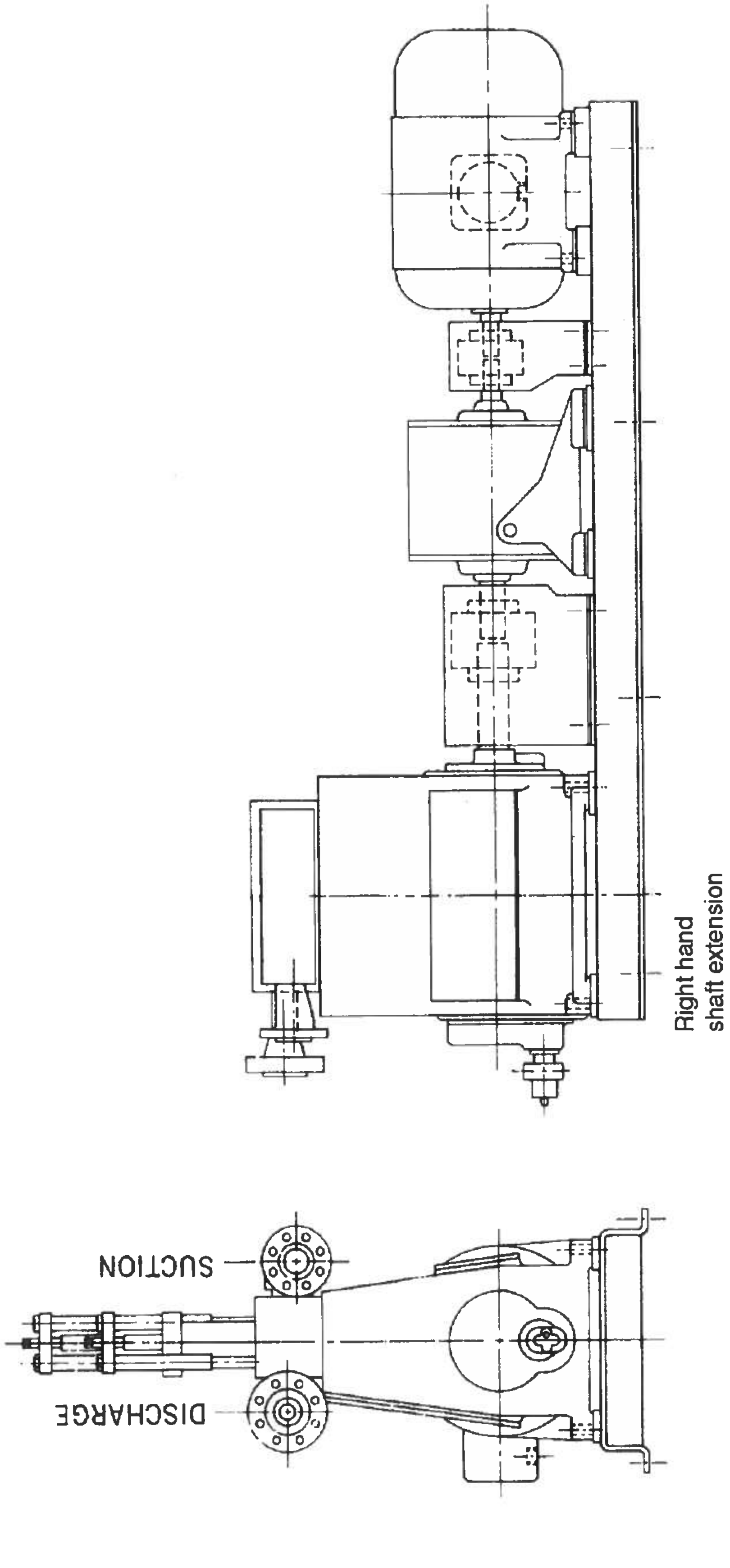


Figure 6.7 — Vertical triplex plunger pump, on base, gear reduction

pinion shaft extends. (It does not designate in which direction the shaft rotates.)

Horizontal power pumps are termed right hand or left hand as viewed when standing behind the power end with the liquid end being the most distant part. A left hand pump has the shaft extending out of the left side of the power end. A right hand pump has the shaft extending out of the right side of the power end (see Figure 6.6).

Vertical power pumps are termed right hand or left hand pumps as viewed when standing at and facing the suction manifold of the pump. A left hand pump has the shaft extending out of the left side of the power end. A right hand pump has the shaft extending out of the right side of the power end (see Figure 6.7).

6.1.11 Liquid end

That portion of the pump which handles the liquid. It consists of a liquid cylinder, valves and other components.

6.1.11.1 Liquid end parts

The following sections describe major liquid end components which are shown in Figures 6.24 through 6.27. A listing by part name is shown in Table 6.1, which also shows the standard abbreviation.

6.1.11.2 Liquid cylinder

A chamber(s) in which the motion of the plunger(s) or piston(s) is imparted to the liquid. The cylinder can be made integral with a suction and discharge manifold or can be made with separate manifolds (see Figure 6.8).

6.1.11.3 Cylinder liner

A replaceable liner which is placed in the cylinder of a piston pump. The piston reciprocates within the liner (see Figures 6.27 and 6.28).

6.1.11.4 Manifolds

A suction manifold is a chamber which accepts liquid from the suction port(s) and distributes it to the suction valves (see Figure 6.9).

A discharge manifold is a chamber which accepts liquid from the individual discharge valves and directs it to the discharge port(s) (see Figure 6.9).

6.1.11.5 Valve chest cover

A cover for the valves within the cylinder (see Figure 6.28).

6.1.11.6 Valve plate (valve deck)

A plate that contains the suction or discharge valves (see Figure 6.28).

6.1.11.7 Piston

A cylindrical body which is attachable to a rod and is capable of exerting pressure upon a liquid within the liquid cylinder. A piston usually has grooves for containing rings which seal against the cylinder or cylinder liner.

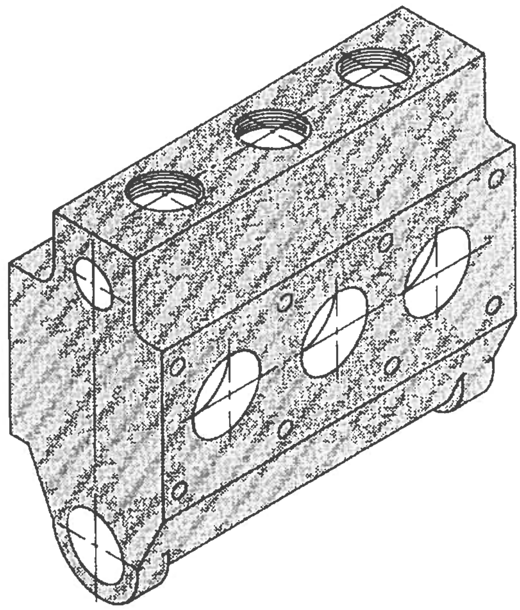


Figure 6.8 — Liquid cylinder

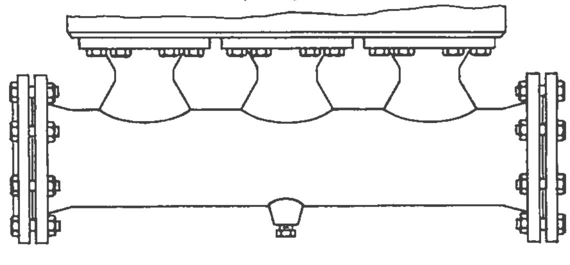


Figure 6.9 — Manifold

A piston in a reciprocating pump can be double-acting.

The pistons in Figures 6.10 and 6.11 have followers which retain the packing rings.

6.1.11.8 Plunger

A smooth rod which is attachable to a crosshead and is capable of exerting pressure upon a liquid within the liquid cylinder (see Figure 6.15). Sealing rings for a plunger are stationary, the plunger sliding within the rings.

A plunger is single-acting, requiring a double-acting pump to have two plungers on each crosshead axis (see Figure 6.6).

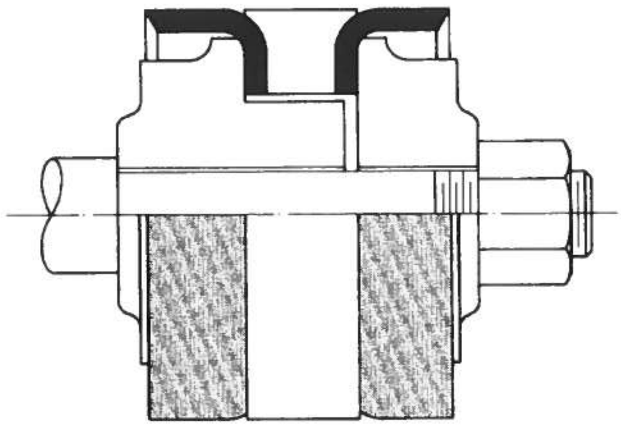


Figure 6.12 — Cup type piston

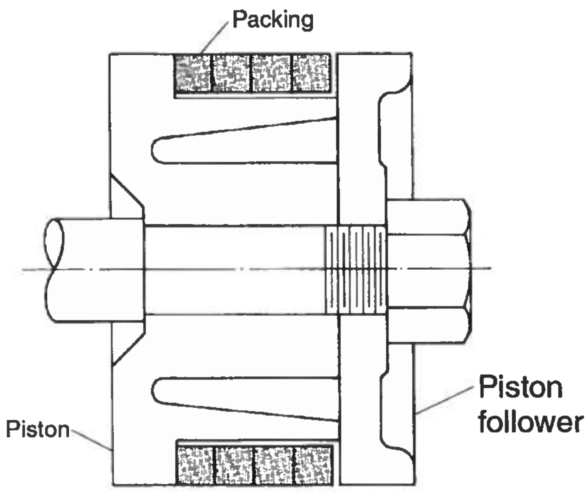


Figure 6.10 — Piston assembly

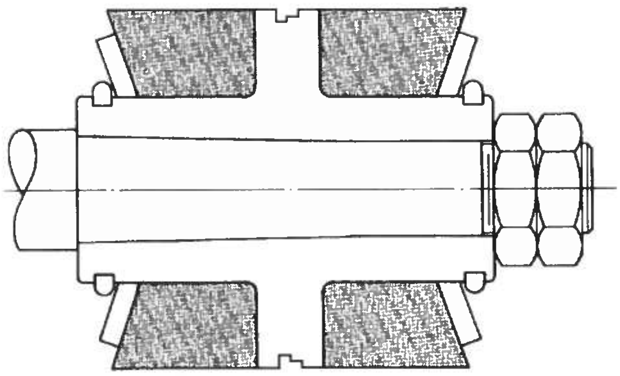


Figure 6.13 — Slush piston

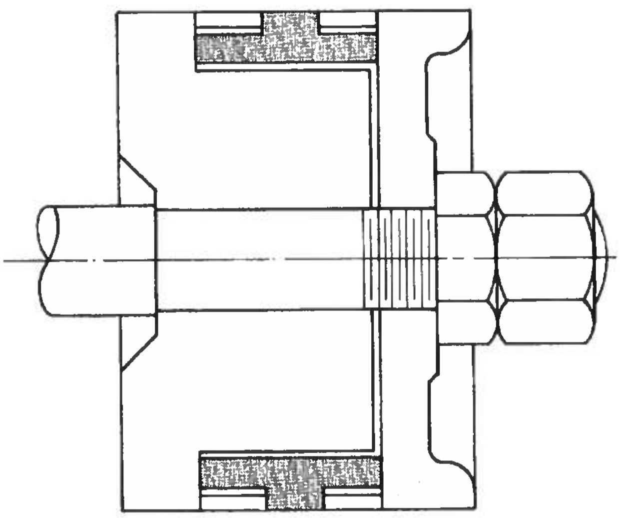


Figure 6.11 — Bull and snap piston

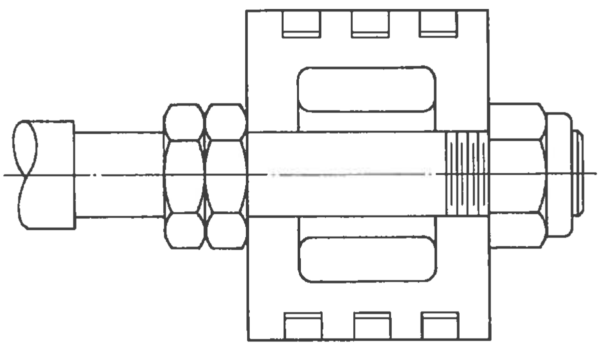


Figure 6.14 — Individual ring piston

6.1.11.9 Stuffing box

A cylindrical cavity through which the plunger or piston rod reciprocates and in which liquid leakage is controlled by means of packing (see Figure 6.16).

A follower ring and throat bushing are used to guide the plunger or rod as it reciprocates. The throat bushing and follower ring contain the packing within the stuffing box.

6.1.11.10 Packing

A material used to provide a seal around the plunger, piston rod, or piston (see Figure 6.17).

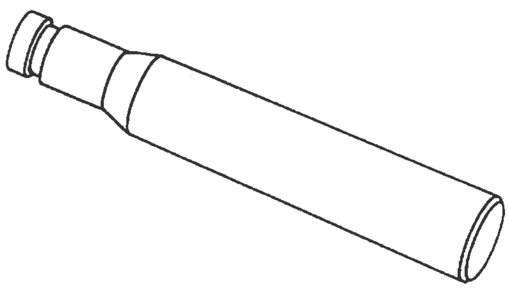


Figure 6.15 — Plunger

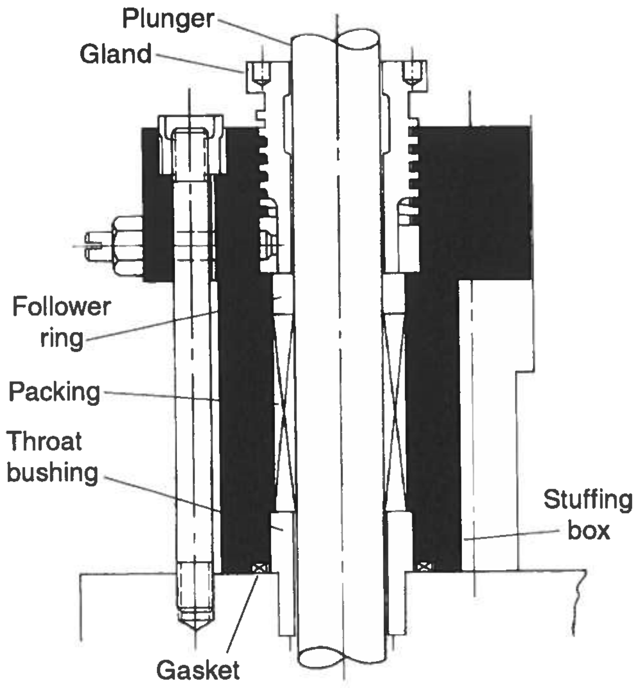


Figure 6.16 — Stuffing box

6.1.11.11 Gland

A part which retains the packing in the stuffing box (see Figure 6.18).

6.1.11.12 Lantern ring (seal cage)

A ring located in the stuffing box to provide space for the introduction of a lubricant or a barrier liquid (see Figure 6.19).

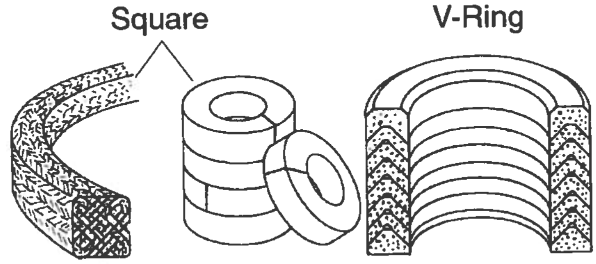


Figure 6.17 — Packing

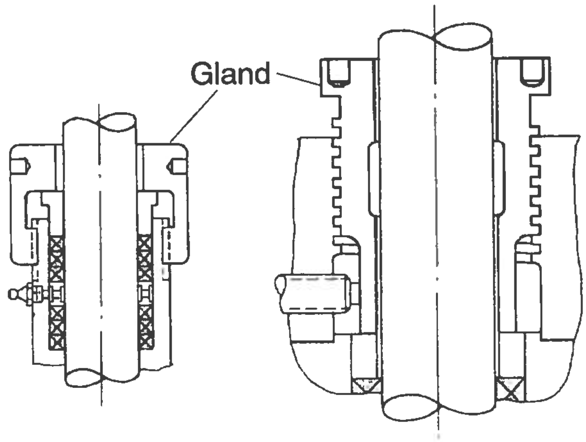


Figure 6.18 — Gland

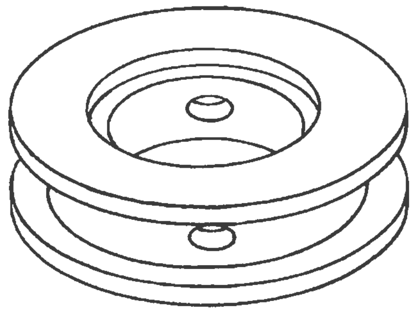


Figure 6.19 — Lantern ring

6.1.11.13 Valve assembly

Usually consists of a seat, valve, spring and spring retainer. It allows liquid to enter and leave each pumping chamber of the cylinder. Each pumping chamber has one or more suction and discharge valve(s) (see Figures 6.20 through 6.23).

6.1.11.14 Upper crosshead

Used in vertical plunger pumps to transmit the reciprocating motion of the side rod to the plunger (see Figure 6.24).

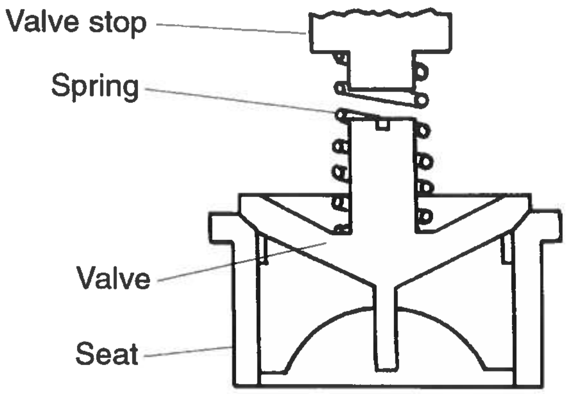


Figure 6.22 — Wing guided valve assembly

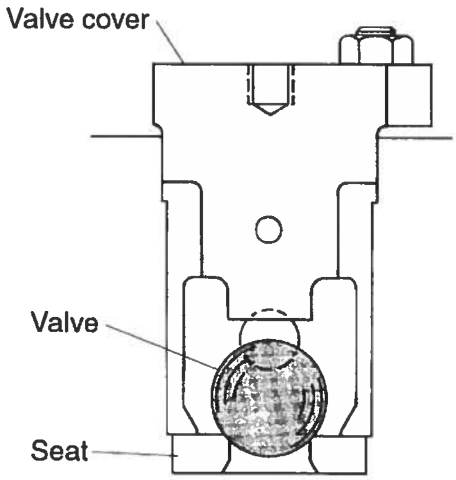


Figure 6.23 — Ball valve assembly

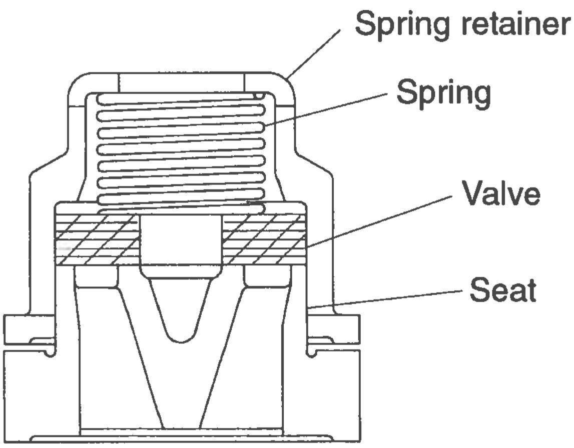


Figure 6.20 — Plate valve assembly

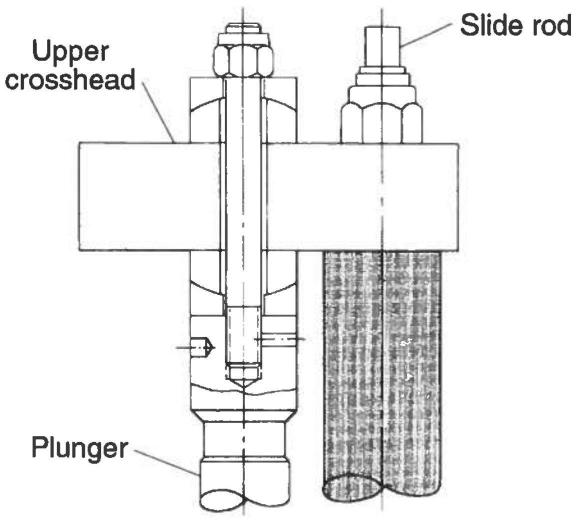


Figure 6.24 — Upper crosshead

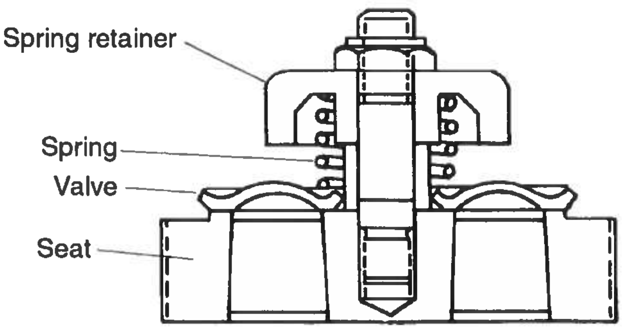
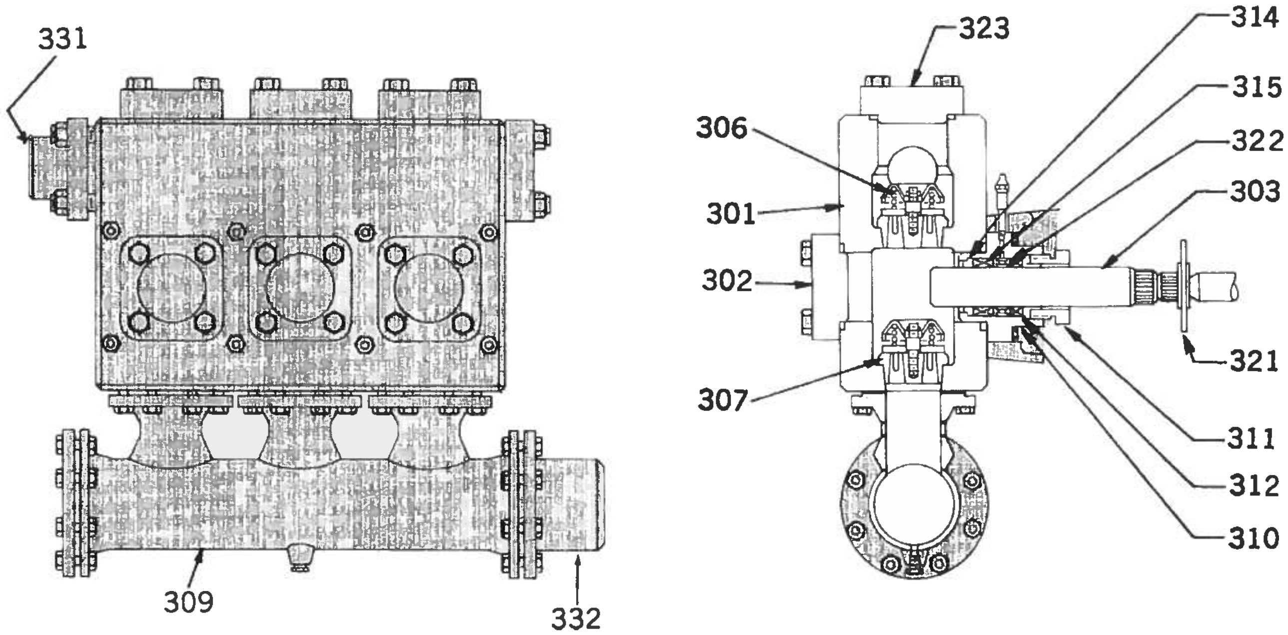
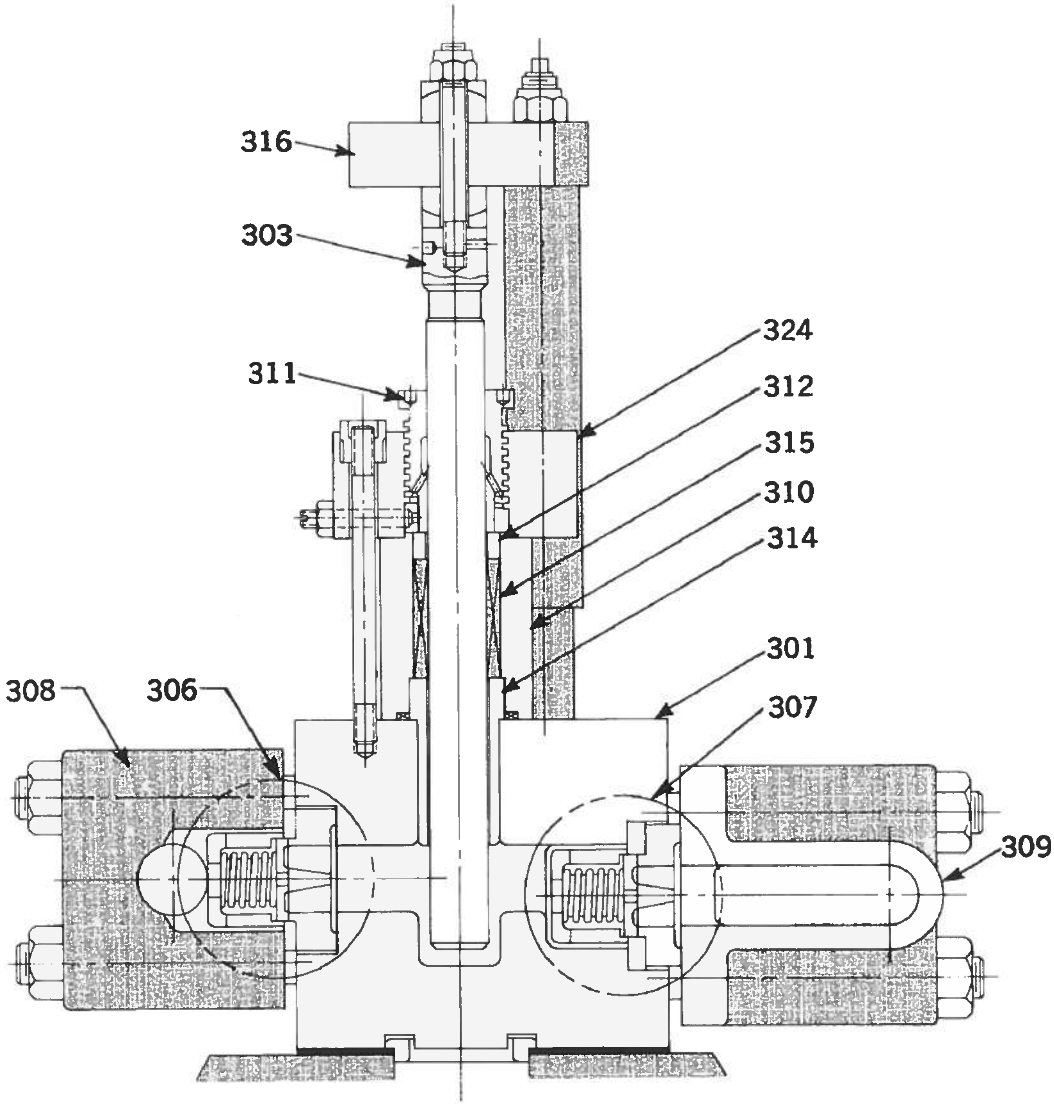


Figure 6.21 — Disc valve assembly



301	Cylinder, liquid	310	Box, liquid stuffing	322	Ring, lantern
302	Head, liquid cylinder	311	Gland, liquid stuffing box	323	Cover, valve
303	Plunger	312	Ring, follower	331	Flange, discharge companion
306	Valve assembly, discharge	314	Bushing, throat	332	Flange, suction companion
307	Valve assembly, suction	315	Packing		
309	Manifold, suction	321	Deflector		

Figure 6.25 — Liquid end, horizontal plunger power pump



301	Cylinder, liquid	309	Manifold, suction	315	Packing
303	Plunger	310	Box, liquid stuffing	316	Crosshead, upper
306	Valve assembly, discharge	311	Gland, liquid stuffing box	324	Ring, gland
307	Valve assembly, suction	312	Ring, follower		
308	Manifold, discharge	314	Bushing, throat		

Figure 6.26 — Liquid end, vertical plunger power pump

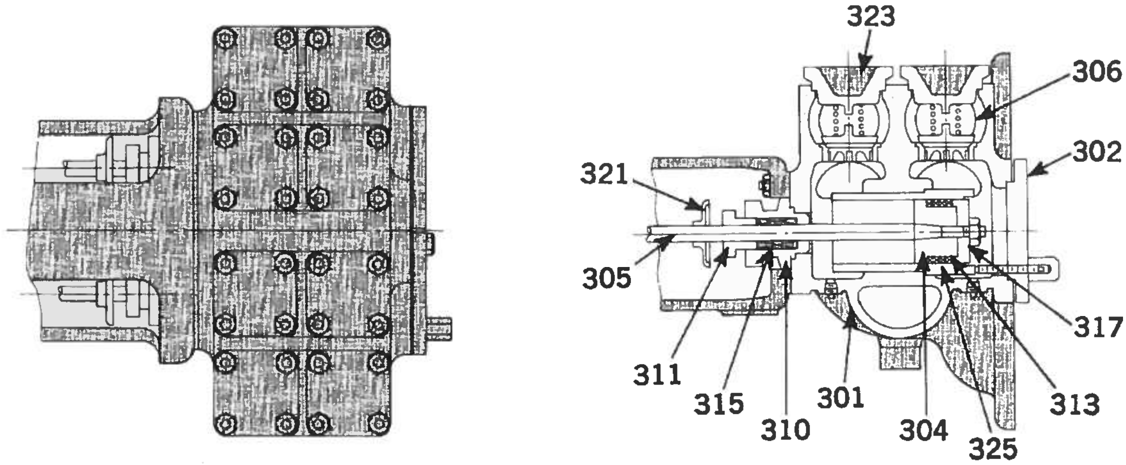


Figure 6.27 — Liquid end, horizontal side pot piston pump

301	Cylinder, liquid	310	Box, liquid stuffing	321	Deflector
302	Head, liquid cylinder	311	Gland, liquid stuffing box	323	Cover, valve
304	Piston, liquid	313	Ring, liquid piston	325	Liner, cylinder
305	Rod, liquid piston	315	Packing		
306	Valve assembly, discharge	317	Follower, piston		

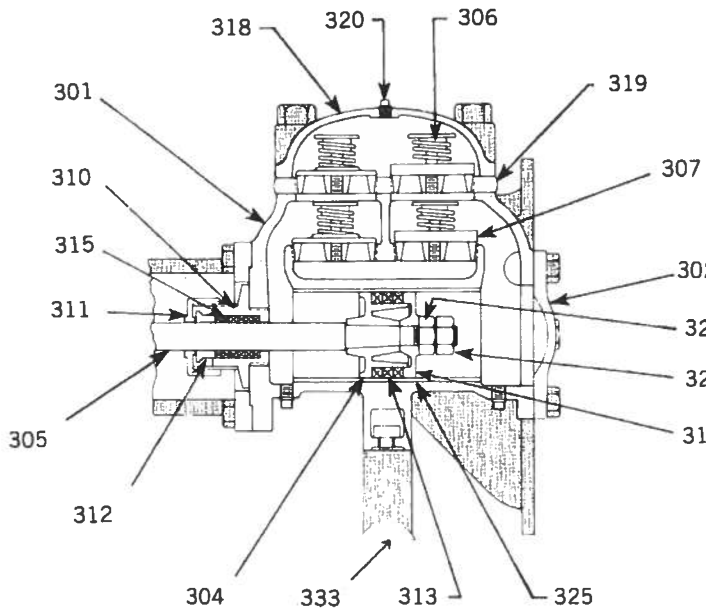


Figure 6.28 — Liquid end, horizontal valve plate piston pump

301	Cylinder, liquid	311	Gland, liquid stuffing box	320	Plug
302	Head, liquid cylinder	312	Ring, follower	325	Liner, cylinder
304	Piston, liquid	313	Ring, liquid piston	326	Nut, piston
305	Rod, liquid piston	315	Packing	327	Nut, piston jam
306	Valve assembly, discharge	317	Follower, piston	333	Foot, liquid cylinder
307	Valve assembly, suction	318	Cover, liquid valve chest		
310	Box, liquid stuffing	319	Plate, valve		

Table 6.1 — List of parts by key name – liquid end parts

Name	Number	Figure in which number appears	Abbreviations
Box, liquid stuffing	310	25, 26, 27, 28	Box liq stfg
Bushing, throat	314	25, 26	Bush thrt
Cover, valve	323	25, 27	Cov val
Cover, liquid valve chest	318	28	Cov liq val chest
Crosshead, upper	316	26	Xhead up
Cylinder, liquid	301	25, 26, 27, 28	Cyl liq
Deflector	321	25, 27	Defl
Flange, discharge companion	331	25	Flg disch comp
Flange, suction companion	332	25	Flg suct comp
Follower, piston	317	27, 28	Fol pist
Foot, liquid cylinder	333	28	Ft liq cyl
Gland, liquid stuffing box	311	25, 26, 27, 28	Gld liq stfg box
Head, liquid cylinder	302	25, 27, 28	Hd liq cyl
Liner, cylinder	325	28	Lnr cyl
Manifold, discharge	308	26	Manf disch
Manifold, suction	309	25, 26	Manf suct
Nut, piston jam	327	28	Nut pist jam
Nut, piston	326	28	Nut pist
Packing	315	25, 26, 27, 28	Pkg
Piston, liquid	304	27, 28	Pist liq
Plate, valve	319	28	Pl val
Plug	320	28	Plug
Plunger	303	25, 26	Plgr
Ringer, follower	312	25, 26, 28	Ring fol
Ring, gland	324	26	Ring gld
Ring, lantern	322	25	Ring lan
Ring, liquid piston	313	27, 28	Ring liq pist
Rod, liquid piston	305	27, 28	Rod liq pist
Valve assembly, discharge	306	25, 26, 27, 28	Val assy disch
Valve assembly, suction	307	25, 26, 28	Val assy suct

6.1.12 Power end

That portion of the pump in which the rotating motion of the crankshaft is converted to a reciprocating motion through connecting rods and crossheads (see Figures 6.39, 6.40, 6.41).

6.1.12.1 Power end parts

The following sections describe major power frame components which are shown in Figures 6.38 through 6.40. A listing by part name is shown in Table 6.2 which also shows the standard abbreviation.

6.1.12.2 Power frame

That portion of the power end which contains the crankshaft, connecting rods, crosshead and bearings used to transmit power and motion to the liquid end. It may consist of one or two pieces (if two, there is one upper and one lower half) (see Figure 6.29).

6.1.12.3 Crankshaft

The stepped shaft which transmits power and motion to the connecting rods. Main bearings and connecting rods are fitted on this member (see Figure 6.30).

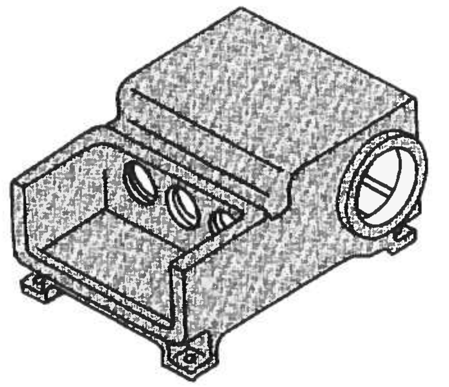


Figure 6.29 — Power frame (one piece)

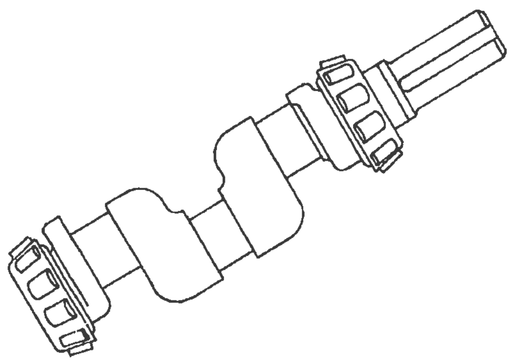


Figure 6.30 — Crankshaft

6.1.12.4 Main bearing

The bearing which supports the crankshaft. Main crankshaft bearings may be sleeve or rolling element type, mounted at each end of the shaft or located elsewhere to provide proper support. These bearings absorb the liquid and inertia loads that are developed by the plunger as it displaces the liquid (see Figures 6.31 and 6.32).

6.1.12.5 Connecting rod

Articulates the motion of the crankshaft to the crosshead. Power is transmitted through compression and/or tension (see Figure 6.33).

6.1.12.6 Crankpin bearing

Transmits the oscillating reciprocating load transmitted by the connecting rod to the crankshaft (see Figure 6.34).

6.1.12.7 Power crosshead

Creates a linear reciprocating motion derived from the crankpin rotary motion through the connecting rod. The reciprocating motion of the crosshead is applied to the plunger or piston via the side rods or crosshead extension (see Figure 6.35).

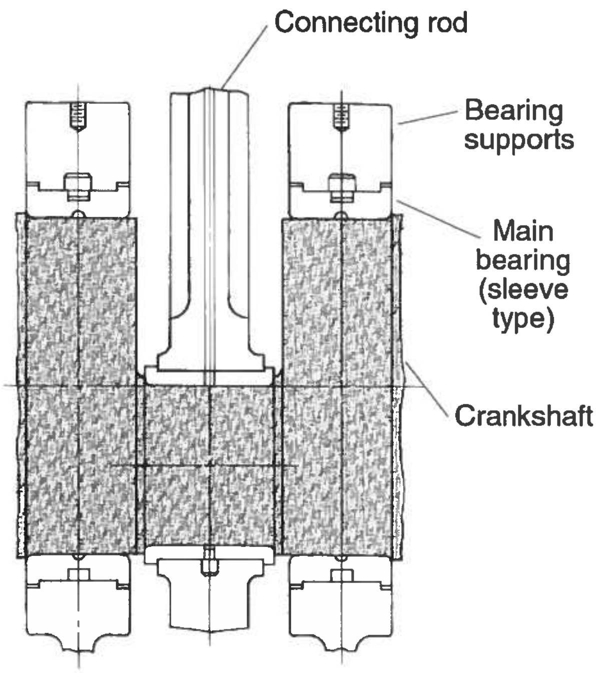


Figure 6.31 — Sleeve bearing

6.1.12.8 Wrist pin

Connects the connecting rod to the crosshead (see Figure 6.36).

6.1.12.9 Wrist pin bearing

Transmits the reciprocating load of the crosshead into the connecting rod (see Figure 6.36).

6.1.12.10 Crosshead extension (plunger extension)

Connects the crosshead to the plunger (see Figure 6.37).

6.1.12.11 Frame extension

Connects the liquid end to the power frame when the liquid end is not bolted directly to the frame. A horizontal extension is sometimes called a cradle (see Figure 6.38).

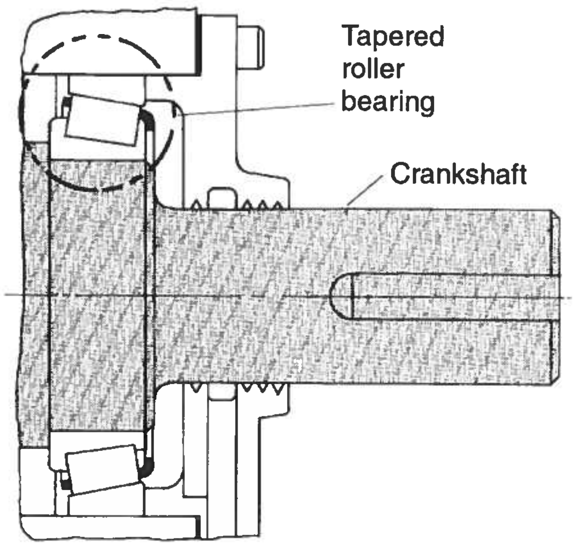


Figure 6.32 — Tapered roller bearing

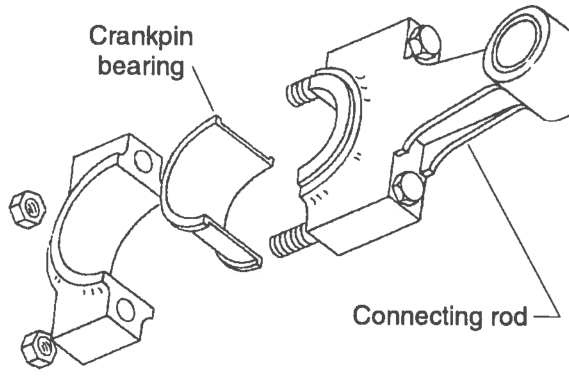


Figure 6.33 — Connecting rod

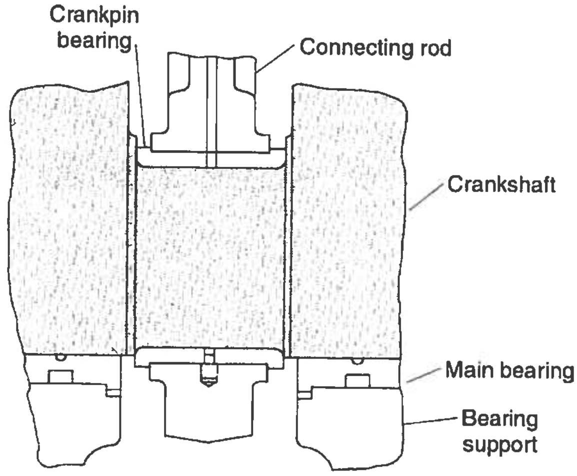


Figure 6.34 — Crankpin bearing

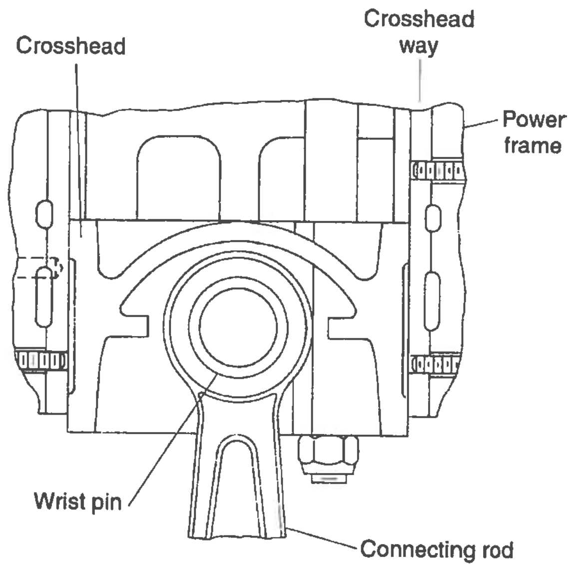


Figure 6.35 — Power crosshead

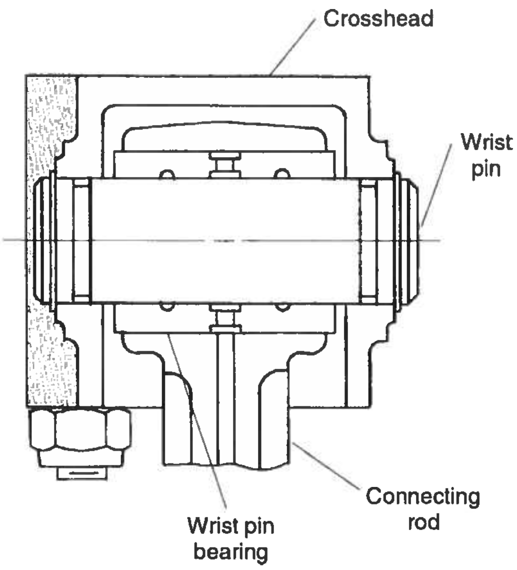


Figure 6.36 — Wrist pin bearing

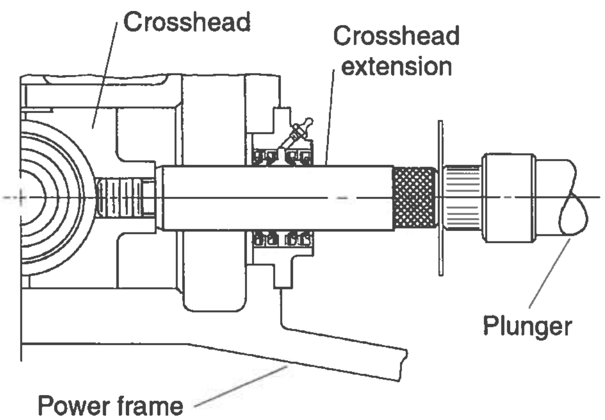


Figure 6.37 — Crosshead extension

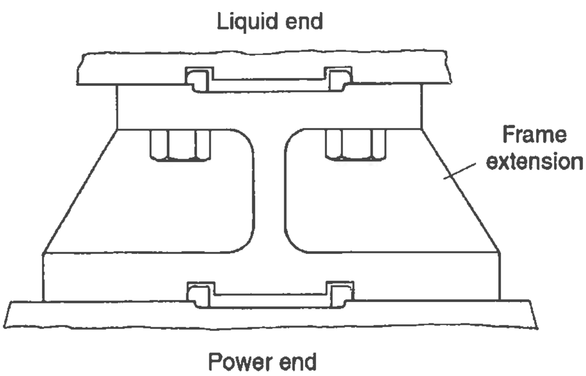
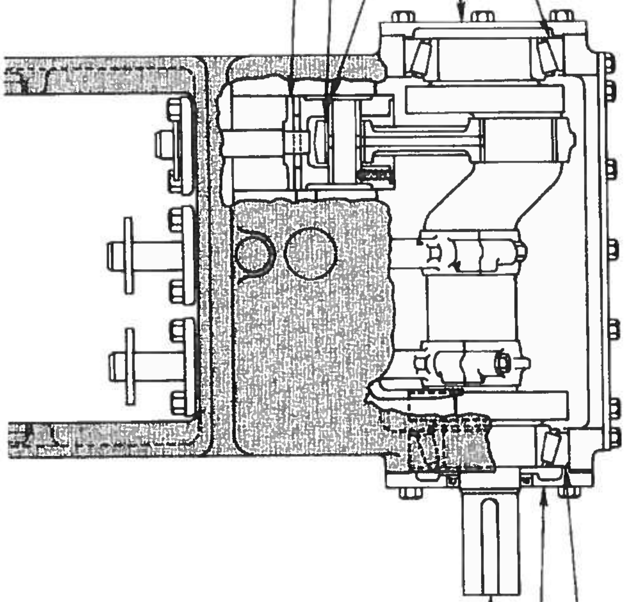
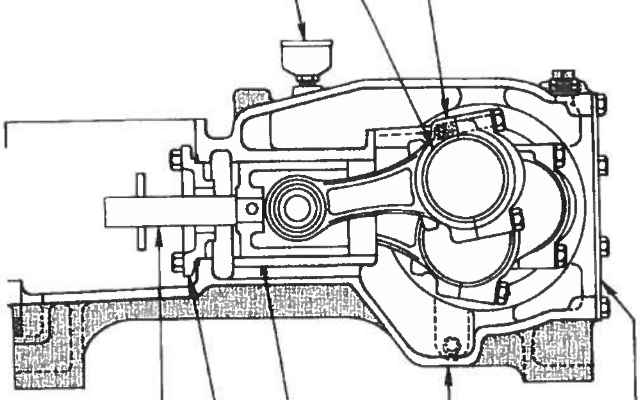


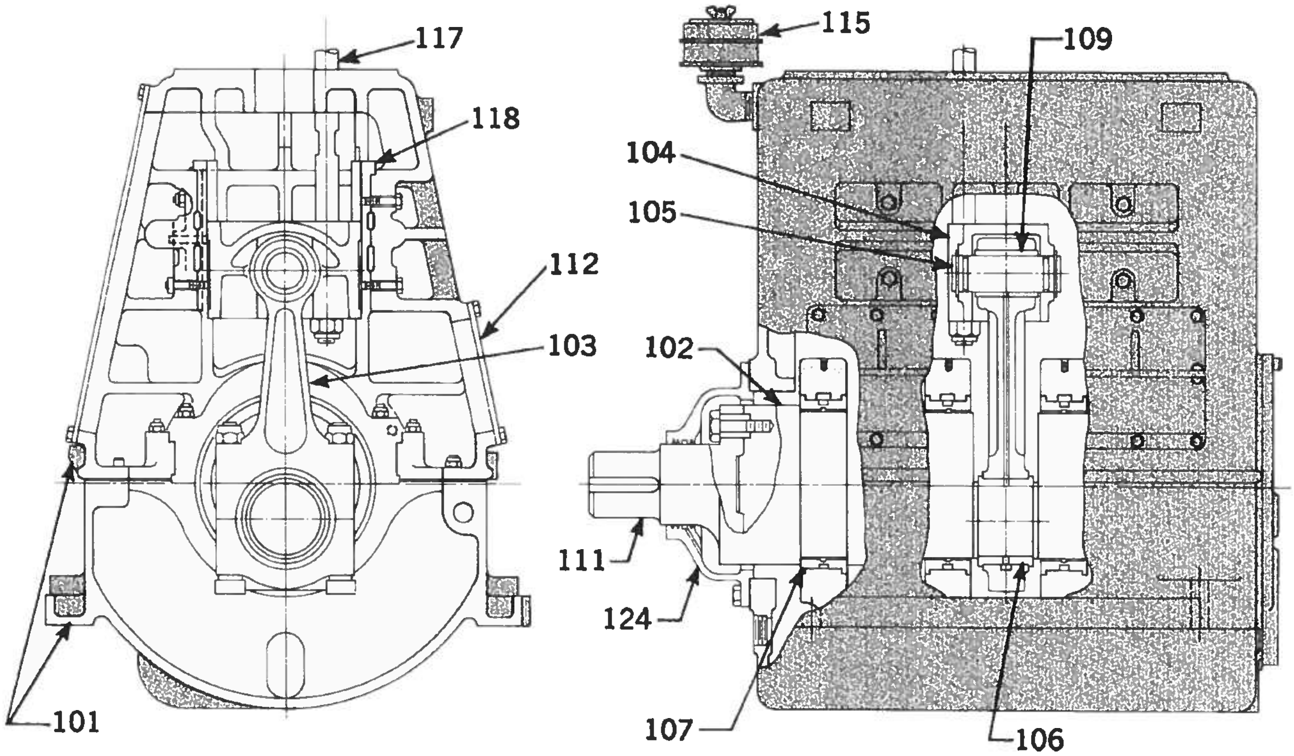
Figure 6.38 — Frame extension





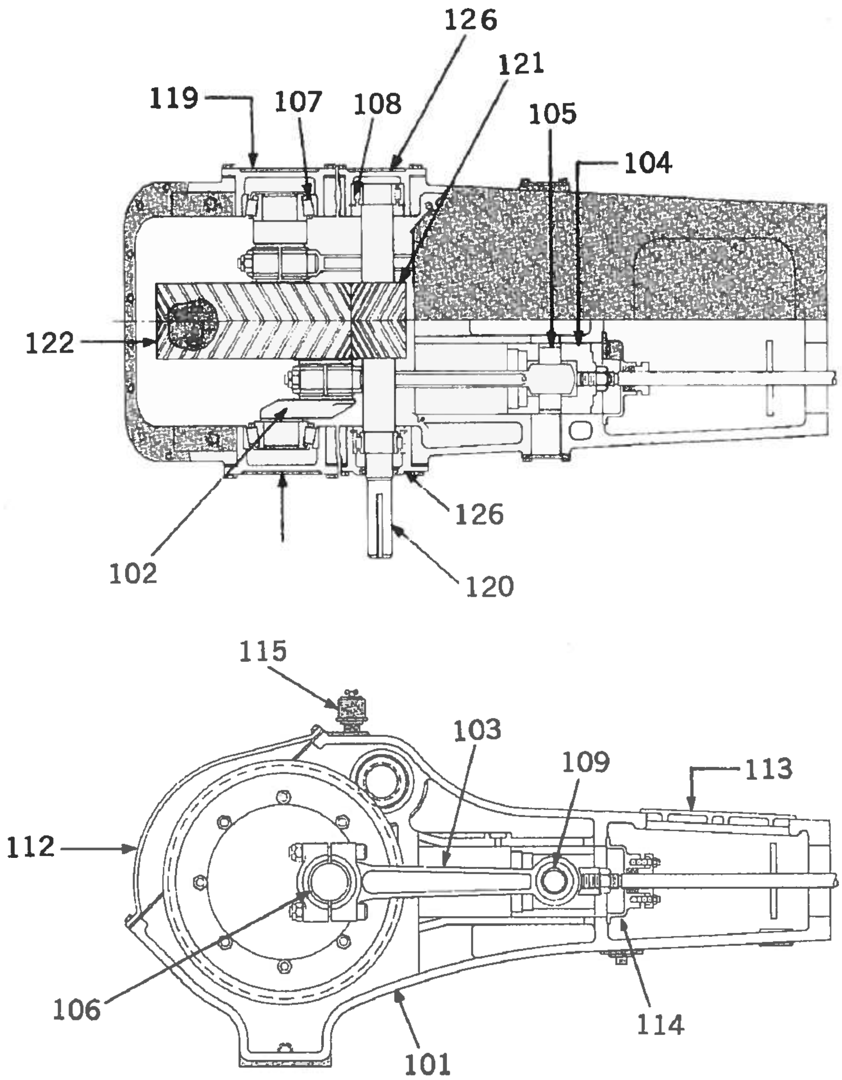
101	Frame, power	106	Bearing, crankpin	114	Box, wiper
102	Crankshaft	107	Bearing, main crankshaft	115	Breather
103	Rod, connecting	109	Bearing, wrist pin	119	Housing, crankshaft bearing
104	Crosshead, power	110	Extension, crosshead		
105	Pin, wrist	112	Cover, crankcase		

Figure 6.39 — Power end, horizontal plunger power pump



101	Frame, power	106	Bearing, crankpin	115	Breather
102	Crankshaft	107	Bearing, main crankshaft	117	Rod, side
103	Rod, connecting	109	Bearing, wrist pin	118	Way, crosshead
104	Crosshead, power	111	Extension, crankshaft	124	Cover, crankshaft extension
105	Pin, wrist	112	Cover, crankcase		

Figure 6.40 — Power end, vertical plunger power pump



101	Frame, power	107	Bearing, main, crankshaft	115	Breather
102	Crankshaft	108	Bearing, pinion shaft	119	Housing, crankshaft bearing
103	Rod, connecting	109	Bearing, wrist pin	120	Pinion shaft
104	Crosshead, power	112	Cover, crankcase	121	Pinion
105	Pin, wrist	113	Cover, cradle	122	Gear
106	Bearing, crankpin	114	Box, wiper	126	Housing, bearing, pinion shaft

Figure 6.41 — Power end, horizontal duplex power pump with integral gears

Table 6.2 — List of parts by key name – power end parts

Name	Number	Figure in which parts appear	Abbreviations
Bearing, Crankpin	106	34, 39, 40, 41	Brg crkpin
Bearing, main crankshaft	107	31, 32, 39, 40, 41	Brg main crksht
Bearing, pinion shaft	108	41	Brg pin sft
Bearing, wrist pin	109	36, 39, 40, 41	Brg wst pin
Box, wiper	114	39, 41	Box wiper
Breather	115	39, 40, 41	Brthr
Cover, crankcase	112	39, 40, 41	Cov crkcase
Cover, crankshaft extension	124	40	Cov crkshft ext
Cover, cradle	113	41	Cov crdl
Crankshaft	102	30, 39, 40, 41	Crkshft
Crosshead, power	104	35, 39, 40, 41	Xhead pwr
Extension, crosshead	110	37, 39	Ext xhead
Extension, crankshaft	111	40	Ext crksft
Extension, frame	123	38	Ext fr
Frame, power	101	39, 40, 41	Fr pwr
Gear	122	41	Gear
Housing, crankshaft bearing	119	39, 41	Hsg crksft brg
Housing, pinion shaft bearing	126	41	Hsg pin sft brg
Pin, wrist	105	39, 40, 41	Pin wst
Pinion	121	41	Pin
Pinion shaft	120	41	Pin sft
Rod, connecting	103	33, 39, 40, 41	Rod conn
Rod, side	117	40	Rod side
Way, crosshead	118	40	Way xhead

6.2 Definitions

The purpose of this section is to define terms used in pump applications. Symbols, terms and units are shown in Table 6.3 and subscripts in Table 6.4.

6.2.1 Flow rate

6.2.1.1 Stroke (L)

One complete unidirectional motion of piston or plunger. Stroke length is expressed in millimeters (inches).

6.2.1.2 Rate of flow (Q)

The quantity of liquid actually delivered per unit of time at suction conditions. It assumes no entrained gases at the stated operating conditions. Rate of flow is expressed in cubic meters per hour (gallons per minute).

6.2.1.3 Speed (n)

The number of revolutions of the crankshaft in a given unit of time. Speed is expressed as revolutions per minute.

6.2.1.4 Displacement (D)

The displacement of a reciprocating pump is the volume swept by all pistons or plungers per unit of time. Deduction for piston rod volume is made on double-acting piston type pumps when calculating displacement.

For single-acting pumps:

(Metric units) $$D = \frac{ALnM}{16.7 \times 10^6}$$

(US units) $$D = \frac{ALnM}{231}$$

For double-acting piston pumps with no tail-rod(s):

(Metric units) $$D = \frac{(2A - a)LnM}{16.7 \times 10^6}$$

(US units) $$D = \frac{(2A - a)LnM}{231}$$

Where:

A = Plunger or piston area;

a = Piston rod cross-sectional area;

L = Stroke length;

n = rpm of crankshaft;

M = Number of pistons or plungers.

6.2.1.5 Slip (S)

Slip of a reciprocating pump is the loss of rate of flow, expressed as a fraction or percent of displacement, due to leaks past the valves (including the backflow through the valves caused by delayed closing) and past double-acting pistons. Slip does not include liquid compressibility or leaks from the liquid end.

6.2.1.6 Plunger or piston speed (v)

The plunger or piston speed is the average speed of the plunger or piston. It is expressed in meters per second (feet per second):

(Metric units) $$v = \frac{nL}{30{,}000}$$ meters per second

(US units) $$v = \frac{nL}{360}$$ feet per second

6.2.2 Pressures

Pressure is the expression of the energy content of the liquid in units of force per unit area. Pressure is expressed in kiloPascal (pounds per square inch).

6.2.2.1 Total discharge pressure (p_d)

The total discharge pressure is the algebraic sum of the discharge gauge pressure, velocity pressure and elevation pressure measured on the discharge side of the pump:

(Metric units) $$p_d = p_{gd} + \frac{[v_d^2/2g + Z_d]s}{.102} kPa$$

(US units) $$p_d = p_{gd} + \frac{[v_d^2/2g + Z_d]s}{2.31} psi$$

6.2.2.2 Total suction pressure (p_s)

The total suction pressure is the algebraic sum of the suction gauge pressure, velocity pressure and elevation pressure measured on the suction side of the pump:

Table 6.3 — Symbols

Symbol	Term	Metric unit	Abbreviation	US Customary Unit	Abbreviation	Conversion factor[a]
A	Area	square millimeter	mm^2	square inches	in^2	645.2
a	Area of piston rod	square millimeter	mm^2	square inches	in^2	645.2
β (beta)	Meter or orifice ratio	dimensionless	—	dimensionless	—	1
C	Coefficient for acceleration head	dimensionless	—	dimensionless	—	1
D	Displacement	cubic meter per hour	m^3/h	US gallons per minute	gpm	0.2271
d	Diameter	millimeter	mm	inches	in	25.4
Δ (delta)	Difference	dimensionless	—	dimensionless	—	1
η (eta)	Efficiency	percent	%	percent	%	1
g	Gravitational acceleration	meter/second squared	m/s^2	feet/second squared	ft/sec^2	0.3048
γ (gamma)	Specific weight			pounds/cubic foot	lb/ft^3	
h	Head	meter	m	feet	ft	0.3048
L	Stroke length	millimeter	mm	inches	in	25.4
M	Number of pistons	dimensionless	—	dimensionless	—	1
n	Speed	revolutions/minute	rpm	revolutions/minute	rpm	1
NPSHA	Net positive suction head avail.	kilopascal	kPa	pounds/square inch	psi	6.895
NPSHR	Net positive suction head required	kilopascal	kPa	pounds/square inch	psi	6.895
ν (nu)	Kinematic viscosity	millimeter squared/sec	mm^2/s	seconds Saybolt Universal	SSU	0.22
π	pi = 3.1416	dimensionless	—	dimensionless	—	1
p	Pressure	kilopascal	kPa	pounds/square inch	psi	6.895
P	Power	kilowatt	kW	horsepower	hp	0.7457
q	Rate of flow (capacity)	cubic meter/hour	m^3/h	cubic feet/second	ft^3/sec	101.94
Q	Rate of flow (capacity)	cubic meter/hour	m^3/h	US gallons/minute	gpm	0.2271
ρ (rho)	Density	kilogram/cubic meter	kg/m^3	pound mass/cubic foot	lbm/ft^3	16.02
s	Specific gravity	dimensionless	—	dimensionless	—	1
S	Slip	percent	%	percent	%	1
t	Temperature	degrees Celsius	°C	degrees Fahrenheit	°F	$(°F-32) \times {}^5/_9$
τ (tau)	Torque	Newton – meter	N·m	pound-feet	lb-ft	1.356
v	Velocity	meter/second	m/s	feet/second	ft/sec	0.3048
V	Specific volume	cubic meters/kiloNewton	m^3/kN	cubic feet/pound	ft^3/kN	6.365
x	Exponent	none	none	none	none	1
Z	Elevation gauge distance above or below datum	meter	m	feet	ft	0.3048

[a] Conversion factor × US units = metric units.

(Metric units) $p_s = p_{gs} + \frac{[v_s^2/2g + Z_s]s}{.102} kPa$

(US units) $p_s = p_{gs} + \frac{[v_s^2/2g + Z_s]s}{2.31} psi$

The velocity pressure, $v^2/2g$, is computed for the liquid velocity at the point of gauge attachment.

The elevation head, Z, is referred to the datum and is positive when above datum and negative when below datum.

6.2.2.3 Total differential pressure (p_H)

The measure of the pressure increase imparted to the liquid by the pump is therefore the difference between the total discharge pressure and the total suction pressure:

$$p_H = p_d - p_s$$

6.2.2.4 Gauge pressure (p_g)

The pressure energy of the liquid determined by a pressure gauge, or other measuring device, relative to the atmosphere.

6.2.2.5 Elevation pressure (p_z)

The potential energy of the liquid due to elevation of the gauge or liquid level above or below the datum, expressed as equivalent pressure.

6.2.2.6 Elevation head (Z)

The vertical distance from the centerline of a pressure gauge or liquid level to the datum.

6.2.2.7 Velocity pressure (p_v)

Velocity pressure is the hydraulic pressure needed to move fluid from rest to the average velocity.

6.2.2.8 Barometric pressure (p_b)

The absolute pressure of the atmosphere at the pumping site. At sea level, the value of p_b is taken at 101

Table 6.4 — Subscripts

Subscript	Term	Subscript	Term
a	Absolute	d	Discharge
acc	Acceleration	dvr	Driver input
b	Barometric	p	Pump
c	Piston or plunger	r	Tail rod
f	Friction loss	s	Suction
g	Gauge	t	Theoretical
H	Total head	Δ (delta)	Differential
i	Inlet	v	Velocity
max	Maximum	V	Volume
min	Minimum	vp	Vapor pressure
mot	Motor	w	Liquid or water
ni	Net Inlet	x	Exponent
o	Outlet	1	Test condition
OA	Overall	2	Specific condition

kPa (14.7 psia) and declines with increasing altitude, as shown in Table 6.5.

6.2.2.9 Friction loss pressure (p_f)

The loss of pressure energy in a liquid due to friction as it flows through pipe and fittings.

6.2.3 Power (P)

The work requirement per unit of time. Power is expressed in kilowatt (horsepower).

6.2.3.1 Pump input power (P_p)

The mechanical power delivered to a pump input shaft, at the specified operating conditions.

Table 6.5 — Barometric pressure versus elevation

Altitude		Average barometric pressure	
meters	feet[a]	kPa	psia
0	0	101	14.7
150	500	99	14.4
300	1000	98	14.2
450	1500	96	13.9
600	2000	94	13.7
750	2500	92	13.4
900	3000	91	13.2
1200	4000	88	12.7
1500	5000	84	12.2
1800	6000	81	11.8
2100	7000	78	11.3
2400	8000	75	10.9
2700	9000	72	10.5
3000	10000	70	10.1
3700	12000	64	9.3
4300	14000	59	8.6
4900	16000	55	8.0

6.2.3.2 Pump output power (P_w)

The power imparted to the liquid by the pump. It is also called water horsepower or liquid horsepower:

$$\text{(Metric units)}\quad P_w = \frac{Q \times p_H}{3600}$$

$$\text{(US units)}\quad P_w = \frac{Q \times p_H}{1714}$$

6.2.3.3 Pump torque

The turning force or moment that causes rotation of the pump shaft.

6.2.4 Efficiencies (η)

6.2.4.1 Pump efficiency (η_p) (also called pump mechanical efficiency)

The ratio of the pump power output to the pump power input expressed as a percent:

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

6.2.4.2 Volumetric efficiency (η_v)

The ratio of the pump rate of flow to displacement expressed as a percent:

$$\eta_v = \frac{Q}{D} \times 100$$

6.2.5 Pistons, plungers and valves

6.2.5.1 Plunger load (single-acting pump)

The computed axial hydraulic load, acting upon one plunger during the discharge portion of the stroke, is the plunger load. It is the product of plunger area and the gauge discharge pressure. It is expressed in Newtons (pounds force).

6.2.5.2 Piston rod load (double-acting pump)

The computed axial hydraulic load, acting upon one piston rod during the forward stroke (toward head end) is the piston rod load. It is the product of piston area and discharge pressure, less the product of net piston area (rod area deducted) and suction pressure. It is expressed in Newtons (pounds force).

6.2.5.3 Valve seat area

The term valve seat area shall be defined as the minimum net clear area through the valve seat with proper deductions for wings, hubs, grids, etc. It is expressed in square millimeters (square inches).

In Figures 6.42 and 6.43, the seat area = A minus B minus C; in Figure 6.44, the seat area = A minus D; in Figure 6.45, the seat area = A.

A = Area of smallest inside diameter of valve seat;

B = Area of center boss in valve seat;

C = Total area of radial ribs in valve seat;

D = Total area of wing guides and any other part of valve projecting into valve seat when valve is open.

6.2.6 Suction conditions

6.2.6.1 Submerged suction

A submerged suction exists when the centerline of the pump inlet port is below the level of the liquid in the supply tank. However, the absolute pressure of the liquid entering the centerline of the pump inlet port may be below atmospheric pressure when the pump is operating at the specified speed. This occurs whenever friction head exceeds the static suction head (submergence) of the pump.

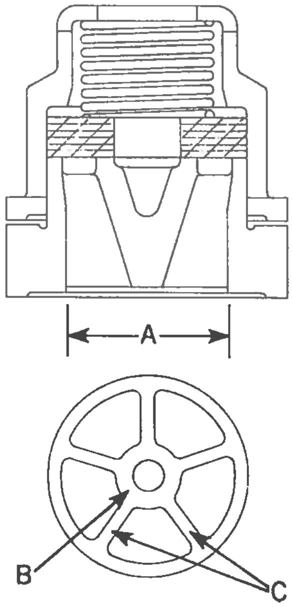


Figure 6.43 — Plate valve

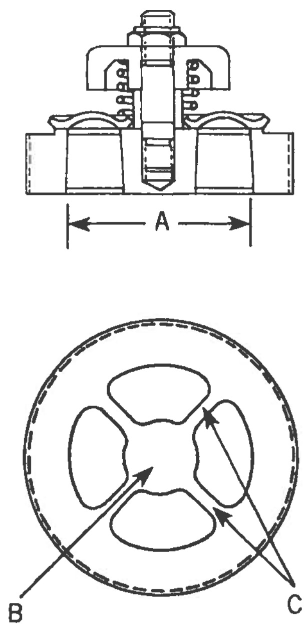


Figure 6.42 — Disc valve

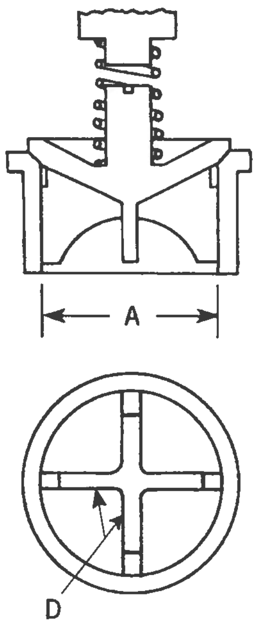


Figure 6.44 — Wing guided valve

6.2.6.2 Flooded suction

Flooded suction implies that the liquid flows from an atmospheric source to the pump without the average pressure at the intake port of the pump dropping below atmospheric pressure with the pump operating at specified rate of flow.

Thus, the static suction head must always exceed friction head in a flooded suction installation.

6.2.6.3 Static suction lift (l_s)

Static suction lift is a hydraulic pressure below atmospheric at the intake port of the pump with the liquid at rest. It is often measured in millimeters (inches) of mercury vacuum. Suction lift may be thought of as "negative" elevation pressure.

6.2.6.4 Net positive suction head (NPSH)

6.2.6.4.1 Net positive suction head available (NPSHA) (Net positive inlet pressure available [NPIPA])

Net positive suction head available is the total suction pressure available from the system at the pump suction condition, minus the vapor pressure of the liquid at pumping temperature, acceleration head loss, friction losses, and pressure pulsations due to acoustical resonances. NPSHA for a reciprocating pump is normally expressed in Newton per square millimeter (pounds per square inch).

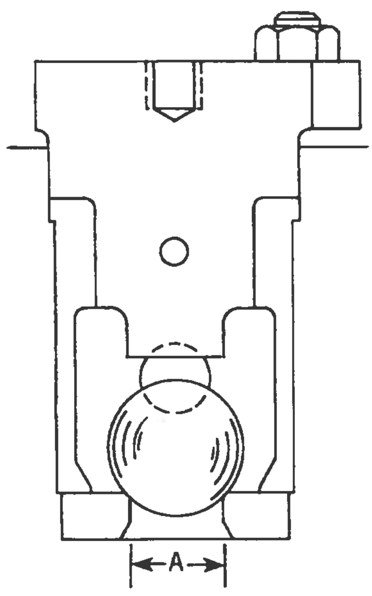


Figure 6.45 — Ball valve

6.2.6.4.2 Net positive suction head required (NPSHR) (Net positive inlet pressure required [NPIPR])

The amount of suction pressure, over vapor pressure, required by the pump to obtain satisfactory volumetric efficiency. This is usually when there is no more than 3% reduction in rate of flow.

The pump manufacturer determines by test the net positive suction head required by the pump at the specified operating conditions.

NPSHR is related to losses in the suction valves of the pump and frictional losses in the pump suction manifold and pumping chambers. NPSHR does not include system acceleration head, which is a system-related factor and can be a significant problem (see Section 6.2.6.6).

6.2.6.5 Total suction lift

Total suction lift is the difference between the absolute operating inlet pressure at the pump inlet port centerline and atmospheric pressure. It is also the sum of suction system frictional losses and the static suction lift.

6.2.6.6 Acceleration head (h_{acc}) acceleration pressure (p_{acc})

Total suction lift, as defined in the preceding sections, represents the average, without reference to the fluctuation above and below this average due to the inertia effect of the liquid mass in the suction line. This pressure fluctuation or acceleration pressure must be taken into account if the pump is to fill properly without separation and pounding or vibration of the suction line.

With the slider-crank driver of a power pump, maximum plunger acceleration occurs at the start or the end of each individual stroke, and this is reflected in a similar discontinuity in the cyclical pattern of the combined flow curve corresponding to each piston or plunger and crank arrangement. The pressure required to accelerate the liquid column is a function of the length of the suction line, the average velocity in this line, the rotative speed, the type of pump, and the relative elasticity of the liquid and the pipe and may be calculated as follows:

(Metric units) $h_{acc} = \frac{lvnC}{Kg}$ or $p_{acc} = \frac{lvnC\rho}{1000K}$

$$\text{(US units)}\ h_{acc} = \frac{lvnC}{Kg} \text{ or } p_{acc} = \frac{lvnCs}{231Kg}$$

Where:

h_{acc} = Acceleration head in meters (feet);

p_{acc} = Acceleration pressure in kPa (psi);

l = Length of suction line in meters (feet);

v = Velocity in suction line in m/s (fps);

n = Pump speed in rpm;

C = Coefficient as follows:

	Single-acting	Double-acting
Simplex	0.628	0.200
Duplex	0.200	0.115
Triplex	0.066	0.066
Quintuplex	0.040	0.040
Septuplex	0.028	0.028
Nonuplex	0.022	0.022

ρ = Liquid density — kg/m^3

s = Specific gravity of liquid

K = A factor representing the relative compressibility of the liquid;

(K = 1.4 for hot water, K = 2.5 for hot oil);

g = Gravitational constant, 9.81 m/s^2 (32.3 ft/sec^2).

NOTE: This calculation provides a conservative estimate of acceleration head losses in piping lengths up to 15 meters (50 feet).

A pulsation dampener properly installed near the pump with a short, full-size connection to the pump or suction pipe can absorb the cyclical flow variation and reduce the low frequency (0-35 HZ) pressure fluctuation in the suction pipe to that corresponding to a length of 5 to 15 pipe diameters, if kept properly charged.

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

Pressure fluctuations due to higher frequency acoustic resonances may be reduced by the addition of all-liquid filters or orifice plates. Correct sizing and location of such devices can only be accomplished as part of a pump/system acoustic analysis.

EXAMPLE (Metric): Given a 50 mm × 125 mm triplex pump running at 360 rpm with a capacity of 16 m^3/h of water with a suction pipe made up of 1 meter of 100-mm pipe and 6 meters of 150-mm pipe, determine the acceleration head and pressure.

Average velocity in 100-mm pipe:

$$v_4 = \frac{\frac{16}{3600}}{.785 \times 0.100^2} = 0.566 \text{ m/s};$$

Average velocity in 150-mm pipe:

$$v_6 = \frac{\frac{16}{3600}}{.785 \times 0.150^2} = 0.252 \text{ m/s};$$

Acceleration head in 100-mm pipe:

$$h_{acc_4} = \frac{1 \times 0.566 \times 360 \times 0.066}{1.4 \times 9.81} = 0.98 \text{ m};$$

Acceleration head in 150-mm pipe:

$$h_{acc_6} = \frac{6 \times 0.252 \times 360 \times 0.066}{1.4 \times 9.81} = 2.62 \text{ m};$$

Total acceleration head:

$$h_{acc} = 0.98 + 2.62 = 3.60 \text{ m}$$

$$p_{acc} = \frac{3.60 \times 1000 \times 9.81}{1000} = 35.3 \text{ kPa}$$

EXAMPLE (US units): Given a 2 inch × 5 inch triplex pump running at 360 rpm with a capacity of 73 gpm of water with a suction pipe made up of 4 feet of 4-inch pipe and 20 feet of 6-inch pipe, determine the acceleration head and pressure.

Average velocity in 4-inch pipe:

$$v_4 = \frac{.321 \times 73}{12.73} = 1.84\ fps;$$

Average velocity in 6-inch pipe:

$$v_6 = \frac{.321 \times 73}{28.89} = 0.811 fps;$$

Acceleration head in 4-inch pipe:

$$h_{acc_4} = \frac{4 \times 1.84 \times 360 \times .066}{1.4 \times 32.2} = 3.88\ ft.;$$

Acceleration head in 6-inch pipe:

$$h_{acc_6} = \frac{20 \times .811 \times 360 \times .066}{1.4 \times 32.2} = 8.55\ ft.;$$

Total acceleration head:

$$h_{acc} = 3.88 + 8.55 = 12.43\ ft.$$

$$p_{acc} = \frac{h_{acc}s}{2.31} = \frac{12.43 \times 1.0}{2.31} = 5.38\ psi$$

6.2.7 Slurry

A mixture consisting of solid particles dispersed in a liquid.

6.2.7.1 Apparent viscosity

The viscosity of a non-Newtonian slurry at a particular rate of shear, expressed in terms applicable to Newtonian fluids.

6.2.7.2 Critical carrying velocity

The mean velocity of the specific slurry in a particular conduit, above which the solids phase remains in suspension and below which solid-liquid separation occurs.

6.2.7.3 Effective particle diameter

The single size used to represent the behavior of a mixture of various sizes of particles in a slurry. This designation is used by some engineers to calculate system requirements and pump performance.

6.2.7.4 Friction characteristic

A term used to describe the resistance to flow which is exhibited by solid-liquid mixtures at various rates of flow.

6.2.7.5 Heterogeneous mixture

A mixture of solids and a liquid in which the solids are not uniformly distributed.

6.2.7.6 Homogeneous flow (fully suspended solids)

A type of slurry flow in which the solids are thoroughly mixed in the flowing stream and a negligible amount of the solids are sliding along the conduit wall.

6.2.7.7 Homogeneous mixture

A mixture of solids and a liquid in which the solids are uniformly distributed.

6.2.7.8 Non-homogeneous flow (partially suspended solids)

A type of slurry flow in which the solids are stratified, with a portion of the solids sliding along the conduit wall. Sometimes called "heterogeneous flow" or "flow with partially suspended solids."

6.2.7.9 Non-settling slurry

A slurry in which the solids do not settle to the bottom of the containing vessel or conduit but remain in suspension, without agitation, for long periods of time.

6.2.7.10 Percent solids by volume

The actual volume of the solid material in a given volume of slurry, divided by the given volume of slurry, multiplied by 100.

6.2.7.11 Percent solids by weight

The weight of dry solids in a given volume of slurry, divided by the total weight of that volume of slurry, multiplied by 100.

6.2.7.12 Saltation

A condition which exists in a moving stream of slurry when solids settle in the bottom of the stream in random agglomerations which build up and wash away with irregular frequency.

6.2.7.13 Settling slurry

A slurry in which the solids move to the bottom of the containing vessel or conduit at a discernible rate but which remain in suspension if the slurry is agitated constantly.

6.2.7.14 Settling velocity

The rate at which the solids in a slurry move to the bottom of a container of liquid that is not in motion. (Not to be confused with the velocity of a slurry that is less than the critical carrying velocity as defined above.)

6.3 Design and application

The purpose of this section is to provide general guidelines for the application of reciprocating power pumps.

6.3.1 Typical services

- Absorption oil charge;
- Amine charge;
- Ammonia injection;
- Boiler feed;
- Carbamate;
- Caustic injection;
- Glycol injection;
- High-pressure water cleaning and cutting;
- Homogenizing (foods, chemicals, fuels);
- Hydraulic systems in steel and aluminum mills;
- Hydrostatic test;
- Pipeline (hydrocarbons, ammonia);
- Reactor charge (nuclear power plant);
- Rerun;
- Reverse osmosis;
- Salt water disposal;
- Secondary recovery (oil field production);
- Slurry (ores, coal, soap, drilling mud) — See discussion in Section 6.3.10;
- Spray drying;
- Standby liquid control (nuclear power plant);
- Transfer;
- Wash-water injection;
- Waste disposal.

6.3.2 Basic speeds

Conditions of installation and variations in design have significant influence in the selection of speed. The values which follow are intended to serve as guidelines for basic speed ratings based on pumping cold water.

Single-acting plunger-type power pumps

Stroke length		Basic speed (rpm)
mm[a]	inch[a]	
50	2	750
75	3	530
100	4	420
125	5	360
150	6	315
175	7	290
200	8	262

[a] Values rounded for convenience.

Double-acting piston-type power pumps

Stroke length		Basic speed (rpm)
mm[a]	inch[a]	
50	2	140
100	4	116
150	6	100
200	8	90
250	10	83
300	12	78
350	14	74
400	16	70

[a] Values rounded for convenience.

For an intermediate stroke length, speed may be interpolated.

It should be noted that these speeds are intended only as reference points. Some manufacturers offer their pumps for operation at or above these basic speeds. Others recommend lower speeds.

When a pump originally designed for low viscosity liquids is used for liquids of higher viscosity, basic pump speed reduction is necessary to obtain proper valve dynamics and prevent liquid separation. The following procedure should be followed to reduce basic pump speeds when viscosity ranges from 65 to 6500 mm^2/s (300 to 30,000 SSU). Only pumps specifically designed for high viscosity service should be used for liquids with viscosities above 6500 mm^2/s (30,000 SSU).

1) Using standard selection criteria, select a pump for the required flow rate and pressure, disregarding viscosity. This determines suction valve size (area) and required operating speed;

2) Determine liquid viscosity at pump temperature;

3) Calculate the average suction valve liquid velocity for required flow rate:

(Metric units) $v = \dfrac{556Q}{MA}$

(US units) $v = \dfrac{.642Q}{MA}$

Where:

v = Average valve liquid velocity, m/sec (ft/sec.);

Q = Flow rate, m^3/hr (gpm);

M = Number of suction valves;

A = Suction valve flow area, (each) mm^2 ($in.^2$);

4) Using Figure 6.46, determine the percent of basic speed reduction required for given liquid viscosity and suction valve liquid velocity. Always select the higher liquid velocity curve when actual velocity falls between the two curves;

5) Multiply basic speed for pump type selected in Step 1 by the percent reduction determined in Step 4;

6) If the reduced basic speed is above the operating speed for unit selected in Step 1, selection is satisfactory;

7) If reduced basic speed is below the operating speed for unit selected in Step 1, the selection is invalid for high viscosity liquid, and a larger pump size should be selected. Repeat Steps 2 through 5 for new selection;

Once proper selection is found, the following items should be reviewed to assure reasonable pump performance for high viscosity applications:

8) Check power end lubrication with manufacturer when running at reduced speeds;

9) Ensure NPSHA is greater than NPSHR.

EXAMPLE (Metric): High viscosity pump selection

Given: $Q = 23\ m^3/h$, $p_d = 14{,}000$ kPa, $p_s = 500$ kPa, Viscosity = 2000 mm^2/s

1) Select pump based on Q and p_d, disregarding viscosity;

Selection 50 × 125 × Triplex n = 327 rpm @ 96% VE;

2) Suction valve area A = 2400 mm^2 (obtained from manufacturer)

$$v = \frac{556(23)}{(3)(2400)} = 1.78 \text{ m/s}$$

3) From Figure 6.45A, percent of basic speed for 2000 mm^2/s is 85%;

4) Viscosity corrected basic speed is .85 (360) = 306 rpm

Basic speed for 125-mm stroke pump is 360 rpm from table above;

5) Since viscosity corrected basic speed is less than required operating speed, a new selection is required. Return to Step 1;

i) New selection 65 × 125 quintuplex n = 178 rpm @ 96% VE;

ii) Suction valve area A – 2400 mm^2 (from manufacturer);

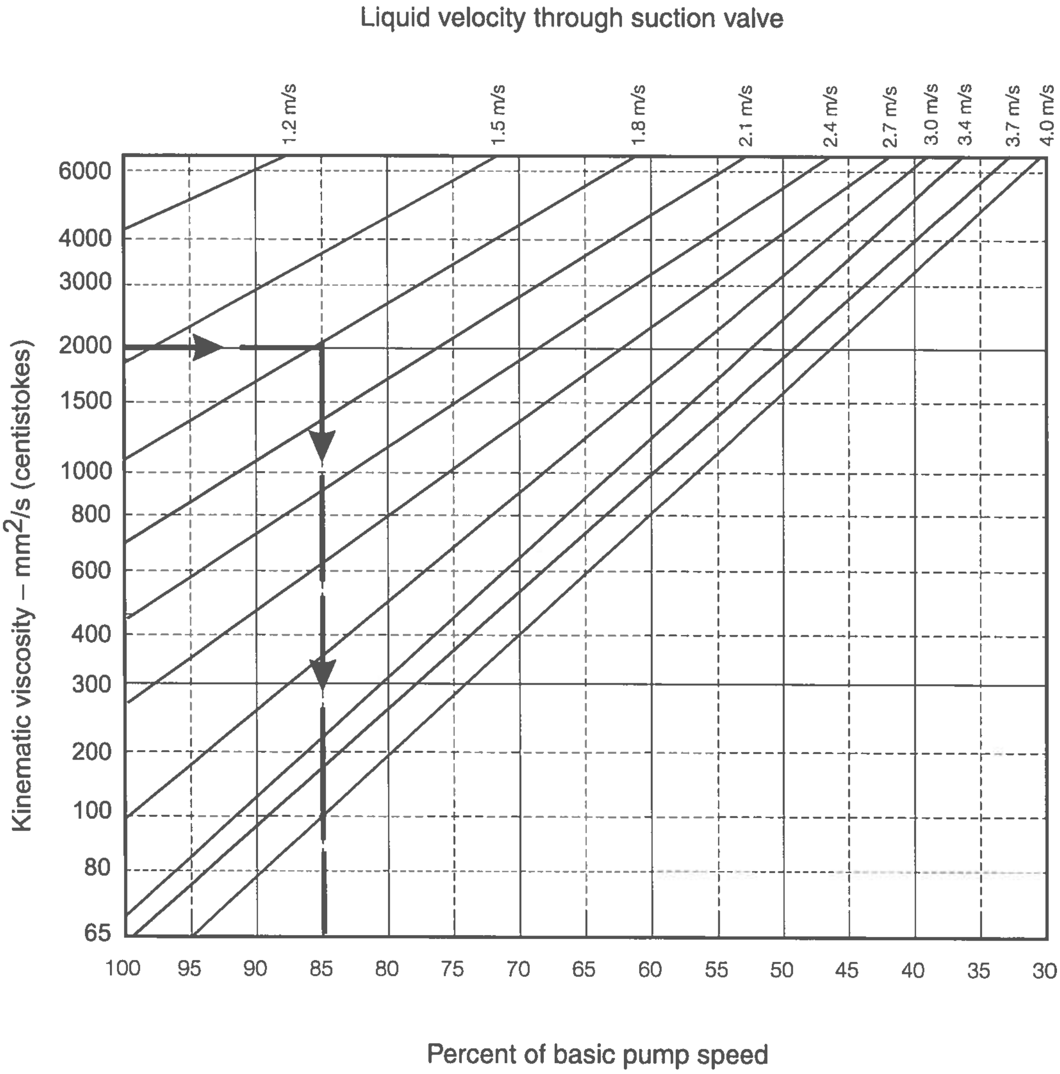


Figure 6.46A — Percent of basic pump speed as a function of average liquid velocity through suction valve (liquid velocity before derating) (Metric units)

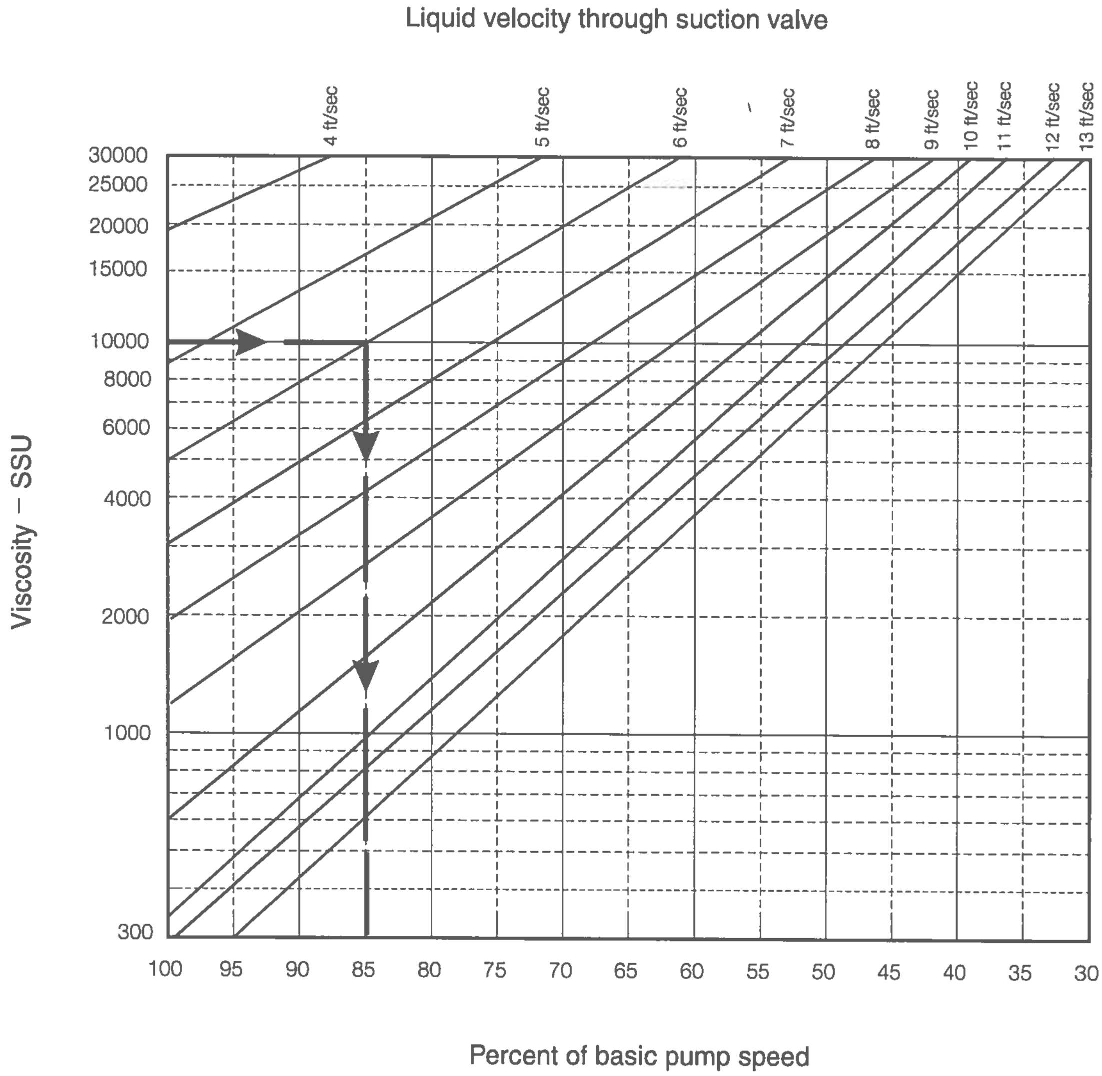


Figure 6.46B — Percent of basic pump speed as a function of average liquid velocity through suction valve (liquid velocity before derating) (US units)

$$v = \frac{556(23)}{(5)(2400)} = 1.07 \text{ m/s}$$

iii) From Figure 6.45, percent of basic speed is 100%;

iv) Viscosity corrected basic speed is 1.00(360) = 360 rpm;

v) Since viscosity corrected basic speed is greater than operating speed, selection is acceptable for high viscosity service.

EXAMPLE (US units): High viscosity pump selection

Given: Q = 100 gpm, p_d = 2000 psi, p_s = 50 psi, Viscosity = 10,000 SSU

1) Select pump based on Q and p_d, disregarding viscosity;

   Selection 2 × 5 × Triplex n = 327 rpm @ 96% VE;

2) Suction valve area A = 3.70 in.^{2} (obtained from manufacturer)

$$v = \frac{.642(100)}{(3)(3.70)} = 5.78 \text{ ft/sec (use 6.0 ft/sec)}$$

3) From Figure 6.45B, percent of basic speed for 10,000 SSU is 85%;

4) Viscosity corrected basic speed is .85 (360) = 306 rpm

   Basic speed for 5-in. stroke pump is 360 rpm from table above;

5) Since viscosity corrected basic speed is less than required operating speed, a new selection is required. Return to Step 1;

   i) New selection 2.65 × 5 quintuplex n = 178 rpm @ 96% VE;

   ii) Suction valve area A – 3.70 in.^{2} (from manufacturer);

$$v = \frac{.642(100)}{(5)(3.70)} = 3.47 \text{ ft/sec (use 4.0 ft/sec)}$$

   iii) From Figure 6.45, percent of basic speed is 100%;

   iv) Viscosity corrected basic speed is 1.00(360) = 360 rpm;

   v) Since viscosity corrected basic speed is greater than operating speed, selection is acceptable for high viscosity service.

6.3.3 Discussion of speeds

6.3.3.1 Factors affecting pump maximum operating speed

– *Liquid characteristics:* Temperature, viscosity, corrosiveness, compressibility, the presence of solids and the presence of dissolved or entrained gas;

– *Application details:* NPSH available, piping design and layout, pulsation dampeners (if any), the ambient temperature, shelter, foundation, driving machinery, protective shut-down devices used, the accessibility of factory service personnel, spare parts and overhaul facilities, as desired;

– *Pump design:* Including valve material, size and type, piston, diaphragm or plunger construction, the choice of packing and packing lubrication, if any, materials used in liquid end and trim, the method of driving pump, and NPSHR.

6.3.3.2 Type of duty

– *Continuous duty:* 8 to 24 hours per day, fully loaded;

– *Light duty:* 3 to 8 hours per day, fully loaded;

– *Intermittent duty:* Up to 3 hours per day, fully loaded;

– *Cyclical operation:* ½ minute loaded out of every 3 minutes;

– *Maintenance level:* Attended or unattended operation. Skill, training and tools of operating and maintenance personnel.

6.3.3.3 Medium speeds

Power pump speeds at or near the manufacturer's published "rated" or "normal" curve includes those applications when clean, cold liquids are involved and provides long life and economical operation, if all important application details are carefully handled and regular, skilled maintenance is provided.

Medium speed selection requires excellent piping layout, good environment, adequate NPSHA, periodic preventive maintenance and lubrication, rigidly fixed piping, and solid pump and prime mover foundations or bases. It may require automatic safety shut-down devices, suction and discharge dampeners and plunger or piston rod packing lubrication.

Medium speeds may be too fast for slurries, marginal NPSH situations, or unattended operation.

6.3.3.4 Slow speeds

Selection of an operating speed below the manufacturer's "rated" or "normal" speed curve is often desirable when any strongly adverse factor is present, such as the following:

- Abrasive liquid (slurry);
- Hazardous liquid;
- Extreme pressure;
- Corrosive chemical;
- High viscosity;
- Unattended operation;
- Poor maintenance;
- No spare parts, or no standby pump;
- High liquid temperature;
- High ambient temperature;
- Extremely long life desired;
- High-cost downtime of related facilities;
- Extreme isolation of site;
- Radioactive liquid;
- Dissolved gas in liquid;
- Borderline suction (NPSHA) situation.

Operation at extremely slow speeds may require supplementary power end lubrication. Cooling of the power end oil may be necessary when hot liquids or ambients occur. Always consult the manufacturer when very hot or very cold liquids are involved. Revisions may be required in construction for these types of applications.

6.3.3.5 High speeds

Selection of speed above manufacturer's "rated" or "normal" curve and/or near his "maximum" or "intermittent" curve (if any) is sometimes merited when intermittent, attended service is involved. High speed selection requires very close attention to all application details, skilled operators and proper pump design. A suction booster pump may be required to obtain sufficient NPSHA.

High speeds imply that only optimum application factors are present and reduced life may occur. Some pumps are inherently designed for high-speed, short-duration and infrequent usage. All conditions of such service should be well understood by all parties prior to the sale. Oil well fracturing, acidizing and cemented plunger pumps are examples of this type of high-speed, intermittent application.

6.3.4 Starting power pumps

6.3.4.1 Pump torque characteristics

Selection of pump driver type can be influenced by the cyclical torque characteristic of a reciprocating power pump. This is especially true when slow speed pumps are employed. Such equipment often lacks enough mechanical mass to smooth out any torque variations imposed by the pump.

Torque fluctuations imposed by reciprocating power pumps on driving equipment vary according to the kind or type of power pump, its number of cylinders and the inertia of the pump and driver rotating masses. Thus, some pumps are inherently "smoother" than others because less cyclical variations in driving torque occur.

Peak torque requirements of power pumps can often be dampened by the use of large-diameter drive pulleys or sprockets with high torsional inertia. These act as "flywheels."

Reduction of peaks in power torque may be possible by reducing discharge pressure surge peaks, since torque and discharge pressure are closely related. Hence, pulsation dampeners which effectively dampen liquid surging also help smooth out torque variations.

Single-acting simplex power pumps have the most uneven torque requirement of all the types of reciprocating power pumps. As the number of cylinders and pumping strokes-per-revolution increase, the torque gets smoother. Thus, a quintuplex pump is said to be very "smooth."

Objectionable amperage fluctuations and heating in polyphase induction motors driving simplex or duplex power pumps can often be minimized by choosing NEMA Design "C" or Design "D" motors. These types of A.C. induction motors provide a "soft" driving torque that reduces line voltage disturbances caused by Design "B" motors.

Large double-acting duplex power pumps in particular tend to cause driver speed surging. This type can also cause heavy amperage surging in an induction type electric motor, unless a special motor type is used or other preventive steps are taken.

6.3.4.2 Pump torque requirements

Application of power pumps requires careful consideration of their starting and running torque demands. These affect the selection of driver motors, motor starters, engines, gear reducers, belts or chain drives, couplings and universal joints. The effect of such loads on an electrical distribution system requires thought, especially if the pump is large.

The starting torque required by a power pump usually falls into one of two general applications, as follows.

6.3.4.2.1 Starting with liquid bypass

Operating person manually opens a bypass valve, or a power-actuated dump valve programmed to open automatically, which bypasses the liquid during the start and the stop function.

A check valve is employed in the pump discharge line. It remains shut as long as the bypass (dump) valve remains fully or partly open (see Figure 6.47).

The liquid pressure exerted on the plungers (or pistons) is largely that caused by liquid mass and friction as it passes through the bypass valve and piping. When correctly sized, the bypass valve and piping cause low back-pressure. Only a relatively small torque is required while bypassing the liquid to a tank.

With liquid bypass, the total starting torque requirement is mainly related to the mechanical inertia of the pump, couplings, gears and motor rotor. These components are heavy, and substantial starting torque may be required. All the liquid in the pump suction line and in the bypass line must be accelerated from standstill to full liquid velocities.

The torque needed to accelerate the entire mechanical hydraulic inertia system depends on the inertia of all the moving parts, including the liquid, the rate of acceleration and the total system friction.

The rate of acceleration is very important, and the starting torque is directly proportional to it. Stated another way: peak torque is inversely proportional to the time duration of acceleration.

A few pumps are provided with mechanical suction valve unloader devices. These devices reduce the total starting torque requirement to that needed to accelerate the mechanical items, such as pump crankshaft, gears, couplings, etc. They stop liquid pumping action by mechanically holding open the pump's suction valves, allowing the liquid inducted into each liquid cylinder to be delivered back into the pump suction. Because no liquid pumping occurs, the driving machinery need provide no torque to accelerate the liquid. Only the torque to overcome mechanical inertia and friction is needed during the start.

After the pump and driver have reached full speed, the suction valve unloading devices are retracted, and normal pumping action then commences. At this point, the driver must supply additional torque to accelerate the liquid system and also to meet the total running torque requirement caused by discharge pressure.

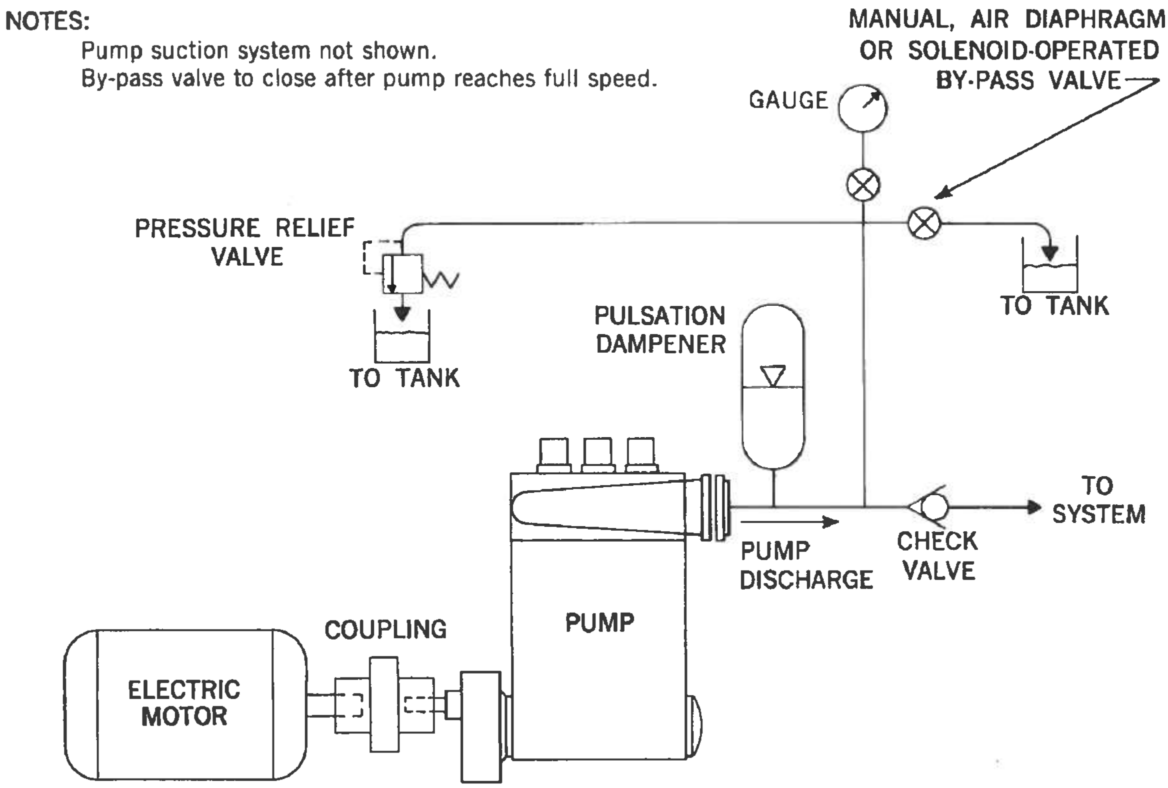
6.3.4.2.2 Starting without liquid bypass

Starting a power pump against discharge pressure may reduce the life of power end bearings and other pump and drive components and may prevent complete priming of the liquid end, resulting in rough operation and reduced rate of flow. This practice should therefore be avoided except in emergency situations.

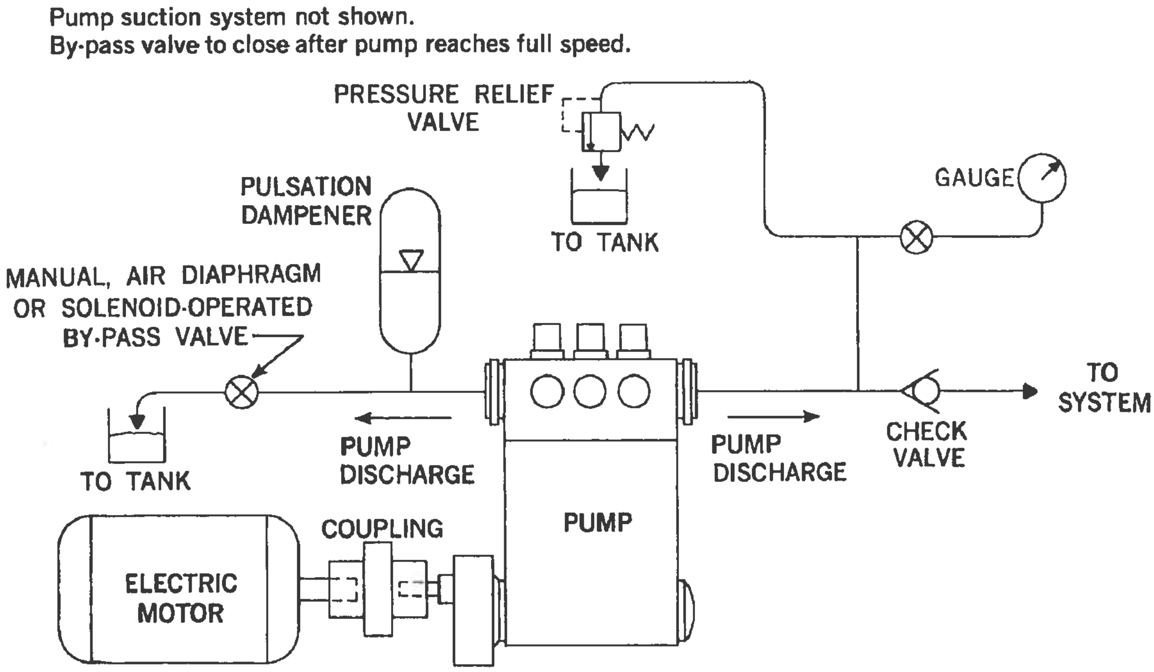
Starting without liquid bypass may be divided into two categories:

First, there are multiple-pump applications where the pump must start against full line pressure.

The discharge pressure already exists, having been developed by the other pumps, and the starting pump must accelerate against it. No liquid bypassing or suction valve unloading is provided, and it may be termed "full-load" starting.



SCHEMATIC: FOR PUMPS WITH SINGLE DISCHARGE CONNECTION



SCHEMATIC: FOR PUMPS WITH DUAL DISCHARGE CONNECTIONS

Figure 6.47 — Schematics of liquid bypass systems

With full-load starting, the torque requirement is high, since the driver must accelerate itself, couplings, gears, pump crankshaft, rods, crossheads and plungers. Additionally, it must accelerate all the liquid in the pump's suction and discharge lines. It must also develop the torque required to move the plungers or pistons against the line pressure already present. If the pump is engine-driven, a clutch or drive coupling of adequate torque and thermal capacity to meet these demands is chosen.

Full-load, across-the-line motor starting can cause a prolonged in-rush of high current. Serious motor overheating or damage may result unless the motor and its starting equipment have been generously selected and sized. An A.C. induction motor provides a fixed starting torque, related to the applied voltage.

Since the starting torque developed by an induction motor is related to the applied voltage and to the size and design of the motor, an across-the-line starter provides the maximum starting torque. A given induction motor develops the same locked-rotor (starting) torque and amperage at this voltage, regardless of the nature of the driven load. High-inertia loads (within motor capability) simply require longer accelerating time than do low-inertia loads.

No advantage is really provided by specifying an induction motor with an extremely high locked-rotor torque rating. Such motors do accelerate faster, but they draw more amperage, and cause more power system disturbance during the start. An A.C. induction motor with a locked-rotor torque rating of 150% of full-load torque is usually sufficient for full-load, across-the-line pump starting. Secondly, another full-load starting situation may occur when a single pump starts with no liquid bypassing provided. Pump discharge pressure is then related very largely to pump speed, rate of flow and acceleration rate.

Consider a single pump which forces liquid through a nearly level long piping system. Pump discharge pressure is zero at the instant of starting. Because resistance to flow is caused by liquid inertia and by pipe friction and fitting restrictions, pump pressure increases in relation to liquid velocity, often such that pump pressure is proportional to the square of flow rate. Thus, it is proportional to the square of pump speed and to the rate of acceleration.

Analysis of the starting torque requirement is complex and depends on the inertia of the accelerating liquid, the size and length of piping, liquid viscosity and density, and the elasticity of the piping. If the mass of liquid is very large, a discharge dampener and a check valve may be advantageous, since these permit the pump and its driver to accelerate faster by first delivering liquid into the dampener, rather than into the line.

No general rules may be given as to the motor locked-rotor torque needed, because each liquid system is different. However, if a large liquid dampener is provided, pipeline pump induction motors with locked-rotor torques of 125% of rated full-load torque are usually adequate. An alternate is the automatic bypass valve and check valve arrangement, with dampener as shown in Figure 6.47.

The arrangement shown in Figure 6.47 is suitable for both single- and multiple-pump applications. It largely relieves the motor of load from liquid inertia (except suction) and isolates the pump from the discharge system pressure and inertia.

The arrangement affords a convenient means of expelling any air trapped in the pump cylinders before placing pressure load on the pump. This is desirable, especially for multicylinder pumps which sometimes become rough and "air bound" after servicing or prolonged idleness. Simply open the liquid bypass valve to allow the liquid to discharge back to the tank, thereby expelling the air. When running smoothly, close bypass valve and thus load the pump.

6.3.4.3 Use of soft start drivers

Another design used to facilitate starting is the use of a hydraulic drive coupling or eddy-current drive coupling. These cause the pump to accelerate more slowly, while the driver quickly reaches full speed. Such a drive coupling permits a reduced rate of pump and liquid acceleration, reducing the torque required.

Reduced voltage starting, with less shock to the pump, the gearing, and the electrical system may be used. Reduced voltage starters include the primary resistor, primary reactor, auto-transformer, part-winding, and wye-delta types. All reduce the motor torque and amperage developed, since all reduce the starting voltage effectively applied to the motor. Hence, the pump always must be unloaded, or the motor may not accelerate to full-load speed.

Reduced voltage starting causes the least disturbance to line voltage and avoids the high torsional stresses and gear tooth loadings which accompany full-voltage, full-load starting.

Experience indicates that a normal-starting-torque electric induction motor (see Table 6.6) provides adequate starting torque when a by-pass valve or suction unloader device is used. (Locked-rotor torque is that developed at standstill by an induction motor, when full line voltage is applied.)

Special conditions, such as starting against discharge pressure, may require a special driver with high starting torque capability.

6.3.5 Electric motor locked-rotor torques

Table 6.6 summarizes minimum locked-rotor torque ratings for NEMA Design "B" 60 Hertz squirrel-cage induction motors expressed as a percent of full-load torque (see *NEMA MG1*).

6.3.6 Inlet system for power pumps

An inlet system for a reciprocating power pump must provide a flow of liquid at a relatively constant pressure to the pump, at a pressure sufficient to prevent cavitation. Cavitation occurs whenever the system dynamic pressure drops below the liquid's vapor pressure, resulting in the creation and eventual collapse of vapor bubbles. If the vapor bubbles are entrained in the liquid or if cavitation occurs in the pump, abnormal pressure pulses or spikes occur due to the collapsing vapor bubbles. It should be noted that entrained undissolved gas in the liquid produces substantially the same pressure spikes as cavitation. When severe, cavitation-produced pulsations and/or gas ingestion can result in one or more of the following conditions:

1) Vibration in the inlet and outlet piping system and related components;
2) Piping fatigue failure;
3) Decrease in volumetric efficiency;
4) Reduced life or failure of liquid end components (packing, valve components, gaskets, etc.);
5) Reduced life or failure of power end components (bearings, crankshaft, crossheads, etc.);
6) Reduced life or failure of drive train components;
7) Failure of pressure boundary parts subjected to pressure pulsations.

It is recommended that the design of the inlet system for a power pump follow these guidelines:

1) The liquid source shown as a tank in Figure 6.48 should be designed with the following features:
   i) Sufficient size to allow entrained gas bubbles to rise to the surface;
   ii) Lines feeding liquid into the tank should extend below minimum liquid level;
   iii) Completely submerged baffle plate separating incoming from outgoing liquid;

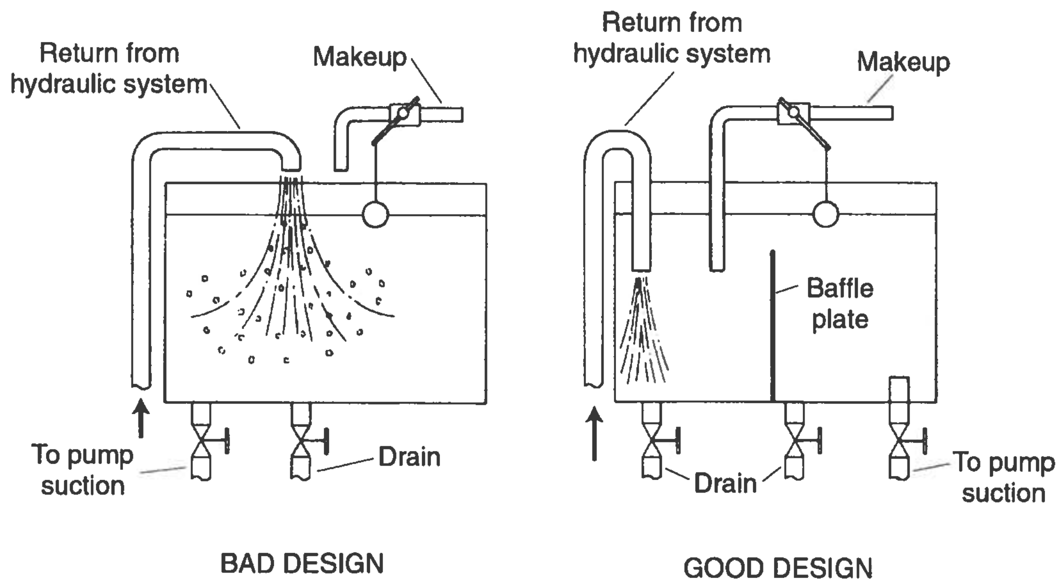


Figure 6.48 — Suction tanks

iv) Vortex breaker at outlet connection (to pump);

v) Inlet piping following the guidelines shown in Figure 6.55.

Table 6.6 — Minimum locked-rotor torque ratings

Rating (kW)	Rating (HP)	1800 RPM Motors	1200 RPM Motors
.75	1	275%	170%
1.12	1 ½	250%	165%
1.50	2	235%	160%
2.25	3	215%	155%
3.75	5	185%	150%
5.60	7 ½	175%	150%
7.50	10	165%	150%
11.2	15	160%	140%
15.0	20	150%	135%
19.0	25	150%	135%
22.5	30	150%	135%
30.0	40	140%	135%
37.5	50	140%	135%
45.0	60	140%	135%
56.0	75	140%	135%
75.0	100	125%	125%
95.0	125	110%	125%
112	150	110%	120%
150	200	100%	120%
190	250	80%	100%
225	300	80%	100%
260	350	80%	100%
300	400	80%	
340	450	80%	
375	500	80%	

NOTE: In the range from .75 through 56 kW (1 through 75 horsepower), the 1800 RPM motors show higher locked-rotor torque ratings than do the 1200 RPM motors. However, from 95 through 260 kW (125 through 350 horsepower), the 1200 RPM motors have larger ratings.

Locked-rotor torques of large motors must be carefully evaluated before final selection.

2) In multiple-pump installations, each pump should be provided with a separate inlet line from the liquid source rather than connecting two or more pumps to a common manifold. The possibility of mutually reinforced pulsations is thus avoided. However, if manifolding is necessary, the manifold and/or inlet piping should have a cross-sectional area equal to or greater than the sum of the cross-sectional areas of the inlet connections of the individual pumps. The connection of the individual pump's piping to the common manifold should be as shown in Figures 6.49 and 6.50.

3) Inlet piping diameters should be equal to or greater than the diameter of the pump's inlet connection. When the inlet piping is larger than the pump's inlet connection, only eccentric reducers should be used for piping size transitions. The eccentric reducer should be placed as close to the pump as possible, with its straight section on top to prevent formation of gas pockets that could cause pressure spikes. See Figure 6.51;

4) High points in the piping system should be minimized to avoid the accumulation of gas. All "horizontal" runs should slope up toward the pump. Any high points in the system

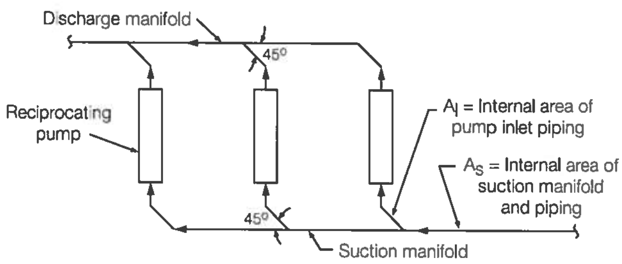


Figure 6.49 — Recommended installation of multiple pumps to common manifolds

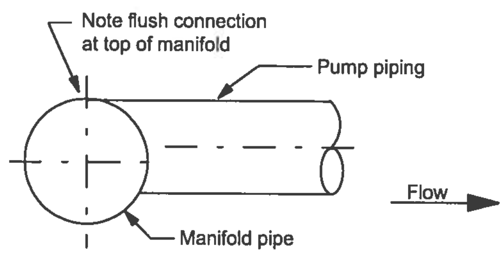


Figure 6.50 — Recommended connection of piping sections

should be provided with vent or bleed-off connections;

5) Inlet piping should be as short and direct as possible with a minimum of turns, bends and restrictions. Pulsations resulting from long inlet lines may not be completely eliminated by pulsation dampeners or by raising the NPSHA of the system. All bends or turns should be made with long-radius elbows, 45-degree elbows, or laterals. If long-radius elbows are used, they should be installed no closer than five pipe diameters from the pump inlet. No two elbows should be closer than eight pipe diameters. The 45-degree elbows are greatly preferred to 90-degree elbows. At no time should 90-degree short-radius elbows be used;

6) The inlet system must provide an absolute pressure that exceeds the sum of the NPSHR of the pump, all friction losses and acceleration head loss, or any losses due to acoustic resonances. Additional suction pressure must be provided if the liquid contains dissolved gas. The NPSHA must exceed the NPSHR of the pump by a sufficient margin to prevent any reduction in volumetric efficiency. As a minimum, a margin of at least 28 kPa (4 psi) should be provided (see Figures 6.53 and 6.54);

NOTE: As the liquid viscosity, specific gravity, or the pump speed increases, the NPSHR for the pump increases and may require additional inlet pressure.

7) For new installations or inlet systems with recently welded tanks, pipe fittings, etc., extreme care must be used to prevent dirt, scale, and weld slag from entering the pump. A startup screen or strainer should be installed as close to the pump as possible. This strainer should be conical and have a flow area three times greater than the flow area of the inlet pipe (see Figure 6.52). If there is any doubt about its regular maintenance or cleaning, a strainer should not be used (clogged strainer may cause more damage to a pump than solids);

8) The inlet line valve should have a flow area equal to that of the inlet line;

9) If a foot valve is used (for a source liquid level below the pump inlet opening), the net flow area should at least equal the flow area of the inlet line;

10) An inlet pressure gauge should be located adjacent to the pump. If possible, the gauge connection should be on the horizontal centerline of the pump's suction pipe.

If a system does not provide sufficient NPSH and cannot be redesigned, it is necessary to do one or more of the following:

1) Install a properly sized pulsation dampener adjacent to the power pump liquid cylinder. Consult dampener and pump manufacturers for correct location of device. A properly charged pulsation dampener may significantly reduce the length of pipe used in the acceleration head equation (see 6.3.6.4, Pulsation dampener);

2) Reduce the power pump NPSHR by selecting a larger, lower-speed unit. The lower speed will also reduce acceleration head;

3) Install a booster (charge) pump.

6.3.6.1 Booster pump

A booster for a reciprocating power pump is normally a centrifugal pump but may be a positive displacement

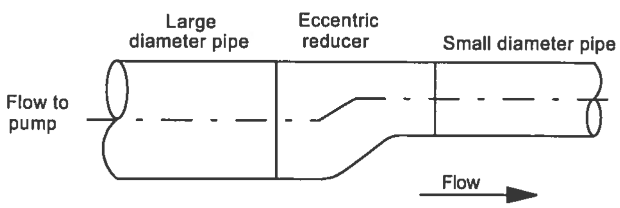


Figure 6.51 — Installation of eccentric reducers

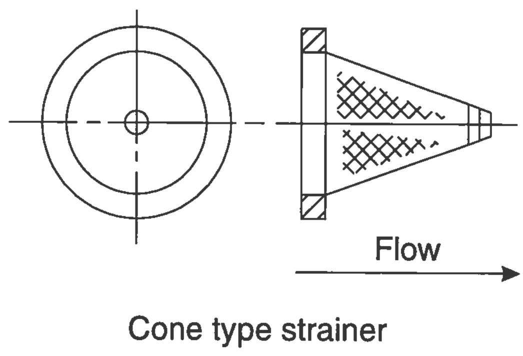


Figure 6.52 — Startup strainers

pump under special conditions (see item 4 below). Care must be exercised in the selection and installation of a booster pump, because improper selection and/or installation can result in increased pulsations and attendant problems. In addition to the recommendations contained in the appropriate section of these Standards, the following are recommended:

1) Install booster pump as close to inlet source as practical (adjacent to inlet line valve in Figure 6.55);

2) The booster pump must add enough pressure to the system to provide sufficient NPSHA to the power pump, allowing for acceleration head, friction losses and pressure pulsations due to acoustical resonances;

3) Install pulsation dampener in inlet line adjacent to the power pump liquid cylinder. Consult dampener and pump manufacturers for proper location of device. The dampener can often be omitted between a centrifugal booster pump and a low-speed power pump under any of the following conditions:

   i) Diameters of inlet and outlet connections of a booster pump are equal to, or larger than, inlet connection on power pump;

   ii) Diameters of all piping between liquid source and power pump are equal to, or larger than, inlet connection of power pump;

   iii) The booster pump is sized for maximum instantaneous rate of flow of the power pump. The following tabulation gives the percentage that the maximum instantaneous rate of flow exceeds the mean rate of flow for each type of power pump;

Type of reciprocating power pump		% over mean rate of flow
Simplex (1)	Single-acting	220%
Duplex (2)	Single-acting	60%
Duplex (2)	Double-acting	27%
Triplex (3)	Single- or double-acting	7%
Quintuplex (5)	Single- or double-acting	2%
Septuplex (7)	Single- or double-acting	1%
Nonuplex (9)	Single- or double-acting	1%

   iv) Acceleration head is calculated not only between booster and power pump but also between liquid source and booster;

4) If the booster pump is a constant-speed positive-displacement pump (such as a motor-driven rotary), a self-regulating bypass valve is required between pumps. The booster pump must be sufficiently oversized to provide the minimum flow required through the bypass valve.

6.3.6.2 Suction tank

It is recommended that a suction tank, if used, be arranged as shown in Figure 6.48 for return and make-up lines.

6.3.6.3 Suction system relationships

The parameters that combine to affect the inlet to the pump are shown diagrammatically in Figures 6.53 and 6.54, which may help in understanding suction system relationships.

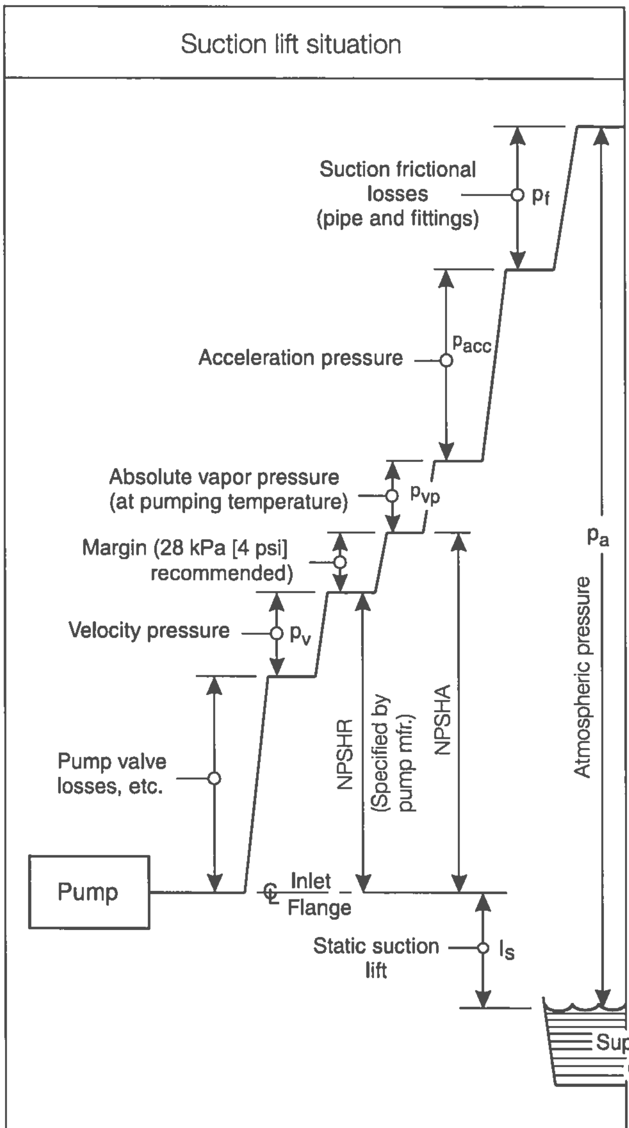
6.3.6.4 Pulsation dampener

A pulsation dampener is a device which reduces liquid pulsations in the suction or discharge piping. It is also sometimes called one of the following:

- Suction chamber;

- Alleviator;

- Discharge chamber;



NOTES

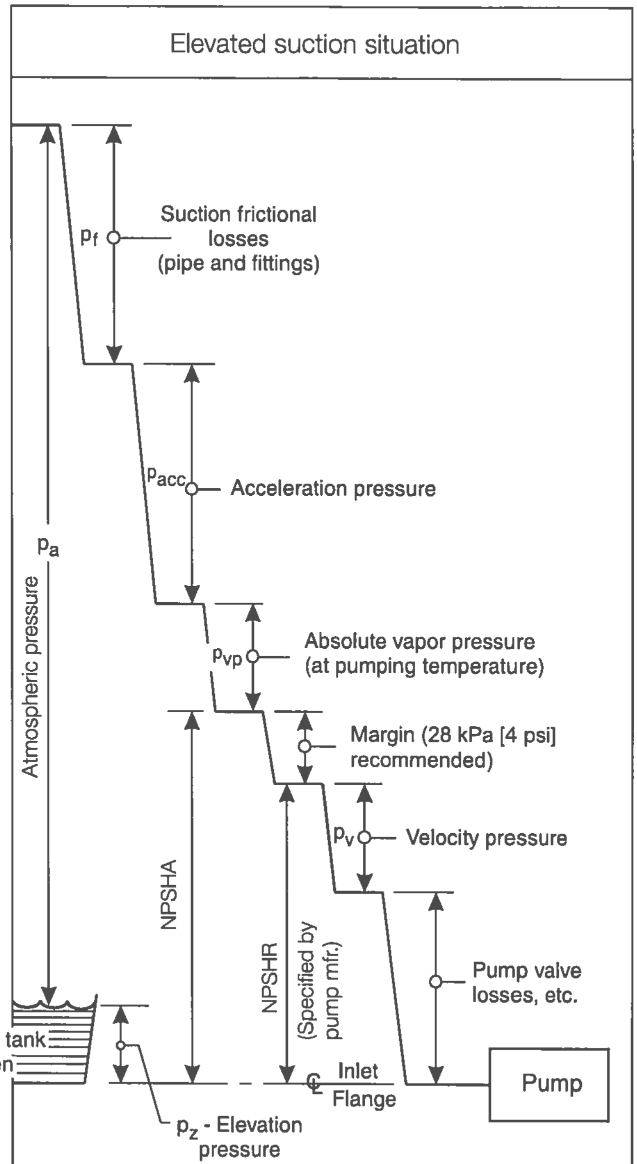
1 If $p_a > (\text{NPSHR} + \text{margin} + p_{vp} + p_{acc} + p_f)$

Then, a suction lift (l_s) is possible, and -

Max. $l_s = p_a - (\text{NPSHR} + \text{margin} + p_{vp} + p_{acc} + p_f)$

2 If actual suction lift $l_s < l_s$ max., then available net positive suction head is -

$\text{NPSHA} = p_a - (p_f + p_{acc} + p_{vp} + l_s)$



NOTES

1 If $p_a < (\text{NPSHR} + \text{margin} + p_{vp} + p_{acc} + p_f)$

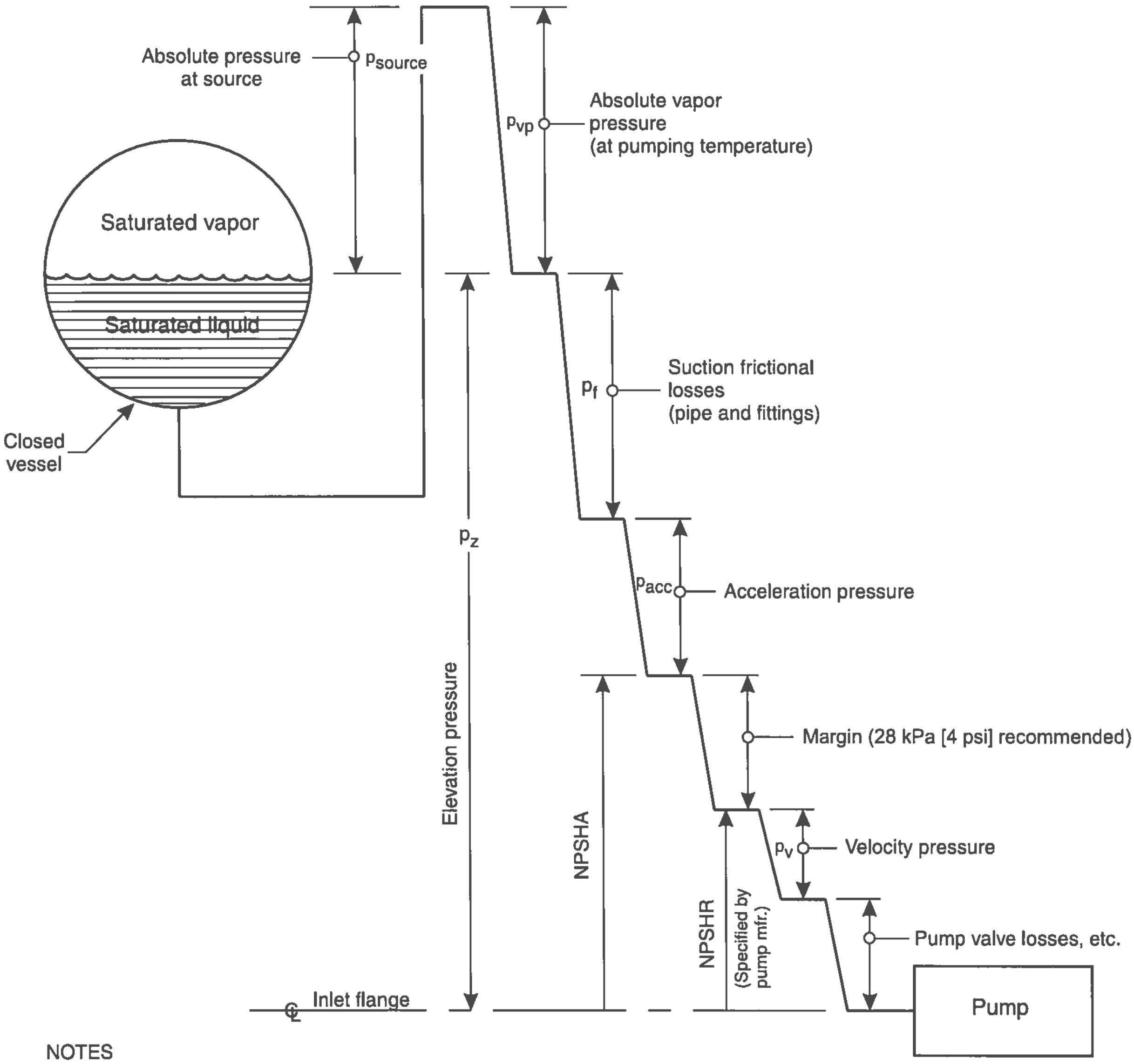
Then, a suction lift is impossible and a positive elevation pressure is necessary

$P_{zmin} = \text{NPSHR} + \text{margin} + P_{vp} + P_{acc} + P_f - P_a$

2 If actual elevation pressure exceeds the minimum required elevation pressure, then -

$\text{NPSHA} = (p_z + p_a) - (p_f + p_{acc} + p_{vp})$

Figure 6.53 — Suction system relationships – open supply



NOTES

1 When absolute pressure at source equals the absolute liquid vapor pressure:

$$p_{source} = p_{vp}$$

Then minimum elevation pressure must equal (or exceed) the sum of all the losses and deductions:

$$p_{zmin} = p_f + p_{acc} + margin + NPSHR$$

2 If actual elevation pressure exceeds the required minimum, then:

$$NPSHA = p_z - [\, p_f + p_{acc} \,]$$

Figure 6.54 — Suction system relationships – closed supply

- Damper;
- Cushion chamber;
- Suction bottle;
- Surge chamber;
- Inlet bottle;
- Suction stabilizer;
- Stand pipe;
- Desurger;
- Air chamber;
- Pulsation suppressor;
- Accumulator.

A good suction and discharge pipe layout for reciprocating pumps of conventional type should not require any devices to compensate for normal variations in velocity of flow in the piping system.

Pulsation dampeners for the inlet piping or discharge lines may be necessary to ensure proper operation of the pump and smooth, quiet operation of the system. Pulsation dampeners are usually required under any of the following conditions:

1) Where the inlet or discharge lines, or both, are of considerable length (greater than 15 meters [50 feet]);

2) When multiple pumps are manifolded together;

3) When the inlet system has insufficient net positive suction head available or to accommodate acceleration head loss;

4) When system requirements specify minimum flow or pressure variations.

The size, type and location of the pulsation dampener will depend upon the type, size and speed of the pump, the liquid and the layout of the piping systems. Recommendations and performance warranties regarding the size and type of pulsation dampener should be obtained from the pulsation dampener manufacturer.

It should be remembered that pulsation dampeners cannot completely alleviate problems related to inadequate piping or system design, insufficient NPSHA, or poor operating practices.

The manufacturers of pulsation dampening equipment base their recommendations on the following:

a) Pump size (bore and stroke);

b) Operating conditions (liquid, pump speed, pressures);

c) Type of pump (duplex, triplex, single-acting, etc.);

d) Estimated attenuation.

Selections specified from the above are not based on any interaction between the pump and piping system into which it is installed.

Acoustic resonances which can be extremely detrimental to both the pump and piping system are possible due to the response of the piping system to the pulsating flow of a positive displacement pump. It is even more likely and critical in multi-pump installations.

These acoustic resonances can be predicted with digital or analog analysis of the piping system and its interaction with the flow and pressure characteristics of the pump. For critical installations, it is strongly recommended that final dampener selection be based on such an analysis.

Pulsation dampeners, if used, should be considered as a part of the piping system, rather than as a pump accessory.

In general, suction pulsation dampeners are more frequently required as a part of the piping system, rather than as a pump accessory.

In general, suction pulsation dampeners are more frequently required than are discharge pulsation dampeners. In the following list, pump types are arranged in order of their relative smoothness of flow:

- Nonuplex power pump — Single-acting;
- Septuplex power pump — Single-acting;
- Quintuplex power pump — Single- and double-acting;

- Triplex power pump — Single- and double-acting;
- Duplex power pump — Single- and double-acting;
- Simplex power pump — Double-acting;
- Simplex power pump — Single-acting.

For pulsation dampeners which require a gas charge, provision should be made to keep them charged with nitrogen or a similar inert gas. A liquid level gauge is desirable to permit a check on the amount of air in a conventional air chamber.

It is necessary to change the gas charge pressure if the system pressure is changed. The dampener manufacturer's recommendations should be closely followed.

Pulsation dampeners, particularly on the inlet, should be located as close as possible to the pump and in such a position that they will absorb the impact of the moving liquid column and thus cushion the pulsations in the most efficient manner.

On high-speed power pumps, the chamber air volume should be at least 1 to 1 1/2 times the pump displacement per revolution.

6.3.7 Discharge piping

To facilitate starting and eliminate air, a bypass valve should be installed close to the pump. Also to protect the pump, a stop valve and a check valve should be employed (see Figure 6.55). If an increaser is used to increase the size of the piping, it should be placed between the check valve and the pump.

The vibration in pipe lines must be minimized, with lines as short and as direct as practicable. When a change in direction is required, long-radius elbows and tees or a 45-degree bend in the pipe itself should be utilized. "Dead ends" must be avoided.

Hydraulic systems, using quick-closing valves or similar mechanisms, must provide some means for absorbing the shock resulting from the sudden opening or closing of valves.

Adequate provisions should be made for anchorage of high-pressure piping.

6.3.7.1 Relief valve

The insertion of a discharge relief valve of suitable size for the rate of flow of the pump, set to open at a pressure above the operating discharge pressure required of the pump, is mandatory because of the safety it affords. The relief valve should be placed in the discharge line close to the pump and ahead of any other valves.

Full-opening, pilot-operated, or shear pin relief valves, or burst-discs, which require little or no over-pressure to develop fully-open flow capacity, are often preferable to spring-loaded relief valves. Spring types may require considerable over-pressure to compress the spring before becoming fully open, creating pump overload.

Pressure-relief valves have a "set" pressure, which may be described as the pressure at which the relief valve cracks and begins to open, allowing some flow to pass through. As additional pressure is applied, above this "set" or "crack" pressure, the spring-loaded relief valve will gradually increase its port area until the valve is fully open.

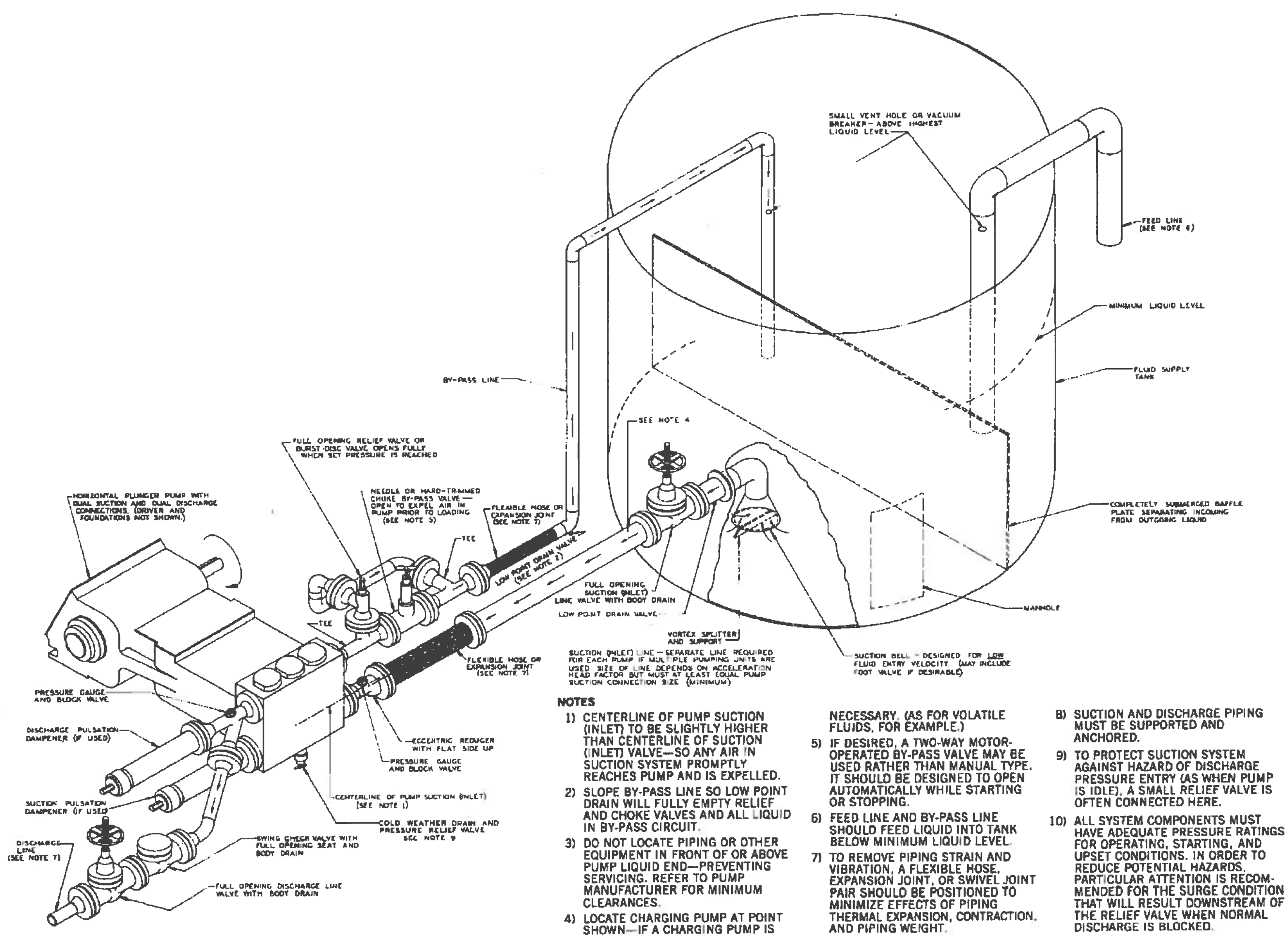
When fully open, the relief valve must have sufficient area so it will relieve the full rate of flow of the pump without excessive over-pressure. Available spring-loaded relief valves differ among manufacturers in the extent of over-pressure needed to open from the barely cracked to the fully open, fully relieved position. This range is generally 10 percent to 25 percent above the set pressure, depending on spring design. By choosing a larger valve, this increase may be reduced.

6.3.8 Calculating volumetric efficiency for water (η_V)

The volumetric efficiency of a reciprocating pump, based on rate of flow at suction conditions, using table of water compressibility, shall be calculated as follows:

$$\text{Metric: } \eta_v = \frac{1 - \left[p_H \frac{\beta_t}{6.895}\left(1 + \frac{c}{d}\right)\right]}{1 - p_H \frac{\beta_t}{6.895}} - S$$

$$\text{US Units: } \eta_v = \frac{1 - \left[p_H \beta_t \left(1 + \frac{c}{d}\right)\right]}{1 - p_H \beta_t} - S$$



NOTES

1) CENTERLINE OF PUMP SUCTION (INLET) TO BE SLIGHTLY HIGHER THAN CENTERLINE OF SUCTION (INLET) VALVE—SO ANY AIR IN SUCTION SYSTEM PROMPTLY REACHES PUMP AND IS EXPELLED.

2) SLOPE BY-PASS LINE SO LOW POINT DRAIN WILL FULLY EMPTY RELIEF AND CHOKE VALVES AND ALL LIQUID IN BY-PASS CIRCUIT.

3) DO NOT LOCATE PIPING OR OTHER EQUIPMENT IN FRONT OF OR ABOVE PUMP LIQUID END—PREVENTING SERVICING. REFER TO PUMP MANUFACTURER FOR MINIMUM CLEARANCES.

4) LOCATE CHARGING PUMP AT POINT SHOWN—IF A CHARGING PUMP IS NECESSARY. (AS FOR VOLATILE FLUIDS, FOR EXAMPLE.)

5) IF DESIRED, A TWO-WAY MOTOR-OPERATED BY-PASS VALVE MAY BE USED RATHER THAN MANUAL TYPE. IT SHOULD BE DESIGNED TO OPEN AUTOMATICALLY WHILE STARTING OR STOPPING.

6) FEED LINE AND BY-PASS LINE SHOULD FEED LIQUID INTO TANK BELOW MINIMUM LIQUID LEVEL.

7) TO REMOVE PIPING STRAIN AND VIBRATION, A FLEXIBLE HOSE, EXPANSION JOINT, OR SWIVEL JOINT PAIR SHOULD BE POSITIONED TO MINIMIZE EFFECTS OF PIPING THERMAL EXPANSION, CONTRACTION, AND PIPING WEIGHT.

8) SUCTION AND DISCHARGE PIPING MUST BE SUPPORTED AND ANCHORED.

9) TO PROTECT SUCTION SYSTEM AGAINST HAZARD OF DISCHARGE PRESSURE ENTRY (AS WHEN PUMP IS IDLE), A SMALL RELIEF VALVE IS OFTEN CONNECTED HERE.

10) ALL SYSTEM COMPONENTS MUST HAVE ADEQUATE PRESSURE RATINGS FOR OPERATING, STARTING, AND UPSET CONDITIONS. IN ORDER TO REDUCE POTENTIAL HAZARDS, PARTICULAR ATTENTION IS RECOMMENDED FOR THE SURGE CONDITION THAT WILL RESULT DOWNSTREAM OF THE RELIEF VALVE WHEN NORMAL DISCHARGE IS BLOCKED.

Figure 6.55 — Suggested piping system for power pumps

Where:

β_t = Compressibility factor at temperature t (see Tables 6.7 and 6.8);

c = Liquid chamber volume in the passages of chamber between valves when plunger is at the end of discharge stroke in mm^3 ($in.^3$) (see Figure 6.56);

d = Volume displacement per plunger in mm^3 ($in.^3$) (see Figure 6.56);

p_H = Discharge pressure minus suction pressure in kPa (psi);

S = Slip, expressed as a decimal.

EXAMPLE (Metric) – Find the volumetric efficiency of a reciprocating pump with the following conditions:

Type of pump	25-mm dia. plunger × 125-mm stroke triplex
Liquid pumped	water
Suction pressure	0 kPa
Discharge pressure	12,300 kPa
Pumping temperature	60°C
c	2,081,000 mm^3
d	578,000 mm^3
S	.02

Find β_t from table of water compressibility (Tables 6.7 and 6.8).

β_t = .00000305 at 60°C and 12,400 kPa. Calculate volumetric efficiency:

$$\eta_v = \frac{1 - \left[p_H \frac{\beta_t}{6.895}\left(1 + \frac{c}{d}\right)\right]}{1 - p_H \frac{\beta_t}{6.895}} - S$$

$$= \frac{1 - \left[(12300 - 0)\left(\frac{.00000305}{6.895}\right)\left(1 + \frac{2,081,000}{578,000}\right)\right]}{1 - (12300 - 0)\left(\frac{.00000305}{6.895}\right)} - .02$$

= 0.96

= 96 %

EXAMPLE (US Units) – Find the volumetric efficiency of a reciprocating pump with the following conditions:

Type of pump	3-in. dia. plunger × 5-in. stroke triplex
Liquid pumped	water
Suction pressure	Zero psig
Discharge pressure	1785 psig
Pumping temperature	140°F
c	127 cu in.
d	35.3 cu in.
S	.02

Find β_t from table of water compressibility (Tables 6.7 and 6.8).

β_t = .00000305 at 140°F and 1800 psia. Calculate volumetric efficiency:

$$\eta_v = \frac{1 - \left[p_H \beta_t\left(1 + \frac{c}{d}\right)\right]}{1 - p_H \beta_t} - S$$

$$= \frac{1 - \left[(1785 - 0)(.00000305)\left[1 + \frac{127}{35.3}\right]\right]}{1 - (1785 - 0)(.00000305)} - .02$$

= 0.96

= 96 %

6.3.9 Calculating volumetric efficiency for hydrocarbons (η_v)

The volumetric efficiency of a reciprocating pump based on rate of flow at suction conditions, using compressibility factors for hydrocarbons, shall be calculated as follows:

$$\eta_v = 1 - \left[S - \frac{c}{d}\left(1 - \frac{\rho_d}{\rho_s}\right)\right]$$

Table 6.7 — Water compressibility factor $\beta_t \times 10^{-6}$ (US units)

Psia	Temperature – °F 32	63	104	140	176	212	248	284	326	356	392	428	464	500	536	572	608	644	680
200	3.12	3.06	3.06	3.12	3.23	3.40	3.66	4.00	4.47	5.11	6.00	7.27							
400	3.11	3.05	3.05	3.11	3.22	3.39	3.64	3.99	4.45	5.09	5.97	7.21							
600	3.10	3.05	3.05	3.10	3.21	3.39	3.63	3.97	4.44	5.07	5.93	7.15	8.95						
800	3.10	3.04	3.04	3.09	3.21	3.38	3.62	3.96	4.42	5.04	5.90	7.10	8.85	11.6					
1000	3.09	3.03	3.03	3.09	3.20	3.37	3.61	3.95	4.40	5.02	5.87	7.05	8.76	11.4	16.0				
1200	3.08	3.02	3.02	3.08	3.19	3.36	3.60	3.94	4.39	5.00	5.84	7.00	8.68	11.2	15.4				
1400	3.07	3.01	3.01	3.07	3.18	3.35	3.59	3.92	4.37	4.98	5.81	6.95	8.61	11.1	15.1	23.0			
1600	3.06	3.00	3.00	3.06	3.17	3.34	3.58	3.91	4.35	4.96	5.78	6.91	8.53	10.9	14.8	21.9			
1800	3.05	2.99	3.00	3.05	3.16	3.33	3.57	3.90	4.34	4.94	5.75	6.87	8.47	10.8	14.6	21.2	36.9		
2000	3.04	2.99	2.99	3.04	3.15	3.32	3.56	3.88	4.32	4.91	5.72	6.83	8.40	10.7	14.3	20.7	34.7		
2200	3.03	2.98	2.98	3.04	3.14	3.31	3.55	3.87	4.31	4.89	5.69	6.78	8.33	10.6	14.1	20.2	32.9	86.4	
2400	3.02	2.97	2.97	3.03	3.14	3.30	3.54	3.85	4.29	4.87	5.66	6.74	8.26	10.5	13.9	19.8	31.6	69.1	
2600	3.01	2.96	2.96	3.02	3.13	3.29	3.53	3.85	4.28	4.85	5.63	6.70	8.20	10.4	13.7	19.4	30.5	61.7	
2800	3.00	2.95	2.96	3.01	3.12	3.28	3.52	3.83	4.26	4.83	5.61	6.66	8.14	10.3	13.5	19.0	29.6	57.2	238.2
3000	3.00	2.94	2.95	3.00	3.11	3.28	3.51	3.82	4.25	4.81	5.58	6.62	8.08	10.2	13.4	18.6	28.7	53.8	193.4
3200	2.99	2.94	2.94	3.00	3.10	3.27	3.50	3.81	4.23	4.79	5.55	6.58	8.02	10.1	13.2	18.3	27.9	51.0	161.0
3400	2.98	2.93	2.93	2.99	3.09	3.26	3.49	3.80	4.22	4.78	5.53	6.54	7.96	9.98	13.0	17.9	27.1	48.6	138.1
3600	2.97	2.92	2.93	2.98	3.09	3.25	3.48	3.79	4.20	4.76	5.50	6.51	7.90	9.89	12.9	17.6	26.4	45.4	122.4
3800	2.96	2.91	2.92	2.97	3.08	3.24	3.47	3.78	4.19	4.74	5.47	6.47	7.84	9.79	12.7	17.3	25.8	44.5	110.8
4000	2.95	2.90	2.91	2.97	3.07	3.23	3.46	3.76	4.17	4.72	5.45	6.43	7.78	9.70	12.5	17.1	25.2	42.8	101.5
4200	2.95	2.90	2.90	2.96	3.06	3.22	3.45	3.75	4.16	4.70	5.42	6.40	7.73	9.62	12.4	16.8	24.6	41.3	93.9
4400	2.94	2.89	2.90	2.95	3.05	3.21	3.44	3.74	4.14	4.68	5.40	6.36	7.68	9.53	12.2	16.5	24.1	40.0	87.6
4600	2.93	2.88	2.89	2.94	3.05	3.20	3.43	3.73	4.13	4.66	5.37	6.32	7.62	9.44	12.1	16.3	23.6	38.8	82.3
4800	2.92	2.87	2.88	2.94	3.04	3.20	3.42	3.72	4.12	4.64	5.35	6.29	7.57	9.36	12.0	16.0	23.2	37.6	77.7
5000	2.91	2.87	2.87	2.93	3.03	3.10	3.41	3.71	4.10	4.63	5.32	6.25	7.52	9.28	11.8	15.8	22.7	36.6	73.9
5200	2.90	2.85	2.87	2.92	3.02	3.18	3.40	3.69	4.09	4.61	5.30	6.22	7.47	9.19	11.7	15.6	22.3	35.6	70.3
5400	2.90	2.85	2.86	2.91	3.01	3.17	3.39	3.68	4.07	4.59	5.27	6.19	7.41	9.12	11.6	15.3	21.9	34.6	66.9

NOTES:

1) Compressibility factor $\beta_t \times 10^{-6}$ = contraction in unit volume per psi pressure
2) Compressibility from 14.7 psia, 32°F to 212°F and from saturation pressure above 212°F

Table 6.8 — Water compressibility $\beta_t \times 10^{-6}$ (US units)

Pressure psia	Temperature – °F			Pressure psia	Temperature – °F		
	68	212	392		68	212	392
6000	2.84	3.14	5.20	22000	2.61	2.42	2.75
7000	2.82	3.10	5.09	23000	2.59	2.38	3.68
8000	2.80	3.05	4.97	24000	2.58	2.33	3.61
9000	2.78	3.01	4.87	25000	2.57	2.29	3.55
10000	2.76	2.96	4.76	26000	2.56	2.24	3.49
11000	2.75	2.92	4.66	27000	2.55	2.20	3.43
12000	2.73	2.87	4.57	28000	2.55	2.15	3.37
13000	2.71	2.83	4.47	29000	2.54	2.11	3.31
14000	2.70	2.78	4.38	30000	2.53	2.06	3.26
15000	2.69	2.74	4.29	31000	2.52	2.02	3.21
16000	2.67	2.69	4.21	32000	2.51	1.97	3.16
17000	2.66	2.65	4.13	33000	2.50	1.93	3.11
18000	2.65	2.60	4.05	34000	2.49	1.88	3.07
19000	2.64	2.56	3.97	35000	2.49	1.84	3.03
20000	2.63	2.51	3.89	36000	2.48	1.79	2.99
21000	2.62	2.47	3.82				

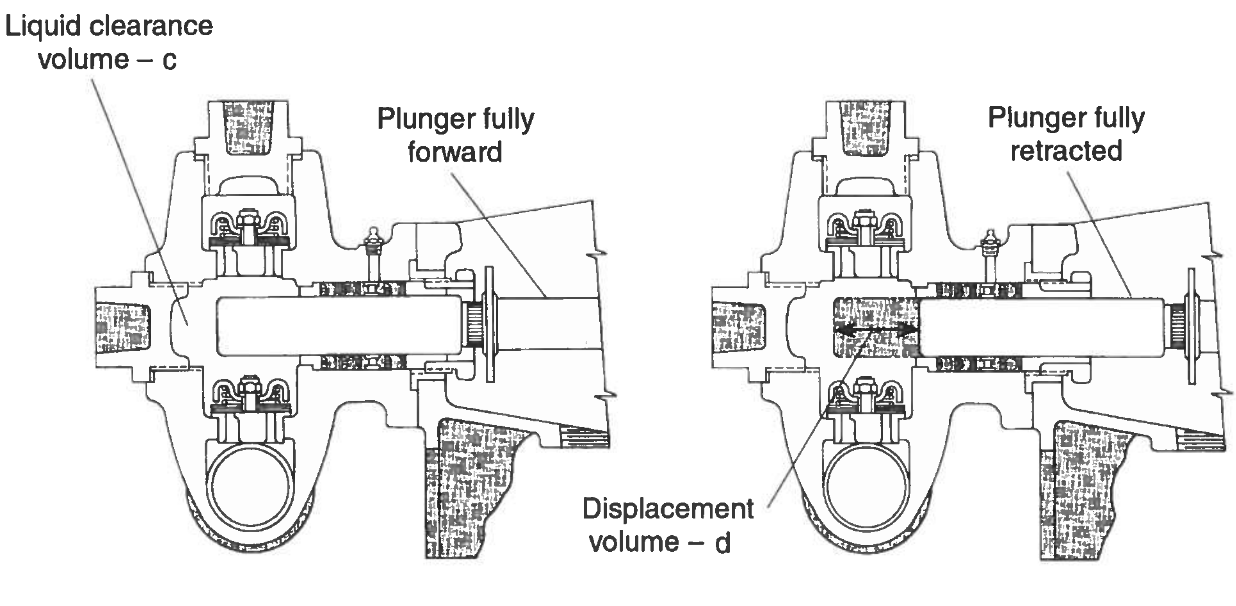


Figure 6.56 — Plunger movement when calculating volumetric efficiency

Where:

η_v = Volumetric efficiency expressed as a decimal;

S = Slip expressed as a decimal;

c = Liquid chamber volume in the passages of chamber between valves, when plunger is at the end of discharge stroke in mm^3 ($in.^3$);

d = Volume displacement per plunger in mm^3 ($in.^3$);

ρ = Density in kg/m^3 (lb/f^3) = $K \times \omega \times \rho_w$

ρ_s = Density in kg/m^3 (lb/ft^3) at suction pressure;

ρ_d = Density in kg/m^3 (lb/ft^3) at discharge pressure;

ρ_w = Density of water 1000 kg/m^3 (62.4 lb/ft^3)

p_s = Suction pressure in kPa (psia);

p_d = Discharge pressure in kPa (psia);

p_c = Critical pressure of liquid in kPa (psia) (see Table 6.9A and B);

p_r = Reduced pressure ratio;

$= \dfrac{\text{Actual pressure in kPa (psia)}}{\text{Critical pressure in kPa (psia)}} = \dfrac{p}{p_c}$;

p_{rs} = Reduced suction pressure ratio = $\dfrac{p_s}{p_c}$;

p_{rd} = Reduced discharge pressure ratio = $\dfrac{p_d}{p_c}$;

t = Temperature, in degrees Kelvin (°K) [Rankine (°R)];

t_s = Suction temperature in degrees Kelvin (Rankine);

t_d = Discharge temperature in degrees Kelvin (Rankine);

t_c = Critical temperature of liquid, in degrees Kelvin (Rankine) (see Table 6.9A and B);

t_r = Reduced temperature ratio;

$= \dfrac{\text{actual temp. in degrees Kelvin (Rankine)}}{\text{critical temp. in degrees Kelvin (Rankine)}}$;

$= \dfrac{t}{t_c}$ (See Figure 6.57);

t_{rs} = Reduced suction temperature ratio $\dfrac{t_s}{t_c}$;

t_{rd} = Reduced discharge temperature ratio $\dfrac{t_d}{t_c}$;

ω = Expansion factor of liquid (see Figure 6.57);

K = Characteristic constant for any one liquid which is established by density measurements and the corresponding values of ω (see Table 6.9A and B);

Table 6.9A — Physical properties of hydrocarbons (Metric)

Carbon atoms	Name	t_c degrees Kelvin	p_c kPa	K
1	Methane	190	4640	3.679
2	Ethane	306	4944	4.429
3	Propane	370	4427	4.803
4	Butane	426	3751	5.002
5	Pentane	471	3323	5.128
6	Hexane	508	2986	5.216
7	Heptane	540	2717	5.285
8	Octane	569	2496	5.340
9	Nonane	596	2289	5.382
10	Decane	619	2124	5.414
12	Dodecane	658	1875	5.459
14	Tetradecane	693	1682	5.483
16	Hexadecane	722	1524	5.48[a]
18	Octadecane	747	1393	5.49[a]
20	Eicosane	767	1289	5.5[a]
25	Pentacosane	814	1076	5.5[a]
30	Triacontane	858	917	5.5[a]
35	Pentatriacontane	894	827	5.5[a]
40	Tetracontane	930	745	5.5[a]
45	Pentatetracontane	967	690	5.4[a]

[a] Based on experimental density, questionable because near melting point.

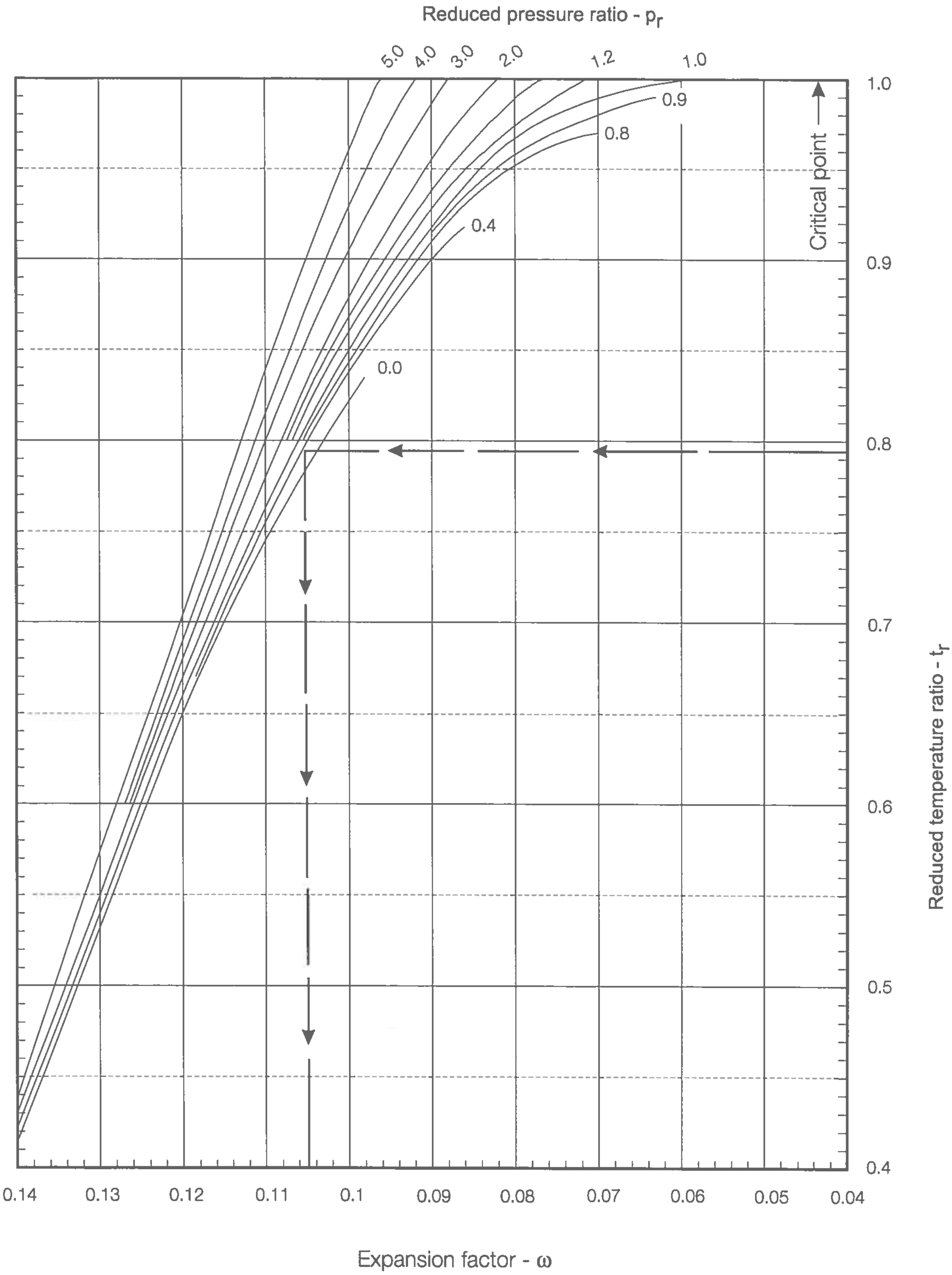


Figure 6.57 — Thermal expansion and compressibility of liquids

Table 6.9B — Physical properties of hydrocarbons (US Units)

Carbon atoms	Name	t_c degrees Rankine	p_c lb per sq in	K
1	Methane	343	673	3.679
2	Ethane	550	717	4.429
3	Propane	666	642	4.803
4	Butane	766	544	5.002
5	Pentane	847	482	5.128
6	Hexane	915	433	5.216
7	Heptane	972	394	5.285
8	Octane	1025	362	5.340
9	Nonane	1073	332	5.382
10	Decane	1114	308	5.414
12	Dodecane	1185	272	5.459
14	Tetradecane	1248	244	5.483
16	Hexadecane	1300	221	5.48[a]
18	Octadecane	1345	202	5.49[a]
20	Eicosane	1380	187	5.5[a]
25	Pentacosane	1465	156	5.5[a]
30	Triacontane	1545	133	5.5[a]
35	Pentatriacontane	1610	120	5.5[a]
40	Tetracontane	1675	108	5.5[a]
45	Pentatetracontane	1740	100	5.4[a]

[a] Based on experimental density, questionable because near melting point.

EXAMPLE (Metric) – Find volumetric efficiency of the previous reciprocating pump example with the following new conditions:

Type of pump	75-mm dia. plunger × 125-mm stroke triplex
Liquid pumped	propane
Suction temp.	21°C = 294°K
Discharge temp.	29°C = 300°K
Suction pressure	1772 kPa
Discharge pressure	13,280 kPa

Find density at suction pressure:

$$t_{rs} = \frac{t_s}{t_c} = \frac{294}{370} = 0.795$$

$$p_{rs} = \frac{p_s}{p_c} = \frac{1772}{4427} = 0.400$$

K = 4.803 (From Table 6.9A, propane)

ω = 0.105 (from Figure 6.57)

$$\begin{aligned}\rho_s &= K \times \omega \times 1000 \\ &= 4.803 \times 0.105 \times 1000 \\ &= 504.3\ \text{kg/m}^3\end{aligned}$$

Find density at suction pressure:

$$t_{rd} = \frac{t_d}{t_c} = \frac{300}{370} = 0.81$$

$$p_{rd} = \frac{p_d}{p_c} = \frac{13280}{4427} = 3.00$$

ω = 0.109 (From Figure 6.57)

$$\begin{aligned}\rho_d &= K \times \omega \times 1000 \\ &= 4.803 \times 0.109 \times 1000 \\ &= 523.5\ \text{kg/m}^3\end{aligned}$$

Therefore:

$$\text{Vol. Eff.} = 1 - \left[S - \frac{c}{d}\left(1 - \frac{\rho_d}{\rho_s}\right)\right]$$

$$= 1 - \left[0.02 - \frac{2{,}081{,}000}{578{,}000}\left(1 - \frac{523.5}{504.3}\right)\right]$$

$$= 0.883 = 88.3 \text{ percent}$$

EXAMPLE (US Units) – Find volumetric efficiency of the previous reciprocating pump example with the following new conditions:

Type of pump	3-in. dia. plunger × 5-in. stroke triplex
Liquid pumped	propane
Suction temp.	70°F = 530°R
Discharge temp.	80°F = 540°R
Suction pressure	242 psig = 257 psia
Discharge pressure	1911 psig = 1426 psia

Find density at suction pressure:

$$t_{rs} = \frac{t_s}{t_c} = \frac{530}{666} = 0.795$$

$$p_{rs} = \frac{p_s}{p_c} = \frac{257}{642} = 0.400$$

K = 4.803 (From Table 6.9B, propane)

ω = 0.105 (from Figure 6.57)

$$\begin{aligned}\rho_s &= K \times \omega \times 62.4 \\ &= 4.803 \times 0.105 \times 62.4 \\ &= 31.4 \text{ lb. per cu. ft.}\end{aligned}$$

Find density at discharge pressure:

$$t_{rd} = \frac{t_d}{t_c} = \frac{540}{666} = 0.81$$

$$p_{rd} = \frac{p_d}{p_c} = \frac{1926}{642} = 3.00$$

ω = .109 (From Figure 6.57)

$$\begin{aligned}\rho_d &= K \times \omega \times 62.4 \\ &= 4.803 \times .109 \times 62.4 \\ &= 32.6 \text{ lb. per cu. ft.}\end{aligned}$$

Therefore:

$$\text{Vol. Eff.} = 1 - \left[S - \frac{c}{d}\left(1 - \frac{\rho_d}{\rho_s}\right)\right]$$

$$= 1 - \left[0.02 - \frac{127}{35.3}\left(1 - \frac{32.6}{31.4}\right)\right]$$

$$= 0.883 = 88.3 \text{ percent}$$

6.3.10 Piston and plunger pumps for slurry service

6.3.10.1 Typical service

Reciprocating pumps are used to handle slurries for in-plant process and pipeline applications.

As pipeline systems grow in length and rate of flow, they require higher pressures. This is advantageous for a reciprocating pump which has the desirable characteristic of maintaining high efficiency at any desired rate of flow and pressure. The type used, be it piston or plunger, single- or double-acting, depends on the particular application. Developed pressures range up to 20,000 kPa (3000 psi) with overall efficiencies of 80 to 90 percent.

Basic construction may or may not be the same as for clear liquid application. The differences may be in type of valves, addition of surge chambers, or liquid injection into the lower portion of the stuffing box.

Reciprocating slurry pumps are so designed that the liquid end parts which are subject to deteriorating effects of slurries can be easily and quickly replaced without dismantling any other major pump component. These parts are usually replaced in accordance with a preventive maintenance program. The scheduled replacement time is based on the user's experience with the slurry pumped. Replacement timing should be such that the part still performs adequately and does not wear to the point of causing failure of other parts of the machine.

Hydraulic passages should be sized so that the lowest velocity of the liquid will be above the critical carrying velocity of 1.2 to 1.8 m/s (4 to 6 ft/s) as an average. The highest velocity should be below that which causes excessive erosion. Typical average operating velocities through a reciprocating slurry pump's passages are 1.8 to 3.6 m/s (6 to 12 ft/s).

Lubrication and flushing of packing are extremely important. Metered, clear, external injection, which is timed to the position of the plunger during its stroke, or continuous flow injection is employed to achieve this. The mode of flushing will depend on whether dilution of the liquid pumped by the flushing liquid can be tolerated.

To protect the main stuffing box packing, clear liquid is usually injected into the stuffing box between the bottom of the throat bushing and the packing. The injection lines are selected to withstand full working pressure and have a safety check valve located between the stuffing box and the injection liquid source to prevent accidental backflow of slurry into the clear liquid system.

Valves for use in slurry service are designed for velocities between 1.8 to 3.6 m/s (6 and 12 ft/s) to reduce erosion and abrasion of the valve seat and other valve components. Valve construction usually has replaceable valve inserts that are made of an elastomer or polymer. Metal-to-metal ball valves may be used depending on the slurry, material, carrier liquid and temperature.

Special considerations must be given to the slurry abrasion, attrition, particle size and concentration. Slurry particle size has an influence on the valve lift and the ability of an elastomer valve to seat. Experience to date shows that slurry concentrations of up to 65 percent by weight can be handled successfully.

Suction pressure on a pump handling a slurry is usually higher than when handling clear liquid. This is to take into account the acceleration head of the solids and gases entrained with the liquid. Likewise, pulsation dampeners are usually larger on the suction and discharge side of the pump for the same reasons.

To facilitate starting and stopping a slurry pump, it should be fitted with adequate connections so the liquid end passages can be flushed of the slurry with clear liquid. This is especially true when there are to be extended periods during which the pump will be shut down.

Rod and plunger packing requires special considerations when dealing with abrasive materials. In a piston pump, the piston runs in a renewable metal cylinder or liner. The liners are made of abrasion and corrosion resistant metals to resist wear for each specific slurry. Piston rods and plungers are also coated to resist wear.

Where the abrasion of the slurry is not great and the pressures are below 14,000 kPa (2000 psi), large-volume piston pumps are more suitable. The transportation of coal slurry falls into this class of service.

6.4 Installation, operation and maintenance

Reciprocating power pumps, when properly installed and when given good care and regular maintenance, operate satisfactorily for a long period of time. The general principles that must be considered to ensure trouble-free operation are discussed in the following sections.

Reciprocating power pumps are built in a wide variety of designs for many different services. The manufacturer's instruction book furnished with each machine should be carefully studied and followed, as there may be specific requirements of a particular machine or application which cannot be covered in a general discussion.

6.4.1 Safety

The following precautions should be taken when working on reciprocating power pumps to ensure safety of personnel:

- electrical power should be turned off. All valves to liquid end should be closed and the liquid end drained;

  CAUTION: Care should be taken to dispose of toxic or flammable liquids or vapors properly;

- the work area should be kept clear and any unnecessary items removed;
- all lifting devices should be checked for condition and capacity limits before using;
- before dismantling, assembling, or performing maintenance on the pump, the proper tools, correct parts, and manufacturer's instruction book should be available;
- all safety precautions should be followed, as directed by the safety engineer.

6.4.2 Storage

All reciprocating power pumps are tested, inspected and protected against corrosion for the period of shipment and installation only.

If the pump is not to be installed at once, the pump and parts such as packing, special wrenches, etc., should be stored in a clean, dry location, free from temperature extremes, in an approximately level position and without distortion. Coat all machined surfaces with heavy, non-corrosive oil. Inspect frequently to see that the surfaces are free of corrosion. Renew oil coating when necessary. Before putting unit into operation, clean thoroughly with a high-grade rust remover.

Where it is known that a pump will be in storage or taken out of service, such as a relocation, plant shutdown, etc., for more than three months, it is good insurance to completely fill the power and liquid ends with high-grade non-corrosive lubricating liquid. Inspect periodically for possible leakage.

6.4.3 Location of pump

Locate the pump as close to the liquid supply as possible in a clean, dry, and accessible place, so it can be inspected at regular intervals during operation. Provide ample room for maintenance.

6.4.4 Protection against seepage or flood

If it is necessary to place the pump in a pit or other low area, provision should be made to protect the pump from seepage or flood.

6.4.5 Provision for servicing space

Whether mounted on the floor, on a foundation above the floor level, or in a pit, sufficient room should be allowed for removal of plungers, pistons, rods, crankshafts, etc., and/or inspection of wearing parts as recommended in the instruction book.

6.4.6 Foundation

Small pumps can be located on the floor or supporting surface, provided that the supporting installation area is sufficiently strong to support at least 150% of the total unit weight, including the driver.

Large and medium-size pump foundations should be reinforced concrete, resting on firm soil or piling, and about 300 mm (12 inches) above the surrounding floor level. It should be entirely independent of walls or footings, building supports or floor structures. Proper concrete mixture, correct reinforcements, sufficient mass and satisfactory footing are essential to give rigid, permanent support to prevent vibration. The bearing pressures allowed on soil vary widely depending on the underlying nature of the soil, local building laws, etc. These data, available in engineering handbooks and manuals, should be carefully investigated.

6.4.6.1 Foundation bolts

Locate the foundation bolts according to the elevation drawing. Set bolts in sleeves 2 to 3 times the bolt diameter to allow for variation in pump parts, skids, or baseplates (see Figure 6.58). The sleeves should be held rigidly yet allow the bolts to be moved. Do not fill the sleeves with grout until the entire unit has been accurately aligned, supported and leveled.

The frame or baseplate when so designed should be completely filled with grout, utilizing the openings provided as filling and vent holes. Do not leave leveling pieces, shims or wedges in place.

If leveling screws are used, back off after grout has hardened. The machine must not finally rest only on leveling pieces or shims of questionable area, wedges with line contact or screws with point contact. Foundation bolts should not be tightened until the grout has fully hardened, usually about 48 hours after pouring.

6.4.7 Installation

Most pumps are aligned with the driver before leaving the factory.

It is, of course, important that the entire pump be properly supported and leveled, being careful NOT to spring the pump out of alignment when fastening it to the foundation. The erection of large pumps that are shipped in sections should be under the supervision of a competent erector, as it is necessary to carefully align and support the liquid ends with their power ends after they have been placed upon the foundation.

Alignment should be rechecked after the piping has been completely installed and connected to the pump.

6.4.7.1 Leveling the unit

Metal blocks and shims or metal wedges having a small taper should be placed close to the foundation bolts. On large units, small jacks made of cap screws and nuts are very convenient. The supports should be directly under the part carrying the greatest weight and spaced closely enough to give uniform support with minimum deflection of the unit.

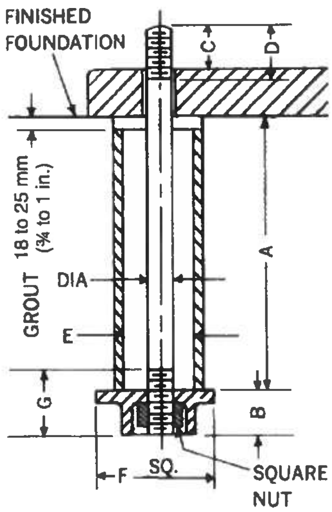
A gap of about 19 to 25 mm (¾ to 1 inch) should be allowed between the baseplate and the foundation for grouting.

Adjust the metal supports or wedges until the shafts or rods of the pump are level.

On power pumps where couplings are used, do not connect the coupling until all pump and driver alignment operations have been completed. Check the coupling faces, suction and discharge flanges of the pump for horizontal and vertical position by means of a level. Correct the position, if necessary, by adjusting the supports or wedges, as required.

6.4.7.2 Piping

Pipes must line up naturally. They must not be pulled into place with flange bolts, as this may force the pump out of alignment. To minimize strain on the pump, pipes should be supported independently of the pump.



Dia	A	B	C	D	E	F	G
Metric (millimeters)							
12	300	50	18	25	50	125	50
16	280	50	24	32	50	125	50
19	460	50	30	40	80	125	50
22	535	50	35	45	80	125	50
25	610	65	38	50	80	150	65
29	685	65	45	60	100	150	65
31	760	65	45	60	100	150	65
38	915	100	55	70	100	225	100
45	1065	100	65	90	125	225	100

Dia	A	B	C	D	E	F	G
US Units (inches)							
½	12	2	7/8	1 1/8	2	5	2
5/8	15	2	1	1½	2	5	2
¼	18	2	1¼	1¾	3	5	2
7/8	21	2	1 3/8	2	3	5	2
1	24	2½	1½	2¼	3	6	2½
1 1/8	27	2½	1 5/8	2 3/8	4	6	2½
1¼	30	2½	1¾	2 5/8	4	6	2½
1½	36	3½	2	3 1/8	4	9	3½
1¾	42	3½	2¼	3½	5	9	3½

Figure 6.58 — Foundation bolt data

6.4.7.3 Forces and moments

The most desirable arrangement is one where suction and discharge piping line up naturally with the respective pump connections. When this is done, no force or moment is exerted on the pump that could result in stresses in the pump or its foundation. Thermal expansion, when handling hot or cold liquids, must also be considered. When the piping tends to expand or contract, it exerts a force and often a twist (torque or moment) on the point of restraint, such as the pump nozzle. Pipe strains are a common cause of misalignment, hot bearings, worn couplings and vibration. Where accurate piping alignment cannot be accomplished, the pump manufacturer should be provided with the calculated forces and moments of the system for a determination of design adequacy.

Variations in flow and pressure, changes in direction of flow, cavitation, worn pistons, pump valves, etc., all contribute to piping vibration. Therefore, suction and discharge piping must be rigidly fixed in all directions and not just lightly strapped down. Flush, clean and blow out all piping before connecting to the pump. Use pipe dope and tape sparingly on male threads only.

6.4.7.4 Flanges and fittings

Flange fittings, unions and flexible connectors should be located close to the pump in all pipe lines to facilitate removal of the pump.

Alignment must be rechecked after suction and discharge piping have been bolted to the pump, to test the effect of piping strains. When handling hot or extremely cold liquids, disconnect the nozzle flanges after the unit has been in service to check the direction in which the piping expansion is acting. Correct for strain effect as required to obtain true flange alignment.

6.4.7.5 Priming

Reciprocating power pumps operating with a suction lift are not necessarily self-priming. Piston or plunger motion can only lower the pressure within the cylinder to the point where atmospheric pressure in an open-suction supply system can tend to force the liquid up the suction pipe into the cylinder.

The ratio of clearance volume to displacement has a direct effect on the priming ability of the pump. To evacuate the suction line, the pump must compress air within the cylinder and discharge some of it on each stroke. Before additional air can be taken in from the suction line, the air remaining in the clearance volume must expand to less than the suction line pressure. Thus, a clearance volume limits the priming ability of a dry pump.

Filling the clearance volume with liquid makes some slow- and moderate-speed piston pumps self-priming at a reasonable operating lift.

Some pumps are provided with priming openings where liquid can be injected directly into the piston or plunger chamber.

High-speed plunger pumps are usually designed to operate with a flooded suction or a booster pump.

6.4.7.6 Relief valve set pressure

The pump relief valve has the purpose of protecting the pump, driver, drive train and system. Select a set pressure from Table 6.10.

If an attempt is made to set a relief valve too close to the average discharge pressure, the valve will crack, and leak slightly due to pump pulsations. Leakage will quickly ruin the relief valve seat. Hence, the valve must remain closed during normal operation. It must not leak.

Always install a pressure gauge ahead of the relief valve so that it reads the true pump pressure while relieving.

The exhaust from the relief valve should always be directed to the supply tank and not to pump suction.

The line from the relief valve to the tank must be of full size. If the line is of great length, compute the pressure-drop through it. Add this line pressure loss to that within the relief valve itself when sizing the relief valve and when estimating relieving pressure. Where possible, pipe the relief valve exhaust to an open drain so that any leakage can be observed.

A relief valve is *mandatory protection* against the possibility of a mistakenly closed discharge stop valve.

6.4.7.7 Drive alignment after piping installation

After piping has been installed, the pump-prime mover drive alignment should be checked again and, if necessary, corrected.

6.4.7.8 Gaskets, pipe dope and pipe tape

The gaskets, pipe dope and pipe tape used in the system piping are exposed to the same conditions of high or low temperatures, pH values, etc., as the pump parts. Careful selection is necessary to avoid joint failure and the air and liquid leaks that follow.

6.4.7.9 Flexible coupling

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. Any improvement in alignment beyond the coupling manufacturer's minimum specification extends pump, coupling and driver service life by reducing bearing loads and wear.

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.

For applications where pumps are operated at elevated temperatures, final alignment may not be possible at operating temperature. Therefore, proper allowance should be made for the increase in pump shaft height due to thermal expansion. Recommended practice for cast iron or steel pumps is that a vertical allowance of 0.001 mm per mm (0.001 inches per inch) of pump shaft height above the base for each increase of 85°C (150°F) should be added to the height of the driver shaft.

6.4.7.10 Gear drive

Gears must have a running clearance or backlash for proper operation. This should be 0.010 mm per 1 mm (.010 inch per 1 inch) of circular pitch or 0.0314 divided by the diametrical pitch. A range of plus or minus 10% is allowable.

When the operating temperature of the gears exceeds the room temperature at which they are installed, it becomes necessary to set the gears with additional clearance to allow for expansion. This becomes appreciable for gears over 1.2 meters (48 inches) in diameter and for temperature differences over 11°C (20°F). The backlash may be measured by mounting a dial indicator so that the indicator is perpendicular to the tooth surface at the pitch line and then rocking the pinion back and forth against the gear locked in position. Another method is to pass a soft lead wire through the mesh and add together the thicknesses of the two flattened sections where it contacted the pitch line on both sides of a tooth. If the ends of the teeth are accessible, the backlash can be measured with thickness gauges.

The shafts must be level and the same distance apart on both sides of the gears. The evenness of tooth bearing can be determined by coating several pinion teeth with Prussian blue and revolving the pinion and gear together to bring the coated teeth into the mesh point. The contact pattern should show evenly along

Table 6.10 — Suggested trial set pressures of pump relief valves

Type pump	Excess of set pressure over specified pressure
Double-acting, Duplex power	25%
Double-acting, Triplex power	10%
Double-acting, Quintuplex power	10%
Single-acting, Simplex power	25%
Single-acting, Duplex power	20%
Single-acting, Triplex power	10%
Single-acting, Quintuplex power	10%
Single-acting, Septuplex power	10%

the pitch line across the face. If the ends of the teeth are accessible, the evenness of tooth bearing can be measured by the relative pressure on pieces of paper inserted at the mesh at each end of one tooth.

6.4.7.11 V-belt drive

All drives must be aligned. The driver and driven shafts must be parallel, with the V-belts at right angles to these shafts. Misalignment causes undue belt wear, or turn-over in the grooves. Alignment should be checked by placing a straightedge evenly across the rims of both sheaves. If the face of the sheaves are not of equal width, the 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. Adjust either sheave on the shaft to equalize these dimensions.

The driver should be mounted with adequate provision for belt center distance adjustment. Provide a minus adjustment to permit belt installation without stretching and a plus allowance to provide belt take-up.

Do not pry, twist, or force the belts over the sheave grooves, because it damages the belts and greatly reduces the belt life. Reduce the shaft center distance by moving the driver enough to permit fitting the belts in the proper grooves. When the belts are in place, increase the centers until proper belt tension is obtained. Adjust take-up until the belt is in accordance with the force/deflection values in Table 6.11. All of the belts must pull evenly. Belt tension should be reasonable. It is not necessary to have belts "fiddle string" tight.

During the first few days of operation, the belts will seat themselves in the grooves, and the drive must be adjusted to correct tension.

Keep belts clean and free from oil. Clean oily belts with a cloth dampened with soap and water. There should be a free circulation of air around the drive. Excessive heat reduces the life of the belts.

Never install new belts on the same drive with used belts. New belts should be installed in matched sets. Do not use sheaves with chipped or worn grooves.

For hazardous locations, static conducting belts should be used.

Consult V-belt manufacturer's tables and data for recommended V-belt cross section and belt length. When purchasing replacement V-belts, the same size and type should be ordered as furnished originally.

Slipping belts result in lowered rate of flow. Check pump speed with a tachometer. It should equal driver speed multiplied by driver sheave pitch diameter divided by pump sheave pitch diameter. Squealing or smoking belts are sometimes a clue to the slipping of belts but not always.

Establishing correct tension in single or multiple V-belt drives requires the use of a small spring scale, applied at the center of the belt span (see Figure 6.59). Apply spring pressure to produce the belt deflections shown in Table 6.11. Then read the scale and tighten or loosen as needed.

Table 6.11A — Proper spring pull tension for new and used belts (Metric)

Belt size section	Force for used belts	Force for new belts	Belt deflection						
			Belt span – meters						
	Newtons	Newtons	.50	.60	.70	.80	1.0	1.2	1.5
A	6.7–10	10–13	8 mm	10 mm	11 mm	13 mm	16 mm	19 mm	24 mm
B	14–22	22–27							
C	29–43	43–58							
D	49–73	73–98							
3V	13–33	18–44							
5V	58–102	76–133							
8V	98–129	129–156							

Before starting the pump, a suitable guard shall be placed over the belts and sheaves to protect nearby people from harm.

6.4.7.12 Lubrication

The power end of power pumps may use pressure, splash or gravity lubrication. It is very important that the manufacturer's specific instructions or name plate data be carefully followed as to choice of oil, considering the ambient conditions, the amount of oil and the frequency of oil changes. The oil must be checked periodically for contamination. If any doubt exists as to the quality of the oil, change it.

6.4.7.13 Bearings

The bearings of a pump have been carefully adjusted at the factory. It is important that if the bearings or shafts wear, they are adjusted or replaced. A new crosshead may be required, because a crosshead with excessive wear may affect plunger or piston rod alignment.

6.4.8 Plunger or piston rod packing installation

Remove old packing and thoroughly clean the stuffing box. Care should be exercised not to damage or mar the rod when removing old packing or installing new packing.

Replace rough, bent, or scored rods and rods which have a shoulder in the packing area. Replace worn throat or stuffing box bushings. Replace or recondition worn stuffing boxes.

Select proper packing size and type, depending on service (high temperature, water, chemical, etc.). Over- or under-sized packings wear out rapidly. Aluminum packings should not be used with bronze or brass rods.

Table 6.11B — Proper spring pull tension for new and used belts (US units)

Belt size section	Force for used belts	Force for new belts	Belt deflection						
			Belt span – inches						
	Pounds	Pounds	20	24	28	32	40	48	60
A	1.5–2.25	2.25–3.0							
B	3.25–5.0	5.00–6							
C	6.5–9.75	9.75–13							
D	11.0–16.5	16.5–22	5/16 in	3/8 in	7/16 in	1/2 in	5/8 in	3/4 in	15/16 in
3V	3–7.5	4–10							
5V	13–23	17–30							
8V	22–29	29–35							

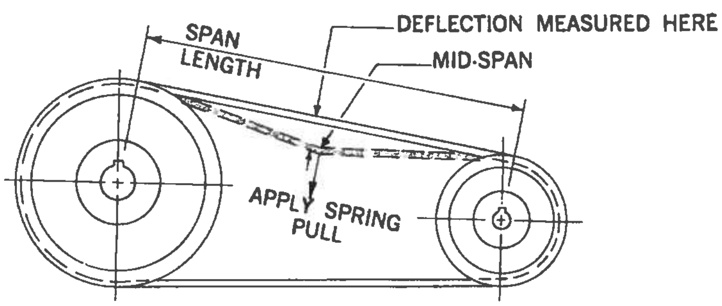


Figure 6.59 — Correct tension for V-belt drives

Packing should always be installed in the form of individual rings for proper seating of the packing. It should not be spiraled into the stuffing box.

If new coiled packing is to be cut into individual rings, use a clean surface and a sharp knife.

Packing rings may be cut with ends butted, beveled or stepped. In general, it is best to use a butt or skive joint for braided packing; a lap-bevel (skive) joint for duck and rubber packings of soft and medium grade; and either a lap, bevel, or step joint for rock-hard duck and rubber rings.

When rock-hard duck and rubber rod packing is to be installed, soak in hot water at least eight hours, if possible, to make it more flexible.

Rings should be installed singly and seated properly before additional rings are inserted. Ring joints should be staggered at 90 degrees.

A scratched rod does not always have to be replaced. A rod that may appear to be badly scored sometimes can be polished with a fine emery cloth and its appearance changed without reducing its diameter more than 0.08 mm (0.003 in). Or, a rod can be ground down to a maximum of 0.25 mm (0.010 inch) below its original diameter and may still use the original bushings (if not worn) and packing. This does not apply to thinly plated rods or high-pressure services.

The rod must operate centrally in the box with a maximum eccentricity in the centers of 0.08 mm (0.003 inch) where pressures are high 69,000 kPa (10,000 psig), and to 0.18 mm (0.007 inch) where pressures are low 3500 kPa (500 psig).

Advise the manufacturer of complete conditions of service when purchasing pump packing. The wrong packing can result in rod, plunger and bushing failure, power end problems (if pumpage reaches the crankcase) and other related problems. If a pump is being shifted to another service, check the packing selection. The original packing may not be suitable for the new application.

Reciprocating pumps cover a very wide range of pumping applications, and the styles and materials of packing differ greatly. Therefore, some pump manufacturers supply stuffing box packings, and some do not. When furnished, some packings are installed in the pump and some are packaged separately. If a graphited packing is left in contact with a rod for a long period of time with no movement of the rod, the rod surface at the packing area will become pitted. Therefore, unless the pump is going into immediate service, the installation of packing is usually left to the purchaser, unless specified "packed and ready to run" by the purchase order.

The above does not apply to piston packing. Piston packing supplied by the pump manufacturer is usually installed at the factory, except when the pump is to be placed into extended storage.

6.4.8.1 Allowance for expansion of packing

In general, no allowance for expansion is necessary for braided packings. A slight gap is advisable when woven material or duck and rubber packings are used against hot water or steam to permit expansion. Metallic channel type packings require a gap of 3 mm (1/8 inch) to allow for metallic expansion when pumping high-temperature liquids.

6.4.8.2 Gland adjustment

After installing braided rod or plunger packing rings, the gland should be evenly tightened in order to seat the rings, and then slackened off and made up snug. Then after starting the pump, it should be watched for several hours and adjusted to obtain sufficient liquid tightness, without excess pressure on the packing. Periodically adjust the gland to compensate for wear and to prevent packing movement and excessive leakage. Hot water installations should be watched for several days after application until the packing becomes stable. Packing must not be allowed to move in the stuffing box. Movement wears the stuffing box bore.

6.4.8.3 Drip

When conditions permit, a slight drip or trickle should be allowed to reduce packing friction and increase the life of packing and rod.

6.4.8.4 Molded ring packings

Lip rings, "V" and "U" rings, packing cups, etc., should be installed in accordance with the directions accompanying each set of packing. It is important that the dimensions of molded packings be accurate. Such packings are usually self-adjusting. If the stuffing box of the liquid end happens to be deeper than necessary for some types of packing, it is sometimes advisable to install a metal ring, leaving just sufficient space for the number of rings specified by the manufacturer of the packing and no more. Packings of the "U" type must be securely held by a follower. Flange or hat type packing must be tightly held by the gland between the gland and the shoulder of the stuffing box. Care must be taken to prevent damage to the lip of any of these types.

Some packing is crushable, some is not. Most adjustable packing is crushable, and so are some non-adjustable packings. It is, therefore, very important that the packing manufacturer's instruction be followed very carefully.

If packing fails frequently, check plunger or rod finish and bushing clearances. When the inside or outside diameter of bushings are worn excessively, the packing will extrude and fail. If bushing clearance is correct and failure occurs, then the packing is running dry or is the wrong selection. It is a waste of good packing to repeatedly repack worn, grooved (or bent) rods, plungers or bushings. Conversely, a new rod or plunger deserves new packing and bushings.

6.4.8.5 Lubrication of packing

Whenever possible, oil lubrication should be provided. Pressure lubrication to a lantern ring with packing above and below the ring is frequently used. Oil dripped on the rod or plunger where it enters the stuffing box is sometimes satisfactory. Water at times provides sufficient lubrication to some packings where oil contamination is a problem.

6.4.8.6 Chemical packings

When the liquid being pumped is corrosive or is a solvent, special packing materials are required to overcome the destructive action of the chemicals and to provide serviceable packing. Packings in contact with foods should be impregnated only with non-contaminating lubricants.

6.4.8.7 Piston packing

The following recommendations are general for piston packing for reciprocating pumps. Specific applications may require deviations because of installation, application, particular conditions of service, etc.

6.4.8.7.1 Hydraulic piston packing

Canvas packing (hydraulic packing) is supplied in a relatively soft packing (regular cure) or hard packing (rock-hard cure). Regular cure is available in coils or molded rings. Rock-hard is available in molded rings.

6.4.8.7.1.1 Types of hydraulic piston packing joints

Straight cut, step cut, or angle cut joints are used with this hydraulic piston packing (see Figure 6.60).

6.4.8.7.1.2 Applications for hydraulic piston packing

Regular cure piston packing is suitable for hot water service up to 82°C (180°F). At higher temperatures, 82°C to 120°C (180°F to 250°F), the rock-hard cure is recommended. On oil and gasoline service, phenolic or metallic ring packing can be used.

6.4.8.7.1.3 Soaking hydraulic piston packing

It is preferable to soak both rock-hard and regular cure packing in warm water before installation. The rock-hard should be soaked until sufficiently pliable to install. Regular cure should be soaked at least 8 hours before installing, if possible.

6.4.8.7.1.4 Fitting hydraulic piston packing

When installing regular cure packing in coil form, cut to the correct length by placing in the cylinder bore or in a form having the exact inside diameter of the cylinder

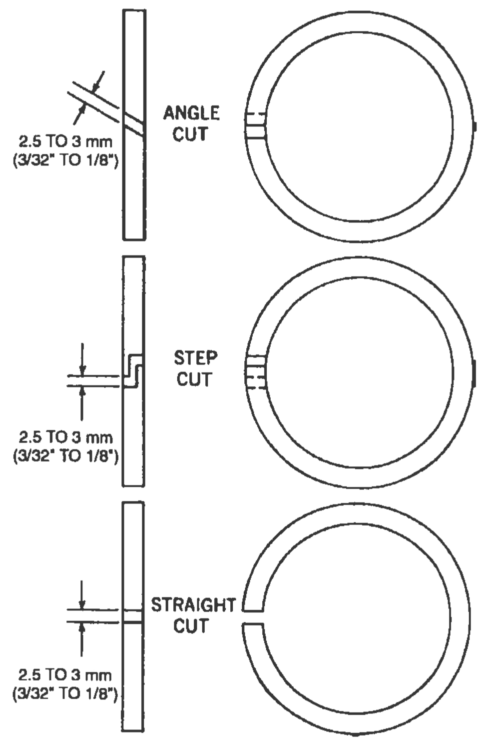


Figure 6.60 — Piston packing joints

bore. When installing rock-hard cure packing, the rings should be placed in the cylinder bore and the end clearance checked to make certain the packing will be correct after installation.

Hydraulic packing should fit the piston as indicated in Figure 6.61.

6.4.8.7.1.5 Clearance for hydraulic piston packing

The side clearance should be 1.6 mm ($^1/_{16}$ inch) when the width of the packing is no more than 75 mm (3 inches), and 3 mm ($^1/_8$ inch) for widths beyond 75 mm (3 inches). For end gaps, however, allow .75 mm ($^1/_{32}$ inch) up to 75 mm (3 inches) diameter, 1.5 mm ($^1/_{16}$ inch) from 75 to 100 mm (3 inches to 7½ inches), and 3 mm ($^1/_8$ inch) beyond the 190 mm (7½ inch) diameter.

Fitting as recommended above allows for swelling and prevents the packing from becoming too tight. If the width of the packing is greater than given in these standards, the sides of the individual rings can be peeled off until the proper width is attained.

6.4.8.7.1.6 Swelling of hydraulic piston packing

When liquid pistons are packed with hydraulic or fibrous packing, trouble may arise from the swelling of the packing, which may cause stiff operation of the pump. When such a condition exists, liners wear rapidly. It is therefore necessary to fit the packing to the liner to prevent this condition.

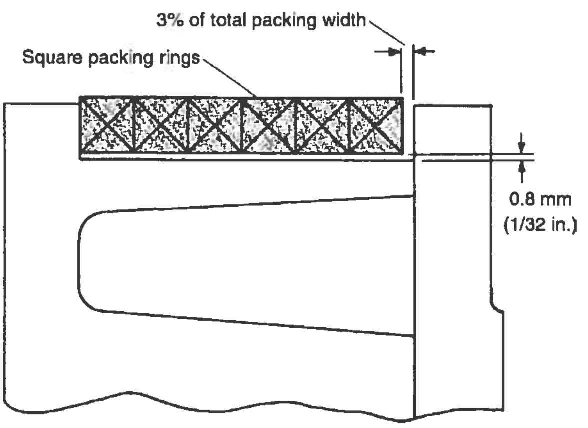


Figure 6.61 — Hydraulic packing

6.4.8.7.2 Metallic piston-ring-type packing

6.4.8.7.2.1 Application of metallic piston rings

In general, metallic bull and piston rings are used for pumping viscous liquids such as crude oil, tar, molasses, etc., and for high-temperature service; also for chemical service where lubrication of other packings is not suitable. Figure 6.62 illustrates the bull ring type of packing.

6.4.8.7.2.2 Material for metallic piston rings

The material used for this type of piston ring may be cast iron, bronze, or monel. These materials are suitable up to the maximum temperatures shown in Table 6.12.

Table 6.12 — Maximum temperature for ring materials

Ring material	Maximum temperature
Cast Iron	400°C 750°F
Bronze	230°C 450°F
Monel	400°C 750°F

Monel metal is sometimes used where corrosion or contamination is involved. Nickel alloy cast iron is also used where corrosion is a problem.

6.4.8.7.2.3 Joints for metallic piston rings

Piston rings are furnished with three types of joints: straight butt-cut, angle-cut and step-cut. See Figure

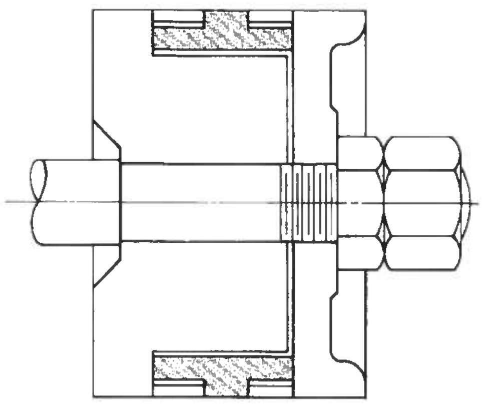


Figure 6.62 — Bull ring packing

6.60. The straight-cut joint is the usual standard. However, angle- or step-cut joints may be used. Tests show that at medium and high speeds, there is practically no difference in the amount of leakage with any of the three types of joints.

6.4.8.7.2.4 Clearance for metallic piston rings

Rings for pump pistons are furnished with an end clearance of approximately 0.30% of the diameter. With an angle-cut joint, the end clearance is made slightly less than with a lap or straight-cut joint in order to keep the circumferential clearance the same for all three types. However, straight-cut joints are preferable, especially on narrow rings. They are stronger and more easily fitted.

6.4.8.7.3 Phenolic piston ring packing

6.4.8.7.3.1 Application of phenolic piston rings

Phenolic type piston rings are used for oil, salt water, creosote, chemicals and hot liquids except chemical in concentrations in excess of those given in Table 6.13.

Table 6.13 — Maximum concentration of chemicals for phenolic type rings

Chemical	Concentration %
Soda	5
Ammonia	5
Hydrochloric acid	5
Brine	10
Caustic	5
Sulfuric acid	10

Canvas-base phenolic rings can stand temperatures up to 120°C (250°F) without deterioration.They should not be used with scored or grooved liners, as the rings will wear quickly.

6.4.8.7.3.2 Forms of phenolic piston rings

Phenolic rings are furnished as required in either one-piece form with built-in tension or in segmental form with metal expanders to provide tension.

6.4.8.7.3.3 Clearance for phenolic piston rings

The preferred cross section is approximately square. On existing designs, the cross section is determined by the groove dimensions. If one-piece rings are used on solid pistons, where the rings have to be stretched over the piston, the wall thickness should not be more than .06 mm per mm (1/16 inch per inch) of diameter.

If all rings are installed in the same space, a floating allowance, lengthwise, of .007 mm per mm (.007 inch per inch) of diameter should be made. The rings should not fit snugly around the body of the piston. The purpose of this "float" is to allow for swelling and uniform alignment against the walls of the cylinder liners.

6.4.9 Cup type pistons

6.4.9.1 Composition cup pistons

Molded composition cups are usually of laminated duck-and-synthetic, nylon-and-synthetic, or oil-resistant rubber molded to desired shapes. Generally, most cup applications fall in the range below 120°C to 150°C (250°F to 300°F), and under 690 kPa (100 psi) pressure. Some cup manufacturers list several grades for different pressure and temperature ranges.

Replacement of piston cups should include an examination of all related piston parts for excessive wear, nicks, scratches, scores, or pitting.

A coating of oil or grease at installation is good procedure.

6.4.9.2 Synthetic rubber piston cups

This type piston packing may operate continuously at higher temperatures than other types (120°C [250°F] and higher). Packing cups can be molded of synthetic rubber and of rubber compounds that resist strong acids and alkalies. They can be held to close tolerances.

Installation procedure is the same as for composition cup pistons (see Figures 6.63 and 6.64).

6.4.10 Installation

The following procedure is recommended for installation of cup pistons:

Be sure there is sufficient clearance between the inner wall of the cup and the outside diameter of the follower disc and piston head. Check to see that the cylinder liner is smooth and free from excessive wear. A scored or badly worn liner should be replaced before installing new packing cups. Clean the cylinder liner and packing cups of any dirt or grit. Lubricate the outside diameter surface of the packing cups.

Push the piston rod far enough into the cylinder to permit assembly of the piston head into the cylinder liner. Place a piston head over the piston rod tightly against the rod shoulder. Then insert the first packing cup, holding the cup horizontally until the lip has entered the liner. If the packing is too hard, soak in warm water (see Figure 6.64). When the edge of the packing cup has entered the liner, turn the cup to a vertical position with the lip facing the piston head. Lightly tap the cup into position over the hub, so it is snug against the piston head.

Slip the spacer disc over the piston rod tightly against the first cup. Then insert the second cup. When the lip of the cup clears the end of the liner, turn the cup to a vertical position with the lip facing toward the outside of the cylinder (see Figure 6.64).

Next place the follower disc over the piston rod tightly against the second cup. Apply and tighten the piston rod nut sufficiently to firmly grip the flanges of the cups between the spacer and the follower and the body of the piston. Do not, however, tighten the piston rod nut to the point where the cup becomes distorted, as illustrated in Figure 6.65.

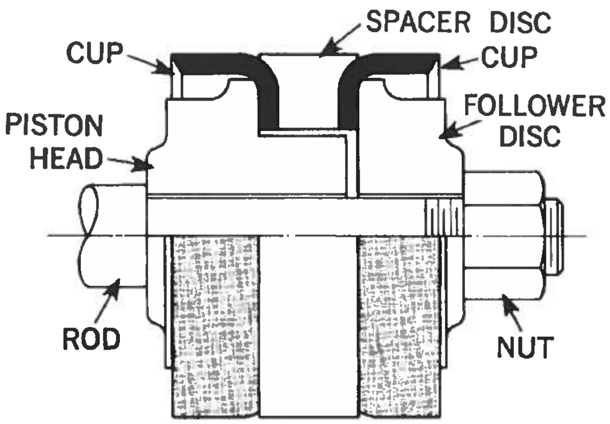


Figure 6.63 — Cup type packing

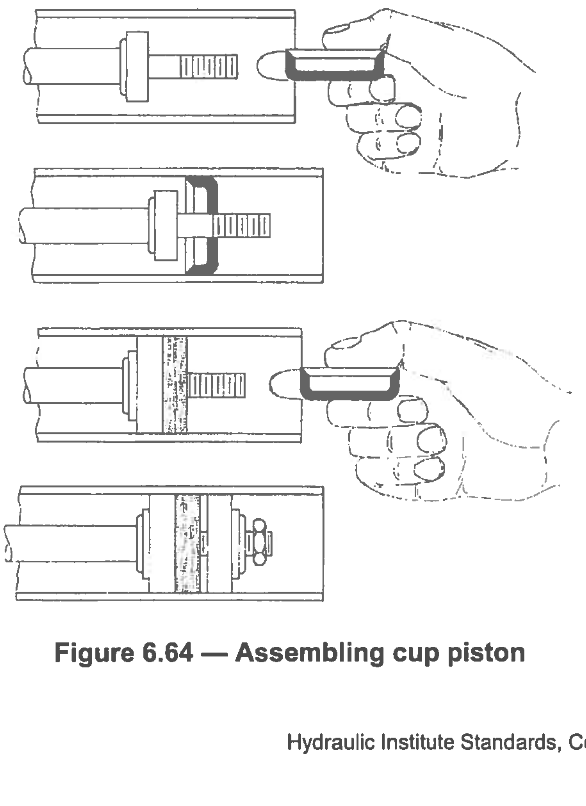


Figure 6.64 — Assembling cup piston

6.4.11 Inspection

The pump should be inspected regularly. Leaky valves should be corrected as soon as they are discovered. Most pump troubles can be traced to worn valves, packing, rods, plungers, or bushings, grooved liners, improper suction conditions, or faulty conditions outside

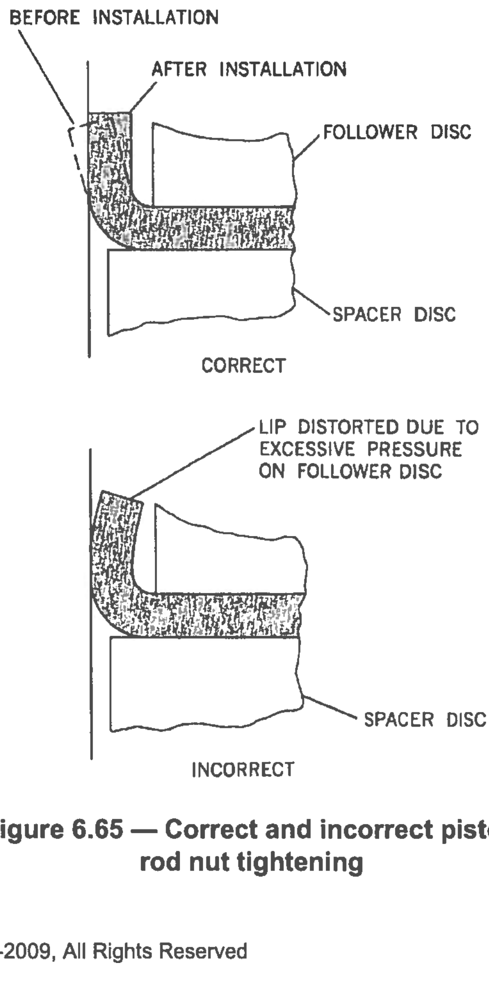


Figure 6.65 — Correct and incorrect piston rod nut tightening

the pump itself. There is NO substitute for regular, thorough preventive maintenance.

For more detailed information on proper care of the pump, see the manufacturer's operating manual.

6.4.12 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 shown in Table 6.14.

Table 6.14 — Malfunctions – cause and remedy

Malfunction	Possible cause	Remedy
Pump fails to deliver required rate of flow	Speed incorrect. Belts slipping	Change drive ratio or tighten belts (if loose). Correct motor speed
	Air leaking into pump	Seal with compounds
	Liquid cylinder valves, seats, piston packing, liner, rods or plunger worn	Reface or lap valves and seats; replace packing, liner, rods or plungers
	Insufficient NPSHA	Increase suction pressure
	Pump not filling	Prime pump. Increase suction pressure
	Makeup in suction tank less than displacement of pump	Increase makeup flow. Reduce pump speed
	Rate of flow of booster pump less than displacement of power pump	Use larger booster pump
	Vortex in supply tank	Increase liquid level in supply tank. Install vortex breaker
	One or more cylinders not pumping	Prime all cylinders. Allow pump to operate at low pressure through bypass valve to eliminate vapor
	Suction lift too great	Decrease lift. Use booster pump
	Broken valve springs	Replace
	Stuck foot valve	Clean
	Pump valve stuck open	Remove debris beneath valve
	Clogged suction strainer	Clean or remove
	Relief, bypass, pressure valves leaking	Repair
	Internal bypass in liquid cylinder	Repair or replace
Suction and/or discharge piping vibrates or pounds	Piping too small and/or too long	Increase size and decrease length. Use booster pump. Use suction and/or discharge pulsation dampeners
	Worn valves or seats	Replace or reface
	Piping inadequately supported	Improve support at proper locations

Table 6.14 — Malfunctions – cause and remedy (*continued*)

Malfunction	Possible cause	Remedy
Pump vibrates or pounds	Gas in liquid	Submerge return, supply or makeup lines in suction supply tank. If operating under a suction lift, check joints for air leaks
	Pump valve stuck open	Remove debris beneath valve
	Pump not filling	Increase suction pressure
	One or more cylinders not pumping	Prime all cylinders. Allow pump to operate at low pressure through bypass valve to eliminate vapor
	Excessive pump speed	Reduce
	Worn valves or seats	Replace or reface
	Broken valve springs	Replace
	Loose piston or rod	Tighten
	Unloader pump not in synchronism	Adjust
	Loose or worn bearings	Adjust or replace
	Worn crossheads or guides	Adjust or replace
	Loose crosshead pin or crankpin Loose pull or side rods or connecting rod cap bolts	Adjust or replace
	Pump running backwards	Correct rotation
	Water in power end crankcase	Drain. Refill with clean oil
	Worn or noisy gear	Replace
Consistent knock	Worn or loose main bearing, crankpin bearing, wrist pin bushing, plunger, valve seat, low oil level	Adjust or replace. Add oil to proper level
	Blocked discharge valve in a multiplex pump	Unblock valve opening
	NOTE: High-speed power pumps are not quiet. Checking is necessary only when the sound is erratic	
Packing failure (excessive)	Improper installation	Install per instructions
	Improper or inadequate lubrication	Lubricate per instructions
	Packing too tight	Loosen gland
	Improper packing selection	Change to correct packing
	Scored plungers or rods	Regrind or replace
	Worn or oversized stuffing box bushings	Repair or replace. Check bore and outside diameter of bushings frequently. (Many times plungers are replaced and bushings ignored. If packing can extrude through clearances, it will fail.)
	Plunger or rod misalignment	Realign. Plungers must operate concentrically in box within .08 mm (.003 inch) maximum offset for pressures up to 7000 kPa (10,000 psi) and .18 mm (.007 inch) for pressures up to 350 kPa (500 psi)

Table 6.14 — Malfunctions – cause and remedy (*continued*)

Malfunction	Possible cause	Remedy
Wear of liquid end parts	Abrasive or corrosive action of the liquid	Check valves and seats frequently at startup to determine schedule for lapping, replacing, etc. Eliminate sand, abrasives, air entering pump
	Incorrect material	Install correct materials
Liquid end cylinder failure	Air entering suction system	Eliminate air NOTE: Pitting often leads to hairline cracks which ends in cylinder failure
	Incorrect material	Install correct materials
	Flaws in casting or forging	Repair or replace
Wear of power end parts (excessive)	Poor lubrication	Replace oil as recommended in instructions. Keep oil clean and at correct temperature. Be sure oil is reaching all bearings
	Overloading	Modify pump or system to eliminate overload
	Liquid in power end	Drain power end. Eliminate cause or source of liquid entering power end. Relubricate
Excessive heat in power end (above 82°C [180°F])	Pump operating backwards	Correct rotation
	Insufficient oil in power end	Fill to proper level
	Excessive oil in power end	Drain to proper level
	Incorrect oil viscosity	Fill with correct oil
	Overloading	Reduce load
	Tight main bearings	Correct clearance
	Driver misaligned	Realign
	Belts too tight	Reduce tension
	Discharge valve of a cylinder(s) stuck open	Fix valve(s)
	Insufficient cooling	Provide adequate cooling for oil or reduce ambient temperature
	Pump speed too low	Increase speed

6.5 Reference and source material

6.5.1 NEMA-MG1-1993, Motors and Generators

NEMA-National Electrical Manufacturers Association
2101 L Street, NW, Suite 300
Washington, DC 20037

This page intentionally blank.

Appendix A

Index

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

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

Acceleration head, 25–27
Acceleration pressure, 25–27
Apparent viscosity, 27

Barometric pressure, 22, 23*t.*
Bull ring packing, 63, 63*f.*

Critical carrying velocity, 27
Cup type pistons
  composition cups, 64, 65*f.*
  installation, 64–65
  synthetic rubber cups, 64, 65*f.*

D *See* Displacement
Definitions, 20–28
Discharge piping, 45, 46*f.*
Displacement, 20
Double-acting pump, 1, 2*f.*, 3*f.*
Duplex pump, 2

Effective particle diameter, 27
Efficiency, 23
Elevation head, 22
Elevation pressure, 22
η *See* Efficiency
η_p *See* Pump efficiency
η_p *See* Pump mechanical efficiency
η_v *See* Volumetric efficiency

Flooded suction, 25
Flow rate, 20
Foundation, 55
  bolts, 56, 56*f.*
Friction characteristic, 27
Friction loss pressure, 23

Gauge pressure, 22

h_{acc} *See* Acceleration head
Heterogeneous mixture, 27
Homogeneous flow, 27
Homogeneous mixture, 27
Horizontal pump, 1*f.*, 1

Hydrocarbon physical properties, 50*t.*, 51*t.*

Inlet system, 38–40
  booster pumps, 40
  connection of piping sections, 39*f.*
  foot valve, 40
  high points in piping system, 39
  inlet line valve, 40
  inlet piping, 40
  inlet piping diameters, 39, 40*f.*
  inlet pressure gauge, 40
  liquid source features, 38
  multiple-pump installations, 39
  pulsation dampener, 41
  screens or strainers, 40, 40*f.*
  suction system relationships, 41, 42*f.*, 43*f.*
  suction tanks, 38*f.*, 41
Inspection, 65–66
Installation, 56
  bearings, 60
  drive alignment, 57
  flanges and fittings, 57
  flexible coupling, 58
  forces and moments, 57
  gaskets, 58
  gear drive, 58
  leveling the unit, 56
  lubrication, 60
  pipe dope and tape, 58
  piping, 56
  piston rod packing, 60–64
  priming, 57
  relief valve set pressure, 57, 58*t.*
  V-belt drive, 59, 59*t.*, 60*t.*, 60*f.*

l_s *See* Static suction lift
L *See* Stroke
Liquid bypass, 35, 36*f.*
Liquid end
  cylinder liner, 5, 11*f.*
  gland, 7, 7*f.*
  lantern ring (seal cage), 7, 7*f.*
  liquid cylinder, 5, 5*f.*

Liquid end (continued)
manifolds, 5, 5*f.*
packing, 7, 7*f.*
parts, 5–8, 9*f.*, 10*f.*, 11*f.*, 12*t.*
piston, 5, 6*f.*
plunger, 3*f.*, 6, 7*f.*
stuffing box, 7, 7*f.*
upper crosshead, 8, 8*f.*
valve assembly, 8, 8*f.*
valve chest cover, 5, 11*f.*
valve plate (check valve), 5, 11*f.*
Liquid expansion factor, 50, 51*f.*
Locked-rotor torque ratings, 38, 39*t.*

Malfunction causes and remedies, 66, 66*t.*–68*t.*
Multiplex pump, 2

n *See* Speed
Net positive inlet pressure available, 25
Net positive inlet pressure required, 25
Net positive suction head available, 25
Net positive suction head required, 25
Non-homogeneous flow, 27
Non-settling slurry, 27
NPIPA *See* Net positive inlet pressure available
NPIPR *See* Net positive inlet pressure required
NPSHA *See also* Net positive suction head available
NPSHR *See* Net positive suction head required

P *See* Power
p_{acc} *See* Acceleration pressure
p_b *See* Barometric pressure
p_d *See* Total discharge pressure
p_f *See* Friction loss pressure
p_g *See* Gauge pressure
p_H *See* Total differential pressure
P_p *See* Pump input power
p_s *See* Total suction pressure
p_v *See* Velocity pressure
P_w *See* Pump output power
p_z *See* Elevation pressure
Percent solids by volume, 27
Percent solids by weight, 27
Piston pumps, 1, 2*f.*
cup type pistons, 64
typical service, 53–54
Piston rod load, 23
Piston rod packing installation, 60
allowance for expansion of packing, 61
chemical packings, 62
drip, 61
gland adjustment, 61
hydraulic piston packing, 62–63
lubrication of packing, 62
metallic piston-ring-type packing, 63–64
molded ring packings, 61
phenolic piston ring packing, 64, 64*t.*
piston packing, 62–64
Plunger load, 23
Plunger or piston speed, 20
Plunger packing installation
See also Piston rod packing installation
Plunger pumps, 1*f.*, 1, 2*f.*, 3*f.*
typical service, 53–54
Power, 23
Power end
connecting rod, 13, 14*f.*
crankpin bearing, 13, 14*f.*
crankshaft, 13, 13*f.*
crosshead extension (plunger extension), 14, 15*f.*
frame extension, 14, 15*f.*
main bearing, 13, 13*f.*, 14*f.*
parts, 13–14, 15*f.*–18*f.*, 19*t.*
power crosshead, 13, 14*f.*
power frame, 13, 13*f.*
wrist pin, 14, 15*f.*
wrist pin bearing, 14, 15*f.*
Pressure, 20–23
Pump efficiency, 23
Pump input power, 23
Pump mechanical efficiency, 23
Pump output power, 23
Pump torque, 23
characteristics, 34
requirements, 35

Q *See* Rate of flow

Rate of flow, 20
Reciprocating power pumps, 1
cup type pistons, 64
discharge piping, 45
foundation, 55
foundation bolts, 56, 56*f.*
inlet system, 38–45
inspection, 65–66
installation, 56–60
liquid end, 5–8, 9*f.*, 10*f.*, 11*f.*, 12*t.*
location, 55
malfunctions, cause and remedies, 66*t.*–68*t.*
power end, 13–14, 15*f.*–18*f.*, 19*t.*
pre-installation considerations, 55–56
protection against seepage or flood, 55
right and left hand shaft extension, 2–5
servicing space, 55
speeds, 29–34
starting, 34–38
storage, 55
types and nomenclature, 1
typical services, 29

Reciprocating power types, 1*f.*
Relief valves, 45

S *See* Slip
Safety, 55
Saltation, 27
Settling slurry, 28
Settling velocity, 28
Simplex pump, 2*f.*, 2, 3*f.*
Single-acting pump, 1*f.*, 1, 2*f.*
Slip, 20
Slurry, 27
Soft start drivers, 37
Speed, 20
Speeds
  and application details, 33
  basic speed ratings and formulas, 29–33
  factors affecting operating speed, 33
  high, 34
  and liquid characteristics, 33
  medium, 33
  and pump design, 33
  slow, 34
  and type of duty, 33
Starting, 34
  with liquid bypass, 35, 36*f.*
  without liquid bypass, 35
  pump torque characteristics, 34
  pump torque requirements, 35
  soft start drivers, 37
Static suction lift, 25
Stroke, 20
Submerged suction, 24
Subscripts, 22*t.*
Suction conditions, 24
Suction system relationships, 41, 42*f.*, 43*f.*
Symbols, 21*t.*

Total differential pressure, 22
Total discharge pressure, 20
Total suction lift, 25
Total suction pressure, 20
Troubleshooting
  See Malfunctions, causes and remedies

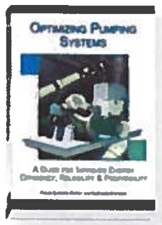
v *See* Plunger or piston speed
Valve seat area, 24, 24*f.*, 25*f.*
Velocity pressure, 22
Vertical pumps, 1, 2*f.*
Volumetric efficiency, 23
  calculating for hydrocarbons, 47–53
  calculating for water, 45–47, 48*t.*, 49*t.*
  water compressibility, 47, 48*t.*, 49*t.*

Water compressibility, 47, 48*t.*, 49*t.*

Z *See* Elevation head

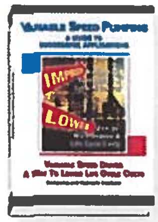
Hydraulic Institute
Creating Pump Standards Since 1917

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

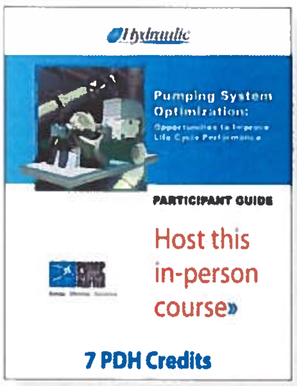
Individual Standards

- Hardcopy
- Downloadable

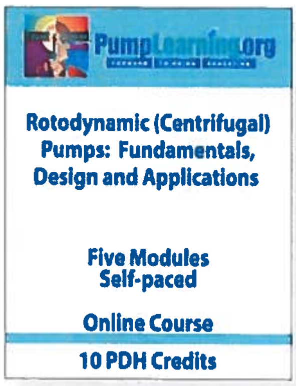
Complete Set — Hardcopy

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

This page intentionally blank.

M113