ANSI/HI 8.1–8.5-2000

Approved American National Standard

ANSI

American National Standard for

Direct Acting (Steam) Pumps

for Nomenclature, Definitions, Application and Operation

ANSI/HI 8.1–8.5-2000

Hydraulic Institute

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

This page intentionally blank.

ANSI/HI 8.1–8.5-2000

American National Standard for

Direct Acting (Steam) Pumps

for Nomenclature, Definitions, Application and Operation

Secretariat
Hydraulic Institute
www.Pumps.org

Approved November 2, 1999
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CAUTION NOTICE: This American National Standard may be revised or withdrawn at any time.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 1-880952-42-4

Contents

Page

Figures

Tables

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 Sytems, 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

8 Direct acting (steam) pumps

8.0 Scope

This Standard applies to direct acting (steam) pumps. It includes types and nomenclature; definitions; design and application; and installation, operation and maintenance.

8.1 Types and nomenclature

8.1.1 Introduction

A direct acting (steam) pump is a reciprocating (steam) engine and a liquid end built integrally together as a unit. Although steam is implied as the driving medium, compressed gases such as air or natural gas can be used.

8.1.2 Types

Figure 8.1 shows the relationship between the following pump types.

8.1.2.1 Horizontal pump

The axial centerline of the cylinder is horizonal.

8.1.2.2 Vertical pump

The axial centerline of the cylinder is vertical.

8.1.2.3 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 cycle (see Figures 8.2 and 8.5).

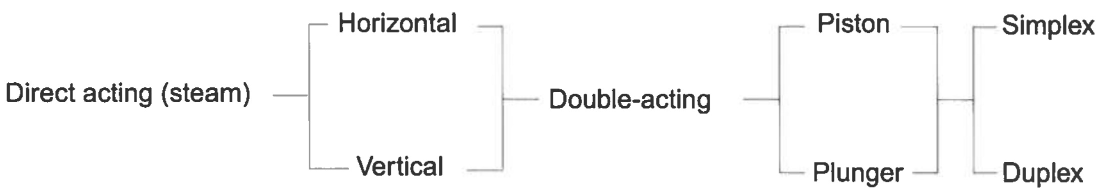


Figure 8.1 — Types of direct acting (steam) pumps

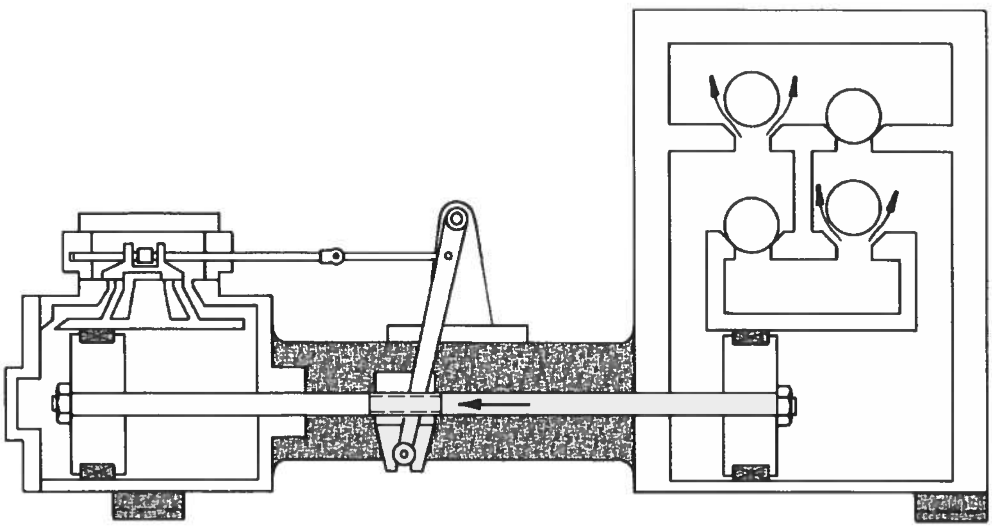


Figure 8.2 — Direct acting, horizontal, double piston, (steam) pump. Drive (steam) end is on left and liquid end is on right.

8.1.2.4 Piston pump

The liquid end contains pistons (see Figure 8.2).

8.1.2.5 Plunger pump

The liquid end contains plungers (see Figure 8.5).

8.1.2.6 Simplex pump

Contains one piston or one pair of opposed plungers driven by one steam cyclinder (see Figure 8.3).

8.1.2.7 Duplex pump

Contains two pistons or two pair of opposed plungers driven by two (steam) cyclinders (see Figure 8.4).

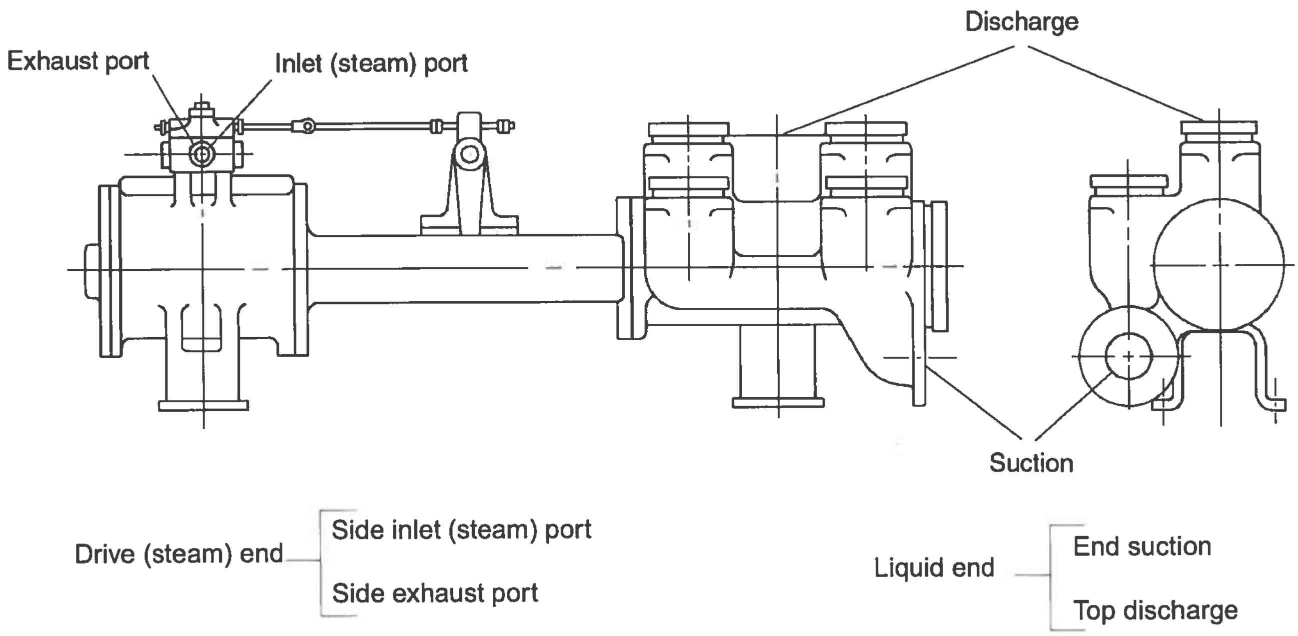


Figure 8.3 — Horizontal simplex direct acting (steam) pump

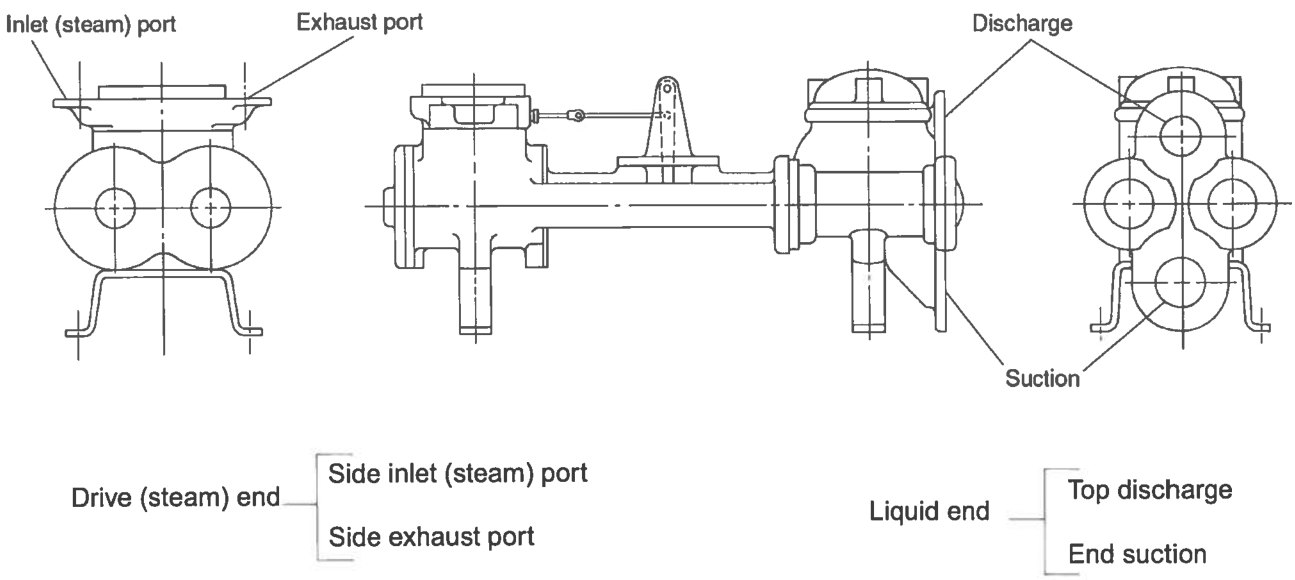


Figure 8.4 — Horizontal duplex direct acting (steam) pump

8.1.3 Nomenclature

8.1.3.1 Purpose

The nomenclature and definitions in these Standards were prepared to provide a means for identifying the various pump components covered by 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 direct acting (steam) pump components but do not necessarily represent recommended designs. Variations in design may exist without violating the intent of these standards.

8.1.3.2 Right and left hand designations

To determine the relative positions of components or hand of pump, proceed as follows:

For horizontal pumps, stand at the drive (steam) end and face the liquid end. For vertical pumps, stand facing the driver (steam) valve chest.

When standing as indicated above, the side on the left is the left hand (LH), and the side on the right is the right hand (RH) side.

8.1.3.3 Liquid end

The liquid end of a direct acting (steam) pump is the same as the liquid end of a simplex or duplex double-acting power pump. Therefore, the table of liquid end parts, the drawings of the piston pump liquid ends and the discussion of valve seat area are also applicable to direct acting (steam) pumps.

The following liquid ends are used on both direct acting (steam) and reciprocating power pumps:

8.1.3.3.1 Piston type

The liquid end contains pistons as shown in Figure 8.2.

8.1.3.3.2 Plunger type

The liquid end contains plungers as shown in Figure 8.5. Since most direct acting (steam) pumps are double-acting, and since the conventional plunger is single-acting, two plungers and two individual pumping chambers are required for a simplex plunger pump (four of each for a duplex), arranged on a common axis as shown in Figure 8.5.

8.1.3.3.3 Valve plate type

Has removable suction and/or discharge valve deck(s) (valve plates). The drawing shows a valve plate pump with a removable discharge valve deck (see Figure 8.6).

8.1.3.3.4 Valve pot type

Has a separate cover over each valve chamber. A side valve pot type has the suction valve pots at the side of the pump (see Figure 8.7).

8.1.4 Drive (steam) end

The drive (steam) end is that portion of a direct acting (steam) pump which distinguishes it from a reciprocating power pump. It provides the force to drive the piston or plungers in the liquid end. The drive end includes the drive cylinder, drive piston(s), rods,

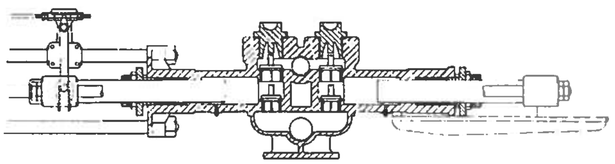


Figure 8.5 — Double-acting plunger type liquid end

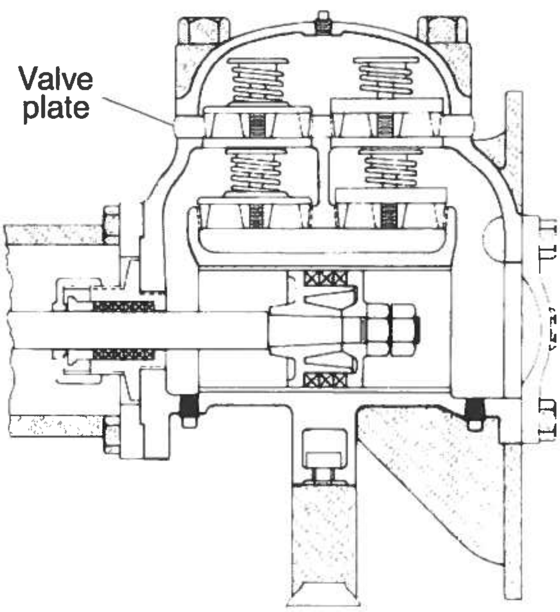


Figure 8.6 — Valve plate type liquid end

valves, valve-actuating mechanism (valve gear), and various additional parts required to supplement the functions of the above.

Figure 8.8 shows a cross section and an exploded view of a typical horizontal duplex drive (steam) end.

8.1.4.1 Drive (steam) cylinder

This is the major component of the drive end and supports most other drive end parts. It forms the chambers which retain the gas, air or steam.

8.1.4.2 Drive (steam) piston

The drive medium (steam, gas or air) acts upon the drive piston, moving it in the desired direction. The piston rod fastened to the drive piston transmits this motion to the liquid end.

8.1.4.3 Main drive (steam) slide valve

The main drive slide valve arrangement is such that it directs high-pressure steam or gas to one end of the cylinder, simultaneously exhausting the spent steam or gas from the opposite end.

8.1.4.4 Valve gear (actuating mechanism)

The valve gear moves the main drive valve at the end of each stroke of the pump so that the drive piston reverses its direction of travel.

8.1.4.5 Auxiliary drive (steam) valve

A simplex pump is equipped with an auxiliary drive valve which is actuated by the valve gear and with steam, gas or air actuates the main drive valve. This construction is necessary to allow a simplex pump to operate at low speeds without stalling.

8.1.4.6 Types of main drive (steam) valves

The following types of steam valves are used:

8.1.4.6.1 D slide valves

D slide valves are flat valves having the general form of the letter "D" (see Figure 8.9) lying on its back. This type main drive (steam) valve is recommended for drive (steam) end operating temperatures up to 260°C (500°F).

8.1.4.6.2 Piston valves

Piston valves are circular valves fitted with self-adjusting piston rings (see Figure 8.10). This main drive (steam) valve design is recommended for drive (steam) end operating temperatures above 260°C (500°F).

8.1.4.7 Types of valve gear adjustments

The following types of valve gear adjustments are used (Figures 8.11, 8.12, and 8.13, on page 6):

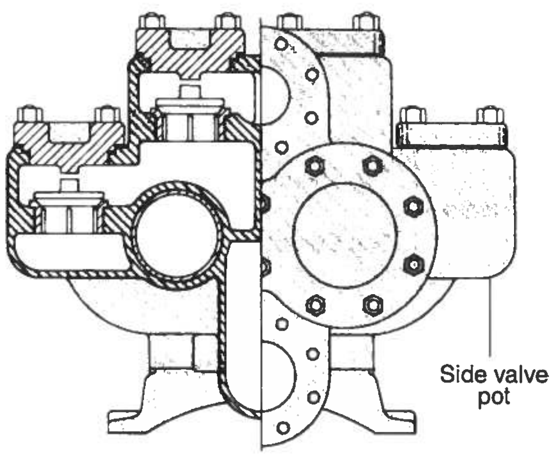
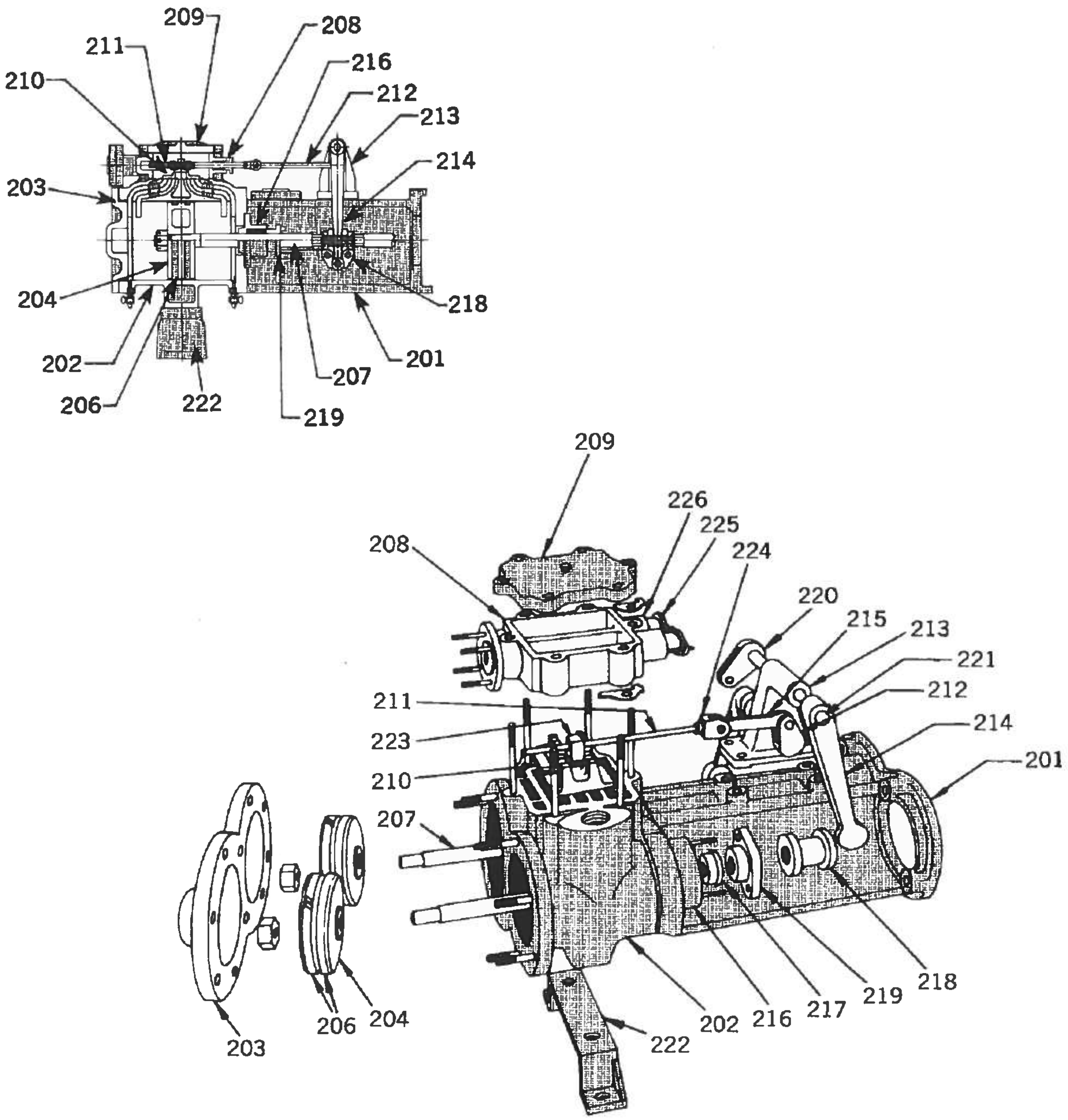


Figure 8.7 — Side valve pot type liquid end (end view, duplex pump)



201 Cradle or yoke
202 Cylinder, drive
203 Head, drive cylinder
204 Piston, drive
206 Ring, piston drive
207 Rod, drive piston
208 Chest, drive
209 Cover, valve chest, drive
210 Valve, main drive
211 Rod, valve
212 Link, valve rod
213 Bracket, rockershaft
214 Arm, rocker, long
215 Arm, rocker, short
216 Box, stuffing, drive rod
217 Bushing, throat
218 Crosshead, drive
219 Gland, stuffing box, drive end
220 Rockershaft, long
221 Rockershaft, short
222 Foot, drive cylinder
223 Nut, rod adjusting
224 Fork, valve rod
225 Gland, stuffing box, valve rod
226 Box, stuffing valve rod

Figure 8.8 — Drive (steam) end – horizontal duplex pump

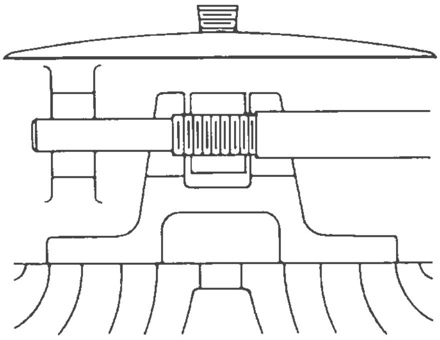


Figure 8.9 — D slide valve

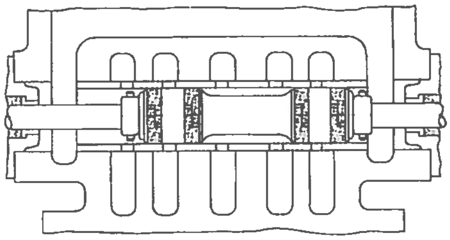


Figure 8.10 — Piston valve

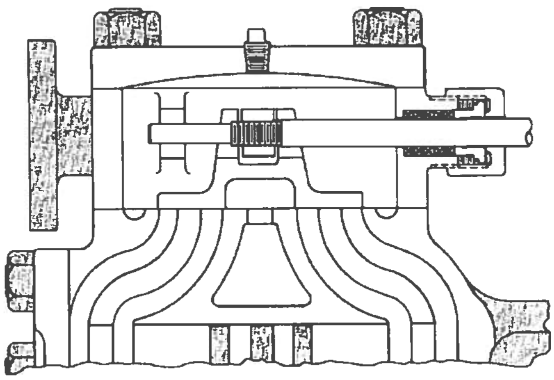


Figure 8.11 — Inside-fixed lost-motion valve gear

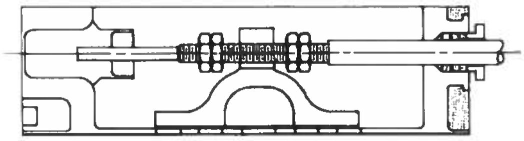


Figure 8.12 — Inside-adjustable lost-motion valve gear

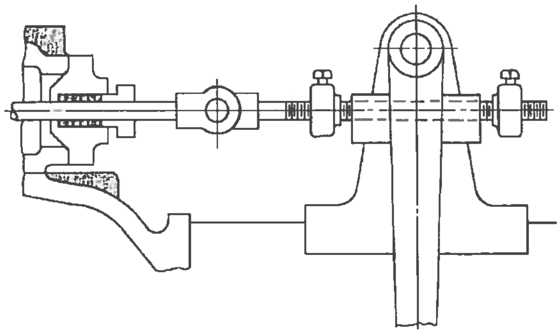


Figure 8.13 — Outside-adjustable lost-motion valve

8.2 Definitions

The purpose of this section is to define terms used in pump applications. Symbols, terms, and units are shown in Table 8.1 and subscripts in Table 8.2.

8.2.1 Stroke (L)

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

8.2.1.1 Pump rate of flow (capacity) (Q)

The rate of flow of a reciprocating pump is the total volume throughput per unit of time at suction conditions. It includes both liquid and any dissolved or entrained gases at the stated operating conditions.

8.2.1.2 Speed (n)

The number of complete cycles of the pump shaft in a given unit of time. Speed is expressed as cycles per minute.

8.2.1.3 Pump displacement (D)

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

For single-acting pumps:

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

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

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

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

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

Where:

D = Displacement, m^3/h (gpm);

A = Plunger or piston area, mm^2 (square inch);

a = Piston rod cross-sectional area, mm^2 (square inch) (double-acting pumps);

L = Stroke length, mm (inch);

n = Speed in cycles per minute;

M = Number of pistons or plungers.

8.2.1.4 Slip (S)

Slip of a reciprocating pump is the loss of rate of flow, expressed as a 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 fluid compressibility or leaks from the liquid end.

8.2.1.5 Plunger or piston velocity (v)

The plunger or piston velocity is the average velocity of the plunger or piston.

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

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

8.2.2 Pressures (p)

The standard unit of pressure is the kilopascal (pound force per square inch).

8.2.2.1 Discharge pressure (p_d)

The liquid pressure at the centerline of the pump discharge port.

8.2.2.2 Suction pressure (p_s)

The liquid pressure at the centerline of the pump suction port.

8.2.2.3 Total differential pressure (p_H)

The difference between the liquid discharge pressure and suction pressure.

$$p_H = p_d - p_s$$

Table 8.1 — 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
D	Displacement	cubic meters per hour	m^3/h	US gallons/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	cycles/minute	cpm	cycles/minute	cpm	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.

8.2.2.4 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 determined when there is a 3% reduction in rate of flow at constant differential pressure and speed.

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.

8.2.2.5 Gauge pressure (p_g)

The pressure energy of the liquid as measured by a pressure gauge relative to the atmosphere.

8.2.2.6 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.

8.2.2.7 Elevation head (Z)

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

8.2.2.8 Velocity pressure (p_v)

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

8.2.2.9 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 kPa (14.7 psia) and declines with increasing altitude as shown in Table 8.3.

Table 8.2 — Subscripts

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

Table 8.3 — Barometric pressure versus elevation

Altitude		Average barometric pressure	
Meters	Feet	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

8.2.3 Suction conditions

8.2.3.1 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 rate of flow. This will occur whenever friction head exceeds the static suction head (submergence) of the pump.

8.2.3.2 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 situation.

8.2.3.3 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 usually expressed millimeters (inches) of mercury.

8.2.3.4 Net positive suction head available (NPSHA)

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 kPa (psi).

8.2.3.5 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.

8.2.4 Pump efficiency (η_p)

For a direct acting (steam) pump, pump efficiency (η_p) (also called "mechanical efficiency") is defined as the ratio of the force exerted on the liquid by the liquid piston (or plunger) to the force exerted on the drive (steam) piston by the steam or gas, neglecting all fluid friction losses through porting and valves.

$$\eta_p = \frac{A_L p_H}{A_{st} \Delta p_{st}}$$

Where:

A_L = liquid piston area;

p_H = liquid total differential pressure (discharge pressure minus suction pressure);

A_{st} = drive (steam) piston area;

Δp_{st} = drive medium (steam, gas or air) total differential pressure (inlet pressure minus exhaust pressure).

8.3 Design and application

8.3.1 Purpose

This section is to provide general guidelines for the application of direct acting (steam) pumps.

8.3.2 Typical services

Direct acting (steam) pumps should be considered whenever a motivating medium (steam or other gas) at sufficient pressure is economically available or wherever other means of driving the pump, such as electric motors or combustion engines, are deemed hazardous. They are particularly suitable when steam at boiler pressure must be "let down" to lower pressure for heating purposes, since the direct acting (steam) pump acting as the pressure-reducing device provides pumping service for negligible energy cost.

8.3.3 Pump speeds

Basic speeds of direct acting (steam) pumps for general service on cool liquids not exceeding 55 centistokes (250 SSU) viscosity are as follows:

Stroke length		Basic speed			
mm	Inches	Meters/ minute	Feet/ minute	Cycles/ minute	Strokes/ minute
75	3	11	37	74	148
100	4	14	47	70	140
125	5	16	53	64	128
180	6	18	58	58	116
200	8	20	67	50	100
250	10	23	74	44	88
300	12	25	81	41	82
350	14	27	87	37	74
400	16	28	92	34	68
450	18	29	97	32	64
500	20	30	100	30	60
550	24	32	106	26	52

Speeds of direct acting (steam) pumps handling viscous liquids, as a percentage of the basic speeds, are as follows:

Viscosity		Correction
Centistokes	SSU	% of basic speed
215	1000	90
430	2000	80
865	4000	70
1300	6000	62
1725	8000	55
2160	10,000	50

For an intermediate viscosity, interpolate between above values.

8.3.4 Acceleration head (h_{acc}) — acceleration pressure (p_{acc})

Acceleration head or pressure is a system phenomenon associated with both direct acting pumps and reciprocating power pumps, due to the acceleration and deceleration of the liquid in the suction piping of these types of pumps.

Acceleration head or pressure is often thought of as being a loss, and it is treated as such when calculating NPSHA; but the pressure drop caused by the acceleration is offset by the increase in pressure when the liquid decelerates. Therefore, the average pressure in the suction line is calculated without consideration of acceleration head.

Total suction lift, as defined in Section 8.2.3.5, represents the average without reference to the fluctuations above and below this average due to the inertia effect of the liquid mass in the suction line. With higher speed of the pump or with relatively long suction lines, this pressure fluctuation or acceleration head must be taken into account if the pump is to fill properly without cavitation, pounding or vibration of the suction line.

The low speeds of direct acting pumps normally keep acceleration head low enough for satisfactory operation. However, it is desirable to perform an acceleration head calculation to ensure proper pump operation is obtained.

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

$$\text{(metric)}\ h_{acc} = \frac{lvnC}{Kg} \text{ or } p_{acc} = \frac{lvnCs}{.102\,kg}$$

$$\text{(US units)}\ h_{acc} = \frac{lvnC}{Kg} \text{ or } p_{acc} = \frac{lvnCs}{2.31kg}$$

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 cpm;

C = 0.200 for simplex double-acting;

C = 0.060 for duplex double-acting;

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, 9.8 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 charged and 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.

There is a similar pressure fluctuation on the discharge side of every direct acting 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 of 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.

8.4 Installation, operation and maintenance

Direct acting (steam) pumps, when properly installed and when given good care and regular maintenance, will operate satisfactorily for a long period of time. The following paragraphs discuss the general principles that must be considered to ensure trouble-free operation.

Direct acting (steam) 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.

8.4.1 Safety

The following precautions should be taken when working on a direct acting pump to ensure safety of personnel:

- motivating steam or gas should be turned off. All valves to the liquid end should be closed and the liquid end drained;

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

- 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.

8.4.2 Storage

All direct acting pumps are 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 fill the drive and liquid ends completely with high-grade non-corrosive liquid. Inspect periodically for possible leakage.

8.4.3 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.

8.4.4 Protection of pump 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.

8.4.5 Provision for servicing space

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

8.4.5.1 Flanges and fittings

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

8.4.5.2 Priming

Direct acting 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.

Therefore 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.

8.4.5.3 Relief valve

A relief valve is required in the discharge line before any other valve if the stalling pressure exceeds the liquid cylinder design pressure. Stalling pressure is based on full inlet drive pressure plus suction pressure.

The pump relief valve has the purpose of protecting the pump and system. Select a set pressure 10% over specified pressure for duplex pumps and 20% over specified pressure for simplex pumps.

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

Always install a pressure gauge upstream 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 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.

8.4.6 Hydraulic action — direct acting (steam) piston pump

If the pump suction valves and seats seal tightly, the discharge piping isolation valve is closed tightly, no relief valve is used between the piping isolation valve and the pump, and the liquid freely bypasses the liquid piston packing, the piston rod can enter the cylinder like the plunger of a single plunger pump. Then drive inlet (steam) pressure acting on the full drive piston area must be balanced by liquid pressure acting on the liquid piston rod area only. This hydraulic action may develop sufficient pressure to burst the liquid cylinder.

When stopping a direct acting (steam) pump, close the drive end (steam) inlet valve down hard. Open the drain cocks or plugs in the drive (steam) cylinder. They must be opened anyway when starting to drain the condensate from the drive (steam) cylinder.

Do not close the discharge piping isolation valve unless the pump is to be shut down and opened for inspection or repairs.

8.4.7 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.

8.4.8 Drive (steam) end lubrication

The drive (steam) cylinder is the driving unit of any direct acting (steam) pump. Poor efficiency here is directly transmitted to the overall operation of the system into which the pump is operating. Due to the constant washing action of the drive media (steam, gas or air), the drive (steam) cylinder requires a good grade of oil. It is very important that the manufacturer's specific instructions be carefully followed as to choice of oil used, with consideration given to the ambient conditions, driving media temperature and the amount of oil to be used. Drive (steam) cylinders are usually lubricated by a force-feed lubricator.

8.4.9 Plunger or piston rod packing installation

Remove old packing and thoroughly clean 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.

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 joining 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 as follows to prevent or minimize the leakage path between the pumpage inside the pump and the atmosphere:

Number of packing rings	Approximate degrees of joint stagger
3	120°
4	90°
5	75°
6	60°
7	53°
8	45°

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 .08 mm (.003 inch). Or, a rod can be ground down to a maximum of .25 mm (.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 piston rod or plunger must operate centrally in the box with a maximum eccentricity in the centers of .08 mm (.003 inch) where pressures are high (70,000 kPa [10,000 psig]) and to 0.18 mm (.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 or plunger and bushing failure 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. Some packings, when furnished, are installed in the pumps; 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 may 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.

8.4.9.1 Allowance for expansion of packing

In general, no allowance for expansion is necessary for braided packings. A slight gap is advisable when 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.

8.4.9.2 Gland adjustment

After installing braided rod or plunger packing rings, the gland should be evenly tightened to seat the rings, and then slackened off and made 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.

8.4.9.3 Piston rod (plunger) packing drip

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

It should be noted that, when toxic or volatile liquids are being pumped, packing drip (or leakage) of these products to the atmosphere is not always wanted or permitted due to environmental regulations. In such situations, special stuffing box designs are available from some pump manufacturers to control product leakage to the atmosphere.

8.4.9.4 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 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 the plunger or rod finish and bushing clearances. When the inside or outside diameter of bushings are worn excessively, the packing extrudes and fails. 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 repack worn, grooved (or bent) rods, plungers or bushings repeatedly. Conversely, a new rod or plunger deserves new packing and bushings.

8.4.9.5 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. In some cases, oil dripped on the rod or plunger where it enters the stuffing box is satisfactory. Water at times provides sufficient lubrication to some packings where oil contamination is a problem.

Since lubricant migration into the product will occur, the type of packing lubricant used is determined by the pump users. They need to select a lubricant that provides good lubricating properties and is compatible with their product and system.

8.4.9.6 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.

8.4.9.7 Piston packing basis of recommendations

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.

8.4.9.8 Forms of 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.

8.4.9.9 Types of hydraulic piston packing joints

Straight-cut, step-cut, or angle-cut joints are used with this type of packing (see Figure 8.14).

8.4.9.10 Applications for hydraulic piston packing

Regular cure piston packing is suitable for hot water service up to 80°C (180°F). At higher temperatures, 80°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.

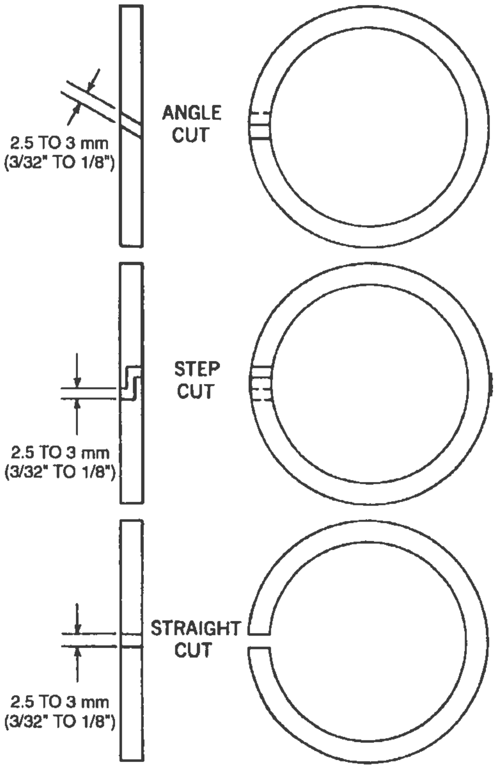


Figure 8.14 — Piston packing joints

8.4.9.11 Soaking 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.

8.4.9.12 Fitting 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 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 8.15.

8.4.9.13 Packing clearance

Packing clearance should be allowed as follows:

Packing width		Side clearance	
Up to 75 mm	Up to 3 inch	1.5 mm	1/16 inch
Over 75 mm	Over 3 inch	3.0 mm	1/8 inch

Packing diameter		End gap	
Up to 75 mm	Up to 3 inch	0.8 mm	1/32 inch
75 to 190 mm	3 to 7.5 inch	1.5 mm	1/16 inch
Over 190 mm	Over 7.5 inch	3.0 mm	1/8 inch

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.

8.4.9.14 Swelling of 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. This is particularly true with direct-acting steam pumps. When such a condition exists, liners wear rapidly. It is therefore necessary to fit the packing to the liner to prevent this condition.

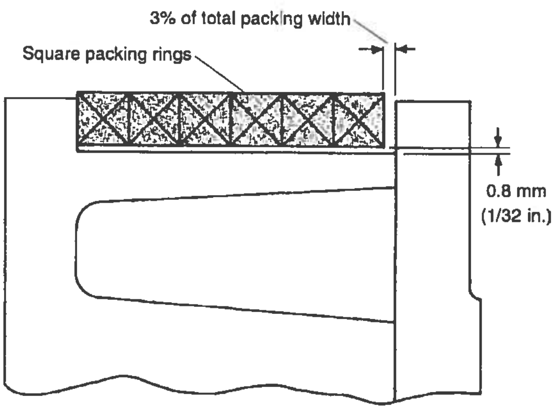


Figure 8.15 — Hydraulic packing

8.4.9.15 Metallic-type piston packing

8.4.9.15.1 Application for metallic packing

In general, metallic liquid piston rings are used for pumping viscous liquids such as crude oil, tar, molasses, etc., for high-temperature service, and also for chemical service where lubrication of other packings is not suitable. Figure 8.16 illustrates three designs of metallic liquid piston ring packing.

8.4.9.15.2 Material for metallic packing

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 8.4.

Table 8.4 — 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 if corrosion or contamination is involved. Nickel alloy cast iron is also used where corrosion is a problem.

8.4.9.15.3 Joints for metallic packing

Piston rings are furnished with three types of joints: straight butt-cut, angle-cut, or step-cut (see Figure 8.14). 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.

8.4.9.15.4 Clearance for metallic packing

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.

Metallic piston rings should not fit snugly around the piston body. The purpose of the clearance is to allow proper ring expansion and uniform ring alignment against the walls of the liquid cylinder liner.

8.4.9.16 Phenolic piston rings application

Phenolic type piston rings are used for oil, salt water, creosote, chemicals and hot liquids, except chemicals in concentrations in excess of those given in Table 8.5.

Table 8.5 — Maximum concentration of chemicals for phenolic type rings

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

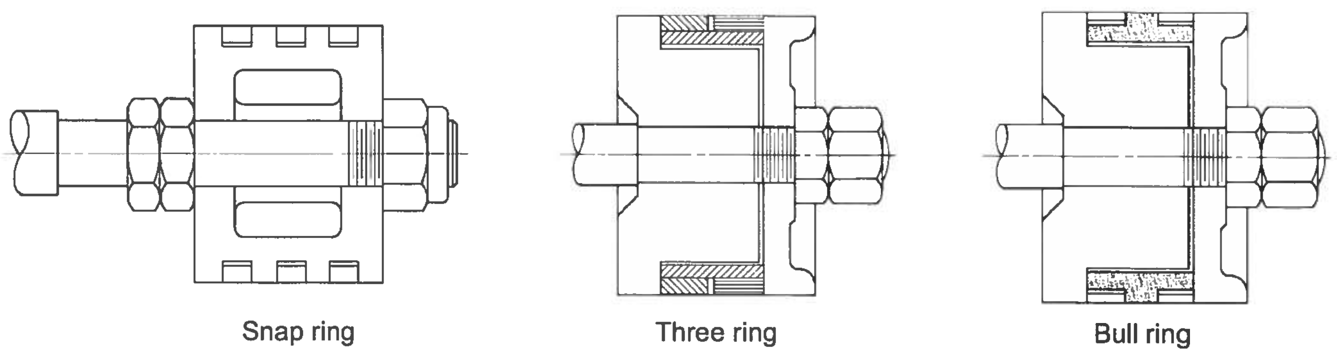


Figure 8.16 — Metallic liquid piston ring pack

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

8.4.9.16.1 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.

8.4.9.16.2 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 longitudinal allowance 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 clearance is to allow for swelling and uniform alignment against the walls of the cylinder liners.

8.4.10 Cup type pistons

8.4.10.1 Composition cup pistons

Molded composition cups are usually of Teflon-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 6900 kPa (1000 psi) pressure. Some cup manufacturers list several grades for different pressure and temperature ranges.

A list of the liquids and materials for which cup packing may be used is as follows:

Liquids/materials suitable for cup packing

Alcohol	Dye solutions	Milk	Sodium silicate
Beer	Edible oils	Molasses	Sodium sulfate solution
Boiler feed water	Fish oils	Naphtha	Starch
Brewer's mash	Formaldehyde	Oil, crude	Sump water
Butter	Fresh water	Oil, refined	Sweet water
Butane[1]	Gasoline	Paint	Tomato juice
Cane syrup	Glucose	Propane[1]	Varnish
Calcium carbonate slurry	Glue	Cold feedwater	Vinegar
Calcium sulfate slurry	Glycerin	River water – dredging operations	Viscose
Casein solution	Glycol	Salt water	Water, various
Cement slurry	Ink	Shortening	Whey
Chocolate	Jam[1]	Sodium chloride solution	
Dairy products	Methane		

[1] Consult manufacturer

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.

8.4.10.2 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 other elastomers that resist strong acids and alkalies. They can be held to close tolerances.

The installation procedure is the same as for composition cup pistons (see Figure 8.17).

8.4.10.2.1 Installation of piston cups

The following procedure is recommended for the installation of cup pistons:

a) Be sure there is sufficient clearance between the inner wall of the cup and the outside diameter of the follower disc and piston head;

b) 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;

c) Clean the cylinder liner and packing cups of any dirt or grit;

d) Lubricate the outside diameter surface of the packing cups;

e) Push the piston rod far enough into the cylinder to permit assembly of the piston head into the cylinder liner;

f) Place the piston head over the piston rod tightly against the piston rod shoulder;

g) 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 8.18). 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;

h) Slip spacer disc over the piston rod and push it tightly against the first cup;

i) 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 8.18).

j) Next place the follower disc over the piston rod tightly against the second cup;

k) Apply and tighten the piston rod nut enough to grip the flanges of the cups firmly 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 8.19.

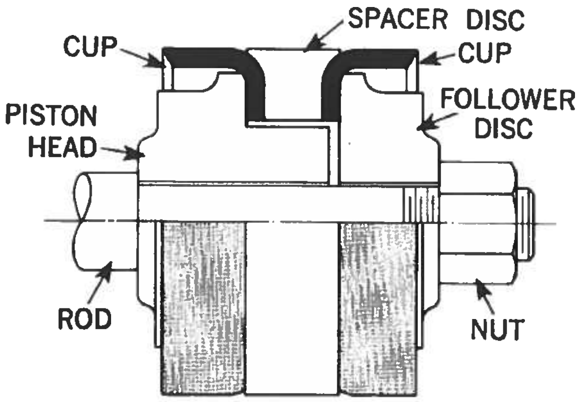


Figure 8.17 — Cup type packing

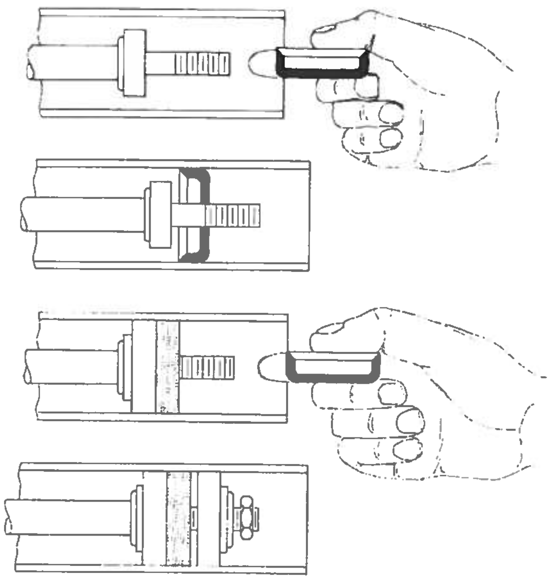


Figure 8.18 — Assembling cup piston

8.4.11 Inspection

A direct acting (steam) 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, piston rods (or plungers), stuffing box bushings, grooved liners, improper suction conditions, or faulty conditions outside the pump itself. There is NO substitute for regular, thorough preventive maintenance.

8.4.12 Main drive (steam) valve setting

8.4.12.1 Main drive (steam) valve setting – duplex pumps

The main drive (steam) valves of duplex pumps that are shipped completely assembled are set at the factory and should not require adjustment in the field; but, when such adjustments are necessary, the following instructions are applicable:

- Set both steam pistons at mid-stroke. To do this, pry against the crosshead until the drive (steam) piston strikes the drive cylinder head.

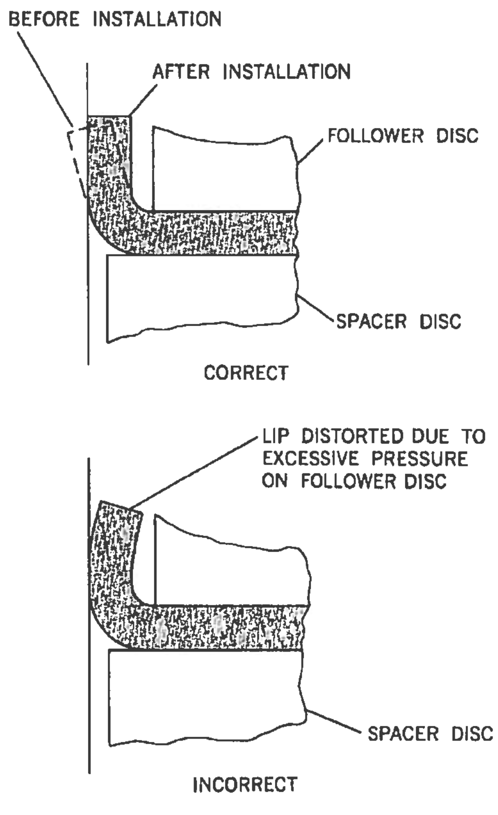


Figure 8.19 — Correct and incorrect piston rod nut tightening

Make a mark on the piston rod close to the face of the drive (steam) end stuffing box gland. Move the piston to the opposite end until it strikes the drive cylinder head, and make a mark on the rod halfway between the first mark and the face of the drive (steam) end stuffing box gland. Now move the piston backward until the second mark is flush with the face of the gland and the piston will stand at mid-stroke.

- For pumps with inside-fixed lost-motion, the following procedure is recommended. Center both drive (steam) pistons. Remove drive (steam) chest cover. Disconnect the links from the knuckles of the valve rods. Place each slide valve so that both inlet (steam) ports are covered. Hold the valve nut exactly in the center of the space between the valve nuts. Screw the valve rod through this nut until the knuckle eye is in line with the link eye, and push the link pin in place.

- Repeat this process with the other side, and the preparation is complete. After everything is adjusted and before replacing the chest cover, move one of the slide valves off-center so as to leave the inlet (steam) port open, otherwise the pump cannot be started. In operation, both valves will never be on center at the same time under any condition of running. It is only when the valves are deliberately placed, as in setting, that it can happen.

- For valve rods with lock nuts at each end of the slide valve, place pistons at mid-stroke, and slide valves so that inlet (steam) ports are covered. Do not disconnect the valve rod from the link. After valve is centered, set and lock the nuts at equal distances from the outer faces of the valve lugs, allowing about half the width of the inlet (steam) port for lost motion on each side. If this allowance gives too much or too little overlap of strokes, the lost motion must be altered by trial until the pump makes the desired overlap.

- For valves with outside adjustment, the following procedure is recommended:

- Set the pistons and valves on center. Move the adjusting nuts on the valve rod link so that they are about half the width of the inlet (stream) port away from the tappet. Repeat this operation on the opposite side. Start the pump. If the overlap is insufficient, the adjusting nuts should be screwed closer together. Care should be taken to turn back all adjusting nuts equally. When the final adjustment is made, lock the adjusting nuts securely in place.

As shown in Figure 8.11, using a thinner nut on the valve rod increases the lost motion and decreases the overlap. A thicker nut decreases the lost motion and decreases the overlap. As shown in Figure 8.12, backing the nuts away from the slide valve increases the lost motion and decreases the overlap. Screwing the nuts closer to the valve decreases the lost motion and increases the overlap.

8.4.12.2 Main drive (steam) valve setting — simplex pumps

On simplex pumps, adjusting the lost motion changes the length of the stroke. Reducing the lost motion reduces the stroke length.

A simple method of setting a simplex steam valve is to operate the pump, then back off an adjusting nut until the pump stalls at the end of the stroke. Then tighten the nut against the tappet until the pump resumes operation. Repeat this operation on the other adjusting nut; then measure the actual stroke length. Actual running stroke length should be equal to that stamped on the pump nameplate.

Many simplex pumps are built with a separate inlet (steam) connection to the auxiliary valve, allowing the pump to operate at a very low speed. If the pump stalls at low speed, or hesitates a long time at the end of stroke, check to see that the auxiliary inlet connection is fed from a point upstream of the main inlet (steam) throttling valve.

8.4.13 Drive (steam) end lubrication

The drive (steam) cylinders should be manually lubricated before initial start-up. Squirt oil in the drive end inlet pipe, drive steam chest and drive (steam) cylinders by removing the chest covers and cylinder drain valves. Unless designed for non-lubricating operation, a drive (steam) end requires a continuous small feed of oil during operation. This is normally accomplished with a mechanical force-feed lubricator. Avoid feeding too much oil to small direct acting (steam) pumps to prevent clogging of small inlet and exhaust (steam) ports.

Adjust lubricator for three drops per minute per cylinder on the small units, to a maximum of ten drops per minute per cylinder on the large pumps.

8.4.14 Drains

The drive (steam) cylinder drains should always be opened before starting to allow condensate to escape. When the pump is shut down in cold weather, all cylinders should be drained to prevent freezing.

8.4.15 Malfunctions, cause and remedy

In the event of malfunction of the pump during operation, refer to Table 8.6 for the most common cause and remedy of predominant malfunctions.

Table 8.6 — Malfunctions – cause and remedy

Malfunction	Possible Cause	Remedy
Pump fails to start	Low inlet (steam) pressure	Increase inlet (steam) pressure
	Water in drive (steam) cylinders or inlet (steam) line	Open drains in drive cylinders and inlet (steam) line. When drains run free of water and only steam issues, allow pump to run until steam end heats up, then close drains
	Auxiliary inlet (steam) line not connected	Connect
	Pump idle for extended period. Heavy lubricating oil causes piston and valve to stick in cylinder	Disassemble and clean
Pump fails to deliver required rate of flow	Low inlet (steam) pressure	Increase
	High back pressure	Decrease
	Excessively tight liquid piston rod and piston packing	Back off gland nuts and squirt oil on all rods
	Speed too slow	Adjust or replace piston packing
	Main drive (steam) valve and piston ring wear	Replace. Remove any ridges in cylinder
	Air leaking into pump	Seal leaking joints with compounds
	Liquid cylinder valves, piston rod packing, piston packing, liner worn	Reface, lap, or replace valves and seats; replace packing, liner
	Valve stuck open	Remove debris under valve
	Insufficient NPSHA	Increase suction pressure
	Pump not filling	Prime pump. Increase suction pressure
	Suction lift too great	Decrease lift. Use booster pump
	Stuck foot valve	Clean
	Clogged suction strainer	Clean or remove
Stroke too short	On simplex pumps, insufficient lost motion in valve gear	Increase lost motion by backing nuts away from tappet
	Excessive drive end cushion at ends of cylinder	Where cushion valves are used, open partway
	Tight piston or rod packing	Loosen, lubricate or replace
	Leakage of drive medium (steam) past slide or piston	Reface or lap in slide valve. Replace piston valve rings and liner
Stroke too long, causing knock	On simplex pump, excessive lost motion	Screw nuts closer to tappet
	Insufficient drive end cushion	Where cushion valves are used, close partway. Check valve and seat for wear or damage
	Little or no drive end exhaust pressure	Increase with orifice in exhaust line
	Worn drive (steam) piston rings	Replace

Table 8.6 — Malfunctions – cause and remedy (*continued*)

Malfunction	Possible Cause	Remedy
One drive (steam) piston strikes head, other side has short stroke	Excessive packing friction on one rod	Loosen gland or remove some packing on the short stroke side
	Liquid end valves leaking on over stroking side	Repair or replace
Pump vibrates and pressure fluctuates	Pump out of alignment	Realign pump. Pump may be pulled out of alignment by temperature or poor piping installation
	Piston packing too tight	Allow for swelling
	Cavitation in liquid end	Decrease suction lift or increase suction pressure, reduce pump speed, decrease liquid temperature, eliminate air leaks
Pump stalls	Main drive (steam) valve improperly set	Reset per instructions in Section 8.4.12
	Main drive (steam) valves worn	Lap valves or replace
	Auxiliary inlet line not connected	Connect
	Piston loose on rod	Tighten nut holding piston on rod
	Drive (steam) piston hitting head	On simplex pump, reduce lost motion; on duplex, partly close cushion valves
	Liquid valves hanging up	Increase clearance around valves. Remove burrs or replace worn or damaged valve parts

8.5 Reference and source material

Hydraulic Institute
9 Sylvan Way
Parsippany, NJ 07054-3802

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.

M116