Guide to the Reciprocating Pump Operations

WHEATLEY GASO

ISO 9001 REGISTERED

The Green Book
5th Edition

Preface

We have written this book to help provide you, the user of a *WGO* reciprocating pump, with the knowledge of how a reciprocating pump works and guidance in your selection and operation of a reciprocating power pump. For specific recommendations, which are not covered within, please contact our applications department and provide them with all pertinent information regarding your pumping application.

Thanks to all the persons and companies who have provided reference material used in this book. Our further thanks and acknowledgment is extended to the "Hydraulic Institute", for their section on *Hydraulic Institute Standards - Reciprocating Pumps, Power (Types, Nomenclature, Ratings, and Applications)*.

We sincerely hope this book will give you some insight into how your reciprocating pump works and why reciprocating pumps are still one of the most efficient and effective way of transporting liquids of all types.

Although all the information in this book generally applicable to all reciprocating pumps, the specific data within applies only to WGO pumps.

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Introduction

We believe the commonly accepted *conventional* nomenclature and descriptions of liquid supply systems for reciprocating pumps is often misleading and provides an inaccurate description of the pumping action. This nomenclature often describes attributes of the pump, which do not exist. Intellectually, we all understand the conventional notations and descriptions, however the concept of the pump being a machine which takes *action* on a liquid often confuses and deceives the intellect into believing that the pump has control over the amount of liquid being *introduced into* the pump.

The pump relies upon outside forces to push the liquid through the inlet manifold and inlet valve into the liquid chamber of the pump. The pump cannot *pull the liquid into the liquid chamber,* as there is no tensile strength to the liquid. The pump merely *creates a partial vacuum in the liquid chamber*, which is then filled by the forces acting upon the liquid on the inlet side of the pump.

Also, please remember when reading this manual, that liquids remain liquids only so long as outside pressure remains high enough to keep the liquid from boiling. If the pressure on the liquid is reduced below the pressure at which the liquid boils at the pumping temperature, the liquid changes to its vapor state and is no longer a liquid. It is the responsibility of the liquid supply system to provide liquid to the pump inlet at sufficient pressure to keep the liquid from boiling during the pumping cycle. Doing this will prevent many dollars from being spent on repair of the pump, the supply system, and the discharge system.

This manual is not intended to solve all the problems, which come up in applications involving *WGO* reciprocating pumps. We do hope it does provide enough insight to allow the reader to place the action of a reciprocating pump in its proper perspective.

Table of Contents

You will want to look your pump over very carefully and get acquainted with the pump prior to starting. Also refer to "Operating Instructions and Maintenance Suggestions" or "Operators Manual" published by WGO for the specific pump type and model.

I. WARNINGS AND PRECAUTIONS BEFORE AND AFTER STARTING A *WGO* RECIPROCATING PUMP.

A. Safety Precautions

1. Never start or operate a pump against a closed discharge line. Catastrophic destruction of piping or pump will promptly occur.
2. Avoid starting against a full discharge load. Bring pump up to speed using a bypass, then gradually place pump on line.
3. Bleed air from pump and piping before starting pump.
4. The cradle cover must be in place while pump is running.
5. Stand clear of pump while operating under pressure.
6. Do not operate pump above "maximum design pressure" indicated on metal tag. ***Operating above the maximum design pressure will result in damage to pump, person or property***. Notify *WGO* at the first sign of a suspected malfunction.
7. Contact *WGO* immediately if operating conditions or type of service change. Upon request you can receive recommendations to meet the new specifications and a new application tag can be sent.

B. Warnings
Before starting a *WGO* reciprocating pump.

1. Packing for piston and plunger pumps:

   a. Piston Pumps
   Piston pump stuffing boxes are not packed when shipped from the factory. These pumps must be packed before starting. If the pump is to be placed in storage before use, the piston rods should be coated with grease to prevent rusting. See the *Pump Storage* section for other steps to take before placing pump in storage.

b. Plunger Pumps

These pumps are packed prior to shipment for immediate service. If temporary storage is intended, the plunger must be protected by taking the following steps: rotate the crankshaft slightly more than one revolution, and apply a film of oil to each plunger surface as the plunger moves out of and into the stuffing box packing. Change the position of the plungers periodically/or at least once a week and reapply oil onto each plunger. These steps will help prevent pitting of the plunger from chemical reactions between the packing and the metal surface of the plunger, during periods of inactivity. See *Pumps Storage* section for more information.

2. Crankcase has been drained by *WGO* before shipping pump to you. Fill with lubricant as recommended by operating instructions to ensure proper lubrication of gears, chain, crossheads, connecting rods and shaft bearings. Then rotate the crankshaft by hand at least once a week until starting in order to avoid rust formation from water condensation on unpainted internal parts. ***Shaft bearings requiring a separate oil bath have been filled at factory with recommended lubricant but should be checked for proper oil level before starting.*** Check oil levels daily. Change lubricant at least every six months. Drain and thoroughly flush the crankcase and bearing housings when making this change.

3. Pump ***must*** set level for proper lubrication and alignment.

4. Read *Operating Instructions and Maintenance Suggestions* or Operator Manual for the specific pump model. If misplaced or lost, please contact your nearest *WGO* distributor or contact *WGO* direct for a copy.

5. A pressure relief valve must be installed in the discharge piping between the pump and any other pipefittings. See page 40 *set pressure* suggestions.

6. All belt and *coupling* guards must be firmly in place.

7. Be sure that all pipe joints and fasteners are definitely tight before putting the *pump* into operation.

8. Check *connecting* rod bearings for proper adjustment - according to operating instructions.

9. *Check* main bearings for proper adjustment - according to operating instructions.

C. Warnings: After starting a *WGO* reciprocating pump.

1. New *non-adjustable lip type* packing rings (plunger pumps only) may be expected to leak slightly for a day or so, but will gradually seat. The packing gland must be retightened and periodically checked after the packing has seated.

   DO NOT attempt to adjust packing while pump is running.

2. Piston pump packing is available in several different configurations. All are *adjustable type* packing. ***Care must be taken to ensure that this packing is not overtightened.*** On most installations, a very slight drip is desirable to aid in lubrication and as visual proof that the packing is not too tight.

3. After Two (2) Hours Operation (also refer to Appendix I)

   a. Check all the crank end connecting rod bearings for proper adjustment. Check connecting rod bolt/nut torque and adjust if necessary.

   b. Check lubricant level and pressure gauge (if forced lubrication system is furnished).

II. FUNDAMENTALS OF OPERATION

A. How a Reciprocating Pump Works

A reciprocating pump is a positive displacement mechanism, liquid discharge pressure being limited only by the strength of the structural parts. Liquid volume or capacity delivered is constant regardless of pressure, and is varied only by driver speed, speed reduction, and/or plunger/piston size changes.

Reciprocating motion is imparted to a plunger/piston by a slider crank linkage, which results in a piston motion closely approximating simple harmonic motion, as shown in *Appendix B*. This reciprocating motion alternately lowers the pressure in front of the plunger/piston when filling the pump, and increases the pressure when emptying the pump. The incoming liquid opens the suction/inlet valve. At the same time the discharge valve is held closed by the downstream line pressure. Outgoing liquid closes the inlet valve and opens the discharge valve. This simple mechanism provides high volumetric efficiency, approximately 95 percent, for most incompressible liquids.

Characteristics of a *WGO* reciprocating pump are:

(a) positive liquid displacement
(b) high pulsations caused by the sinusoidal motion of the plunger/piston
(c) high volumetric efficiency
(d) high mechanical efficiency
(e) low pump maintenance cost

B. Plunger or Piston Rod Load

Plunger or piston *rod load* is an important power end "design" consideration for *WGO* reciprocating pumps. *Rod load* is the force caused by the liquid pressure acting on the face of the plunger/piston. This load is transmitted through the adapter or piston rod to the crosshead, then to the crosshead pin, wrist pin bushing, connecting rod, crank shaft, main bearings and power frame. This load is directly proportional to the discharge gauge pressure and the square of the plunger/piston diameter.

Occasionally, allowable liquid end pressures limit the "allowable" *rod load* to a value below the "design" rod load. ***It is important that liquid end pressure DOES NOT exceed WGO's latest published recommendations.***

Rod loads **are not** generally used when "applying" the pump. They **are** used to establish the power frame design - **not** to determine the allowable pressures, in most cases, for each size plunger/piston.. Also see the *Sizing Pumps in High Inlet Pressure Conditions* section.

C. Calculations of Volumetric Efficiency

Volumetric Efficiency (E_V) is defined as the ratio of actual pump capacity to ideal pump displacement. The E_V calculation depends upon the internal configuration of each individual liquid end cylinder, plunger/piston size, and the liquid being pumped. Given full details regarding differential pressure, pumped liquid mixture, and expected temperature rise on the discharge stroke – *WGO* can calculate this efficiency.

1. Calculating E_V for Water
   See Table 4 page A31 for the water compressibility chart. An example of volumetric efficiency calculation for water is also shown.

2. Calculating E_V for Hydrocarbons
   See Table 6 page A34 for the physical properties of hydrocarbons. For compressible liquids such as these, horsepower calculations are slightly more complex than for incompressible liquids. However, the magnitude of the horsepower required will be slightly less than if calculated for the full displacement.

3. Displacement Table

   Table 1 page 6 can be used to determine pump displacement and whether or not the proper efficiency is being obtained.

Example 1:
Find the *capacity* (Q), in Barrels Per Hour (BPH), for a single acting 3-1/2" x 4" triplex plunger pump operating at 95% volumetric efficiency (E_V) and a speed of 350 rpm.

From Table 1 a 3-1/2" plunger with a 4" stroke will displace 0.167 gallons per stroke. This type of pump will displace liquid at a rate of 3 forward strokes only per revolution. Therefore, the displacement per revolution is 0.167 gal./stroke x 3 strokes/rev. = 0.501 gal./rev. (gpr). At 350 rpm, the total displacement is 175.35 gallons per minute (GPM). Dividing by 0.7 gives 250.5 BPH *displacement* (D).

Therefore, the pump capacity at a 95% volumetric efficiency is

$$Q = \frac{D\,E_V}{(100)} = \frac{250.5\,(95)}{(100)}$$

$$Q = 237.9 \text{ BPH}$$

TABLE 1

Displacement Table

Plunger Diameter	Plunger Area	Stroke Length - inches									
		2	3	4	5	6	7	8	9	10	12
		Displacement per Stroke - U.S. Gallons or Decimal Parts Thereof									
inches	sq. inches										
0.500	0.196	0.002	0.003	0.003	0.004	0.005	0.006	0.007	0.008	0.009	0.010
0.625	0.307	0.003	0.004	0.005	0.007	0.008	0.009	0.011	0.012	0.013	0.016
0.750	0.442	0.004	0.006	0.008	0.010	0.011	0.013	0.015	0.017	0.019	0.023
0.875	0.601	0.005	0.008	0.010	0.013	0.016	0.018	0.021	0.023	0.026	0.031
1.000	0.785	0.007	0.010	0.014	0.017	0.020	0.024	0.027	0.031	0.034	0.041
1.125	0.994	0.009	0.013	0.017	0.022	0.026	0.030	0.034	0.039	0.043	0.052
1.250	1.227	0.011	0.016	0.021	0.027	0.032	0.037	0.043	0.048	0.053	0.064
1.375	1.485	0.013	0.019	0.026	0.032	0.039	0.045	0.051	0.058	0.064	0.077
1.500	1.767	0.015	0.023	0.031	0.038	0.046	0.054	0.061	0.069	0.077	0.092
1.625	2.074	0.018	0.027	0.036	0.045	0.054	0.063	0.072	0.081	0.090	0.108
1.750	2.405	0.021	0.031	0.042	0.052	0.062	0.073	0.083	0.094	0.104	0.125
1.875	2.761	0.024	0.036	0.048	0.060	0.072	0.084	0.096	0.108	0.120	0.143
2.000	3.142	0.027	0.041	0.054	0.068	0.082	0.095	0.109	0.122	0.136	0.163
2.125	3.547	0.031	0.046	0.061	0.077	0.092	0.107	0.123	0.138	0.154	0.184
2.250	3.976	0.034	0.052	0.069	0.086	0.103	0.120	0.138	0.155	0.172	0.207
2.375	4.430	0.038	0.058	0.077	0.096	0.115	0.134	0.153	0.173	0.192	0.230
2.500	4.909	0.043	0.064	0.085	0.106	0.128	0.149	0.170	0.191	0.213	0.255
2.625	5.412	0.047	0.070	0.094	0.117	0.141	0.164	0.187	0.211	0.234	0.281
2.750	5.940	0.051	0.077	0.103	0.129	0.154	0.180	0.206	0.231	0.257	0.309
2.875	6.492	0.056	0.084	0.112	0.141	0.169	0.197	0.225	0.253	0.281	0.337
3.000	7.069	0.061	0.092	0.122	0.153	0.184	0.214	0.245	0.275	0.306	0.367
3.250	8.296	0.072	0.108	0.144	0.180	0.215	0.251	0.287	0.323	0.359	0.431
3.500	9.621	0.083	0.125	0.167	0.208	0.250	0.292	0.333	0.375	0.417	0.500
3.750	11.045	0.096	0.143	0.191	0.239	0.287	0.335	0.383	0.430	0.478	0.574
4.000	12.566	0.109	0.163	0.218	0.272	0.326	0.381	0.435	0.490	0.544	0.653
4.250	14.186	0.123	0.184	0.246	0.307	0.368	0.430	0.491	0.553	0.614	0.737
4.500	15.904	0.138	0.207	0.275	0.344	0.413	0.482	0.551	0.620	0.689	0.826
4.750	17.721	0.153	0.230	0.307	0.384	0.460	0.537	0.614	0.690	0.767	0.921
5.000	19.635	0.170	0.255	0.340	0.425	0.510	0.595	0.680	0.765	0.850	1.020
5.250	21.648	0.187	0.281	0.375	0.469	0.562	0.656	0.750	0.843	0.937	1.125
5.500	23.758	0.206	0.309	0.411	0.514	0.617	0.720	0.823	0.926	1.029	1.234
5.750	25.967	0.225	0.337	0.450	0.562	0.674	0.787	0.899	1.012	1.124	1.349
6.000	28.274	0.245	0.367	0.490	0.612	0.734	0.857	0.979	1.102	1.224	1.469
6.250	30.680	0.266	0.398	0.531	0.664	0.797	0.930	1.063	1.195	1.328	1.594
6.500	33.183	0.287	0.431	0.575	0.718	0.862	1.006	1.149	1.293	1.437	1.724
6.750	35.785	0.310	0.465	0.620	0.775	0.929	1.084	1.239	1.394	1.549	1.859
7.000	38.485	0.333	0.500	0.666	0.833	1.000	1.166	1.333	1.499	1.666	1.999
7.250	41.283	0.357	0.536	0.715	0.894	1.072	1.251	1.430	1.608	1.787	2.145
7.500	44.179	0.383	0.574	0.765	0.956	1.148	1.339	1.530	1.721	1.913	2.295
7.750	47.173	0.408	0.613	0.817	1.021	1.225	1.429	1.634	1.838	2.042	2.451
8.000	50.266	0.435	0.653	0.870	1.088	1.306	1.523	1.741	1.958	2.176	2.611
8.250	53.456	0.463	0.694	0.926	1.157	1.388	1.620	1.851	2.083	2.314	2.777
8.500	56.745	0.491	0.737	0.983	1.228	1.474	1.720	1.965	2.211	2.457	2.948
8.750	60.132	0.521	0.781	1.041	1.302	1.562	1.822	2.083	2.343	2.603	3.124
9.000	63.617	0.551	0.826	1.102	1.377	1.652	1.928	2.203	2.479	2.754	3.305

Example 2:
Find the *rpm* (n) required for a single acting 3" x 5" triplex plunger pump operating at E_V = 85% and *capacity* (Q) = 200 BPH.

$$D = \text{displacement} = \frac{Q\,(100)}{E_v}$$

$$D = \frac{200}{0.85} = 235.29 \text{ BPH}$$

$$gpr = (0.153 \text{ gal./stroke})\ (3 \text{ strokes/rev.})$$

$$gpr = 0.459$$

$$GPM = BPH \times 0.7 = 235.29 \times 0.7$$

$$GPM = 164.7$$

At the given displacement and volumetric efficiency, the pump speed (n) is

$$n = \frac{GPM}{gpr} = \frac{164.7}{0.459} \text{ rpm}$$

$$n = 358.8 \text{ rpm}$$

Example 3:
Find the *displacement* (D), in gallons per minute (GPM), for a double acting 4" x 10" duplex piston pump running at a crankshaft speed of 60 rpm. Pump has 1-1/2" diameter piston rods. This type pump will displace liquid at a rate of; 2 *forward* strokes/rev. and 2 *rearward* strokes/rev. (the 2 *rearward* strokes/rev. will displace less liquid than 2 *forward* strokes due to the amount of volume taken up by the piston rod).
Assume the pump E_V = 95%.

$$pr = gpr_f + gpr_r$$

Where,

gpr_f = gal./rev. for the 2 *forward* strokes

$$gpr_f = 0.544 \times 2 = 1.088$$

gpr_r = gal./rev. for the 2 *rearward* strokes

$$gpr_r = [0.544 - (0.544 - 0.077)\{\text{piston rod}\}] \times 2$$

$$gpr_r = 0.934$$

$$gpr = (1.088 + 0.934) = 2.022 \text{ gpr}$$

$$D = \frac{Q\,(100)}{E_v} = \frac{2.022\,(60)(95)}{100} = 115.3 \text{ GPM}$$

Note 1: One barrel equals 42 U.S. gallons. Gallons per minute (GPM) divided by 0.7 equals barrels per hour (BPH). Displacement is the ideal volume swept by the plunger or piston on the discharge stroke during any selected time period.

Note 2: Under the conditions and definitions given above, the pump must run at a higher *rpm* in order to displace enough volume to deliver the required amount of liquid.

D. Horsepower Calculations (see ***Notes 3, 4, 5,*** and ***6*** at the end of this section)

1. The *required horsepower* (hp) for driving a single acting reciprocating pump is calculated by the following equation:

$$hp = \frac{P_d\ Q\ (100)}{1714\ E_m} - \frac{P_i\ Q\ (E_m\text{-}5)}{1714\ (100)}$$

Where:

D = pump displacement, U.S. gallons per minute (GPM)
Q = pump capacity, (GPM). Where $Q = D \times E_v$
P_d = liquid pressure at pump discharge, lbs/in^2 gauge (PSIG).
P_i = liquid pressure at pump inlet, lbs/in^2 gauge (PSIG).

If $P_i \le 50$ PSIG, then

$$\frac{P_i\ Q\ (E_m - 5)}{1714 \times (100)} = 0$$

E_m = mechanical efficiency of pump, percent (for the above P_d and $P_i \le$ 50 PSIG):

a. 90% for pumps without built-in reducer (power input from crankshaft).

b. 85% for pumps with built-in or bolted-on gear reduction (power input from pinion shaft).

c. Reduce E_m by the mechanical efficiency losses of any intermediate speed reduction between drive and pump.

General E_m values:

V-belt drive	5%
HTD drive	5%
Parallel shaft gear reducer	5%
Etc. . . .	

$E_m - 5$ = efficiency of power recovery due to liquid pressure at pump inlet.

E_v = volumetric efficiency, where $E_v = \frac{Q}{D}$

Example 4:

Find the required *horsepower* (hp) and *speed* (rpm) for driving a single acting 2-3/4" x 5" triplex plunger pump under the following operating conditions:

Pump displacement	=	D	=	138.9 GPM
Pressure at pump inlet	=	P_i	=	200 psig
Pressure at pump discharge	=	P_d	=	2020 psig
Mechanical efficiency of pump	=	E_m	=	75%
Volumetric efficiency of pump	=	E_v	=	80%

$$Q = D \times E_v = 138.9 \times 0.8 = 111.12 \text{ GPM}$$

$$hp = \frac{(Q \times P_d)(100)}{1714 \times (E_m)} - \frac{(Q \times P_i)(E_m - 5)}{1714 \times (100)}$$

$$hp = \frac{(111.12 \times 2020)(100)}{1714 \times (75)} - \frac{(111.12 \times 200)(75-5)}{1714 \times (100)}$$

$$hp = 165.53$$

$$gpr = 0.129 \text{ gal./plunger} \times 3 \text{ plungers/rev.}$$

gpr = 0.387 gal./rev. (see Table 1)

$$n = \frac{GPM}{gpr} = \frac{138.9}{0.387} = 358.9 \text{ rpm}$$

2. The required *horsepower* (hp) for driving a double acting duplex piston pump is calculated by the following equation:

$$hp = \frac{(Q)(P_d - P_i)(100)}{1714 (E_m)}$$

Where,

D	=	pump displacement, gallons per minute (GPM)
Q	=	pump capacity, GPM where ($Q = D / E_v$)
P_d	=	liquid pressure at pump discharge, lbf/in^2 gauge (psig).
P_i	=	liquid pressure at pump inlet, lbf/in^2 gauge (psig).
E_m	=	pump mechanical efficiency, percent (for above P_d and P_i).

a. 90% for pumps without built-in reducer (with power input from crankshaft)

b. 85% for pumps with built-in reducer (with power input from pinion shaft)

c. Reduce E_m by the mechanical efficiency losses of any intermediate speed reduction between driver and pump

General E_m values:	
V-Belt drive	5%
HTD drive	5%
Parallel shaft gear reducer	5%
Etc. . . .	

$$E_V = \text{pump volumetric efficiency, } \% = \frac{Q\,(100)}{D}$$

Example 5:

Find the *required horsepower* (hp) and *pump speed* (n) for driving a double acting 5" x 10" duplex piston pump with the following conditions.

Pump capacity	=	Q	=	281.7 GPM
Liquid pressure at pump inlet	=	P_i	=	50 psig
Liquid pressure at pump outlet	=	P_d	=	330 psig
Mechanical efficiency of pump	=	E_m	=	90%
Volumetric efficiency of pump	=	E_V	=	85%
Piston Rod diameter	=	d	=	1-1/2"

$$hp = \frac{Q\,(P_d - P_i)\,100}{1714\,(E_m)} = \frac{281.7\,(330-50)\,100}{1714\,(90)}$$

$$hp = 51.13$$

$$gpr = [0.850 \times 4] - [\frac{p}{4}(1.5)^2] \times 10 \times 2 \times \frac{1}{231} = 3.24$$

$$D = (Q \times 100) / E_V = (281.7 \times 100) / 85 = 331.41 \text{ GPM}$$

$$n = \frac{D}{gpr} = \frac{331.4}{3.24} = 102.3 \text{ rpm}$$

3. Quick calculation of horsepower requirement of a reciprocating pump.

Where the above formulas are very accurate, they are somewhat detailed. A quick and easy method of calculating horsepower which is almost as accurate as the other methods, is as follows:

A flow of one barrel per hour (1 BPH) of any specific gravity liquid against one pound per square inch gauge pressure (1 psig) requires 0.00040833 *correction factor* for theoretical horsepower. If the pump mechanism is approximately 90% mechanically efficient, the ***conversion factor*** can be corrected, for convenience, to 0.00045 to reflect this efficiency loss.

Therefore,

$$hp = 0.00045 \times P_d \times \text{BPH capacity}$$

$$\text{BPH capacity} = \frac{D\,E_v}{100} = \frac{(\text{BPH displacement} \times E_v)}{100}$$

Where,

P_d = liquid pressure at pump discharge, psig
E_V = pump volumetric efficiency, %

Example 6:

Find the required horsepower (hp) to displace 100 gallons per minute (GPM) against a P_d = 1000 psig at a E_v = 95%.

$$hp = \frac{0.00045 \times 1000 \times 100 \times 95}{0.7 \times 100} = 61.1$$

Note 3: One barrel is 42 U.S. gallons. Gallons per minute (GPM) divided by 0.7 equals barrels per hour (BPH). *Displacement* is the theoretical volume swept by the plunger or piston on the discharge stroke during any selected time period.

Under the conditions and definitions given above, the pump speed must be a higher rpm in order to displace enough volume to deliver the required amount of liquid.

Note 4: Mechanical efficiency (E_m) expressed as a percentage of total horsepower requirement can be used only if the pump is to be applied at or near its maximum designed rating. The horsepower required to operate a large pump at a small horsepower will usually be a considerably higher percentage of the total horsepower for the application.

We suggest that if the hydraulic horsepower - calculated without correction for mechanical efficiency - is less than 50% of the maximum design rating for the pump, you contact WGO for our recommendation for the driver horsepower.

Note 5: If the horsepower requirement is less than 15, as calculated by any of the above methods, we recommend that the motor driver be one size larger than the calculated requirement. This is because the horsepower required by a speed reduction device (belt, gear reducer, etc.) is relatively fixed and cannot be factored into the equation as a percentage of the smaller horsepower requirements.

Note 6: These computations are intended as a *guide to standardization* and may be modified if efficiency ratings for the pump application are lower than $E_v = 95\%$ and/or $E_m = 90\%$.

E. Devices for Liquid Pulsation Control, Suction/Inlet and Discharge

A good inlet/suction and discharge pipe layout for reciprocating pumps of conventional type frequency require no pulsation control devices to compensate for normal variations in velocity flow in the complete piping system.

Where the suction or discharge lines are of considerable length, or the inlet has sufficient head, or liquid handled is hot – a suction or discharge dampening device of suitable size may sometimes be necessary to ensure smooth, quite operation. Dampening devices should be considered as a part of the piping system, rather than as a pump accessory.

The size and pre-charge of the dampener device will depend upon the type of pump, the liquid and the layout of the piping system. Recommendations as to size and type of device(s) should be obtained from the device manufacturer. Be sure to provide full information on the piping installation. Without complete knowledge of the piping system, it is impossible to determine the size and pre-charge of the dampeners. For bladder type dampeners, provisions should be made to keep the unit(s) charged with nitrogen or similar inert gas, in accordance with the device manufacturer's recommendations. An exhausted device is of no value.

Dampeners, particularly on the suction (which are required more frequently than discharge), 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.

A properly sized, located and charged device may reduce the length of pipe used in the acceleration head equation to a value of 5 to 15 nominal pipe diameters. Figure 52 in Appendix A is a *suggested* piping system for power pumps.

III. PUMP MATERIAL SELECTION

The following material selection charts; Table 2 (for plunger pumps) and Table 3 (for piston pumps), are *general* recommendations. For more detailed or alternate recommendations, please consult *WGO*.

TABLE 2

Material Selection Chart for Plunger Pumps

Liquid Description	REF	Liquid End	Valves	Plungers	Packing "J" Style	Packing "V" Style	Packing Braid Style	Packing Thd Gland	Packing Lube Feed
Amine		(9)	(7)	(3)	(4)	(4)	(4)	(6)	(2)
DEA		CS	D/CD, H/TD CF8M SS	HS	NA	SF=4511-4 SSF=4516-4	232	CS or DI	No
MEA	(1)	CS	D/CD, H/TD CF8M SS	HS	NA	SF=6618-4 SSF=1043-4	232	CS or DI	No
Carbon Dioxide			(7)	(3)	(4)	(4)	(4)	(6)	(2)
Dry or Wet	(5)	CS	D/CD, H/TD CF8M SS	HC/HT	NA	SF=805-4	8921-K	Temp. Var.	Yes
Condensate		(5)	(7)	(3)	(4)	(4) & (9)	(4)	(6)	(2)
n-Butane/iso-Butane		CS or SS	D/CD, H/TD CF8M SS	HC/HT	845	SF=809-4 SSF=1068-4	238	CS or DI	Yes
Ethane/Methane		CS or SS	D/CD, H/TD CF8M SS	HC/HT	845	SF=809-4 SSF=1068-4	8921-K	CS or DI	Yes
Liquid Propane Gas or Natural Gas Liquid	LPG NGL	CS or SS	D/CD, H/TD CF8M SS	HC/HT	838/835	SF=809-4 SSF=1068-4	217	Temp. Var.	Yes
Lean Oil		CS or SS	D/CD, H/TD CF8M SS	HC/HT	NA	SF=6625-4 SSF=1073-4	217	CS or DI	Yes
Fuel Oil			(7)	(3)	(4)	(4)	(4)	(6)	(2)
Diesel / Kerosene		CS	D/CD, H/TD CF8M SS	HC/HT	838/835	SF=809-4 SSF=1068-4		CS or DI	No
Glycol			(7)	(3)	(4)	(4)	(4)	(6)	(2)
Ethylene/DiEthylene		CS	H/TD CF8M SS	HC/HT	NA	SF=809-4 SSF=1068-4	217	CS or DI	No
Oil			(7)	(3)	(4)	(4)	(4)	(6)	(2)
Crude, Clean		CS	D/CD, H/TD w/CF8M SS	HC/HT	838/835	SF=809-4 SSF=1068-4	238	CS or DI	No
Crude w/Solids		CS	ARS w/17-4PH SS	HT	NA	NA	8921-K	CS or DI	No
Crude w/H2S	(10)	CS	D/CD, H/TD w/CF8M SS	HC/HT	NA	SF=6618-4 SSF=1043-4	8921-K 217	CS or DI	No
Hydraulic		CS	D/CD, H/TD w/CF8M SS	HC/HT	838/835	SF=805-4	212	CS or DI	No
Methanol			(7)	(3)	(4)	(4)	(4)	(6)	(2)
		CS or SS	D/CD, H/TD w/CF8M SS	HS/HST	NA	SF=6677-4 SSF=1040-4	232	CS or DI	Yes
Water			(7)	(3)	(4)	(4)	(4)	(6)	(2)
Hot/Boiler Feed		NAB	D/CD, H/TD w/CF8M SS	HC/HT	NA	NA	8921-K 249	AB or SS	Yes
Salt/Produced		NAB	D/CD, H/TD w/CF8M SS	SC	NA	NA	241 8921-K	AB or SS	Yes
Salt/Produced w/H2S	(10)	NAB	D/CD, H/TD w/CF8M SS	SC	NA	NA	241 8921-K	AB or SS	Yes
Sea, Non-Aerated		NAB	D/CD, H/TD w/CF8M SS	SC	NA	SSF=1068-4 SF=809-4	241 8921-K	AB or SS	No
Sea, Aerated		Dplx SS	D/CD, H/TD w/Duplex SS	SC	NA	SSF=1068-4 SF=809-4	8921-K	SS	No

TABLE 2
Reference Notes

1. Bronze/metals with bronze (Monel, etc.), should not be used in any liquid exposed components.

2. If contamination of pumpage by packing lubricant is acceptable, the following is recommended.

Packing	Sym.	Comments
All Braid Styles	NA	Use positive back-drip feed lubrication.
"J" Style	J	Use positive pressure feed lubrication.
"V" Style	SF/SSF	Use positive back-drip feed lubrication.

Brake fluid, low temperature transmission fluid, or refrigeration oil should be used as packing lubricant at lower temperatures below -10 degrees F.

3.

Plunger Type	Sym.	Comments
Solid Ceramic	SC	If temperature either exceeds 180 degrees F, or differential is 100 degrees F, a ceramic coated (CC) or other hard surfaced metal base plunger is required.
Ceramic Coated	CC	Ceramic coated over stainless steel metal base.
Hard Surfaced	HC	Colmonoy Alloy #6 coated over stainless steel metal base.
Hard Surfaced	HT	Tungsten & Colmonoy Alloy #6 coated over stainless steel metal base.
Hard Surfaced	HS	Tungsten & Stellite Alloy #157 coated over stainless steel metal base.

4. Always check suitability of packing for pumping temperature and pressure.

Packing Type	Sym.	Comments
All Braid Style	NA	Adjustable, max. operating pressure is 4000 psig, without written factory approval.
"J" Style	J	Non-adjustable, max. operating pressure is 2500 psig.
"V" Style	SF	Single stack height spring-loaded if -4.
"V" Style	SSF	Double stack height spring-loaded if -4, min. operating pressure is 1500 psig.

5.

Minimum Temperature	Forged/Billet	General Specification
greater than +0 degrees F	Carbon/Low-Alloy Steel	ASTM A105 or A516 Grade 70
from +0 to -20 degrees F	Carbon/Low-Alloy Steel	ASTM A350 Grade LF1
from -20 to -50 degrees F	Carbon/Low-Alloy Steel	ASTM A350 Grade LF2
less than -50 degrees F	Stn. Stl./Ni-Al-Brz	ASTM A473 Type 316/ASTM B148 Gr. 955

6.

Material	Sym.	Material	Sym.	Material	Sym.
Carbon Steel	CS	Stainless Steel	SS	Ni-Al-Brz	NAB
Ductile Iron	DI	Al-Brz	AB		

7.

Valve Type	Sym.	Comments
Delrin/Celcon Disc	D/CD	Pressures up to 1500 psig & temperatures up to 150 degrees F.
HTP/Titanium Disc	H/TD	Pressures above 1500 psig & temperatures up to 300 degrees F.
Abrasion Resistant SS	ARS	All pressures and temperatures up to 180 degrees F.

8. Cast CS or DI liquid end are suitable only if the customer requirements allow their use.

9. Traces of Benzene or Toluene also require compatible material for exposed gaskets/seals.

10. Consult factory on material requirements for NACE Specification MR-01-75 latest revision.

TABLE 2
Packing Styles & Materials

Type	Design	Style Number	Base Material	Base Description	Temp. Range degrees F	
Braid	DF/SC	212	NA	ARAMID Fiber w/High Temp. Lubricant	-50	+500
Braid	DF/SC	217	NA	PTFE Filament and ARAMID Fiber Corners	-50	+500
Braid	DF/SC	232	NA	Pure PTFE Filament Yarns w/PTFE Suspensoid	-450	+500
Braid	DF/SC	238	NA	Acrylic Filament Yarns w/PTFE Suspensoid	-50	+500
Braid	DF/SC	241	NA	Synthetic Filament and ARAMID Filament Corners	-50	+500
Braid	DF/SC	249	NA	Carbon Yarn w/PTFE Suspensoid Throughout	-450	+650
Braid	DF/SC	8921-K	NA	ARAMID Fiber Corners Synthetic Yarn w/PTFE	-170	+550
"V"	SF	805-4	Nitrile	Nitrile, TFE, Polyester/Cotton	-40	+250
"V"	SF	809-4	Nitrile	Hard Nitrile, Polyester/Cotton	-10	+500
"J"	Molded	835	Nitrile	Nitrile, Cotton, Phenolic	-10	+220
"J"	Molded	838	Nitrile	Nitrile, Nylon, Phenolic	-10	+180
"J"	Molded	845	Nitrile	Nitrile, TFE, Nylon, Phenolic	-40	+180
"V"	SSF	1040-4	TFE ARAMID	Sintered TFE, ARAMID/Glass	-30	+500
"V"	SSF	1043-4	FEPM	Hard AFLAS, ARAMID/Glass	0	+450
"V"	SSF	1068-4	Nitrile	Hard Nitrile, Polyester/Cotton	-10	+250
"V"	SSF	1073-4	FKM	Hard FKM, ARAMID/Glass	0	+425
"V"	SF	4511-4	TFE	Moly/Glass Filled TFE	-30	+500
"V"	SSF	4516-4	TFE	Moly/Glass Filled TFE	-30	+500
"V"	SF	6618-4	FEPM	Hard AFLAS, ARAMID/Glass	0	+450
"V"	SF	6677-4	TFE ARAMID	Sintered TFE, ARAMID/Glass	-30	+500
"V"	SF	6625-4	FKM	Hard FKM, ARAMID/Glass	0	+425

FKM = Fluorocarbon Elastomer (VITON, FLOUREL).

TFE = TetraFluoroEthylene.

AFLAS = Fluoro Elastomer, TFE and Propylene.

AFLAS is a Trademark of "ASAHI Glass Co."

FLOUREL is a Trademark of "3M Corp."

VITON is a Trademark of "E.I. Dupont"

TABLE 3

Material Selection Chart for Piston Pumps

Liquid Description	Liquid End	Valves	Pistons & Rings/Rubbers	Liners	Piston Rods	Packing	Gland
Creosote							
	CS or CDI	CS/SS WG	Cast Iron w/ Chrome Rings	File Hard Steel	File Hard	Braid	CI Thd
Condensate							
Ethane/Methane (1)	CS	CS/SS WG	Cast Iron w/ Nylon-TFE Rings	File Hard Steel	Hard Coated	Lip	CI Thd
Liquid Propane Gas (1)	CS	CS/SS WG	Cast Iron w/ Nylon-TFE Rings	File Hard Steel	Hard Coated	Lip	CI Thd
Natural Gas Liquid (1)	CS	CS/SS WG	Cast Iron w/ Nylon-TFE Rings	File Hard Steel	Hard Coated	Lip	CI Thd
Lean Oil (1)	CS	CS/SS WG	Cast Iron w/ Nylon-TFE Rings	File Hard Steel	Hard Coated	Lip	CI Thd
Crude Oil							
Clean	CS	CS WG	Cast Iron w/ Chrome Rings	File Hard Steel	File Hard Steel	Lip	CI Thd
Clean w/H2S (2)	CS	SS WG	Cast Iron w/ Chrome Rings	File Hard Steel	File Hard Steel	Braid	CI Thd
w/Solids	CS	Slush NBR	Slush w/ NBR Rubbers	File Hard Steel	Hard Coated	Lip	CI Thd
w/Solids & H2S (2)	CS	Slush NBR	Slush w/ FEPM Rubbers	File Hard Steel	Hard Coated	Braid	CI Thd
Mud							
Drilling	CS	Slush NBR	Slush w/ NBR Rubbers	File Hard Steel	Hard Coated	Lip	CI Bolted
Water							
Fresh	CDI or CS	Slush NBR	Slush w/ NBR Rubbers	File Hard Steel	Hard Coated	Lip	CI Bolted
Mine	CS	Slush NBR	Slush w/ CR Rubbers	File Hard Steel	Hard Coated	Lip	CI Bolted

Symbol	Type	Chemical Name	Temp. Range
CR	Neoprene	Polychloroprene	-20F to +225F
FEPM	Aflas	FluoroElastomer - TetraFluoroEthylene & Propylene	-10F to + 250F
NBR	Nitrile	Butadiene-Acrylonitrile	-10F to + 250F

Reference Notes:

1. Extra Deep Stuffing Boxes with Force Feed Lubrication is recommended for these applications.
2. Consult Factory on material requirements for NACE Specification MR01-75 latest revision.

IV. PUMP APPLICATIONS

A. Types of Reciprocating Pumps

1. Single acting plunger pumps fill on the reverse stroke and exhaust on the forward stroke. The liquid end is sealed from the atmosphere by packing around the plunger. This type of pump is commonly called an "outside packed" pump. *WGO* manufacturers these pumps with one, two, three, or five plungers/cylinders. This type of pump is horizontally operating.

2. Single acting piston pumps exhaust only during the forward stroke of the piston, that is, during one half of the revolution.

3. Double acting piston pumps fill and exhaust on the same stroke, one side of the piston facing the end of the cylinder being filled and the other side of the piston exhausting the other end of the same cylinder. This type of pump is commonly called an "inside packed" pump where the "packing" is a series rings or a set of rubbers which stop leakage from one side of the piston to the other. The piston rod diameter is relatively small compared to the diameter of the piston and is easily packed to avoid leakage. This type of pump is also horizontally operating.

B. Where Reciprocating Pumps are Used

1. *WGO* reciprocating pumps are used in all applications requiring a combination of high differential pressure and relatively low capacities.
   a. Single acting plunger pumps for high capacities and high pressures at high rpm, low initial costs, and average lifetime.
   b. Double acting piston pumps for high capacities and moderate pressure, low rpm, moderate initial cost and extremely long life.

C. Sizing and Selecting a Reciprocating Pump

1. In order to properly select and size a reciprocating pump, *WGO* requires the following information for most applications.
   a. Characteristics of the liquid
      1) Liquid composition (including % and type of solids present)
      2) Outlet and inlet pumping temperatures
      3) Specific gravity of liquid at outlet and inlet pressures & temperatures
      4) The corrosives and/or abrasives present
      5) Vapor pressure at pumping temperature
      6) Viscosity of liquid at pumping temperature

2. Desired pumping capacity, at pumping temperature, in units of: gallons per minute, barrels per hour, or barrels per day.
   a. Whether a future or alternate capacity is contemplated for the pump(s).

3. Pressure conditions required.
   a. Liquid pressure at the pump discharge.
   b. Liquid pressure at the pump inlet/suction (NPSHA).
   c. Whether a future or altemate pressure condition is contemplated for the pump(s).

4. Electrical or other area hazard(s).

5. Flange rating required for the operating pressures and temperatures, if other than standard.

6. Preferred/specified metallurgy of major parts.

7. Driver type, to include power characteristics; electrical voltage/phase/hertz or gasoline/diesel fuel.

8. Space limitations/constraints.

9. Site ambient temperature range.

10. Indoor or outdoor location.

D. Pump storage

In order to prevent permanent damage to pumps, which are placed in storage, the following procedures are recommended:

1. Short Term Storage
   Recommended for period less than 6 months and a minimum requirement for any exported pumps or pump package units without customer procedures.

   a. Where possible, place pump in an enclosed storage facility. If enclosed storage is not available, cover the entire pump and associated exposed attachments with a heavy tarpaulin.

   b. Remove crankcase cover and fill the crankcase with lubricant (approved and specified on pump applications tag).

   c. Replace crankcase cover.

d. Drain all fluids and remove; valve covers, cylinder heads, and inlet and discharge flange plugs.

e. Apply a film covering of lubricant to all exposed parts.

f. Remove cradle cover and coat plunger or rod surface with lubricant.

g. Replace valve covers, cylinder heads and inlet and discharge flange plugs.

h. Rotate crankshaft 10 complete turns and reapply liquid end protection, including plungers (as described in steps d, e, and f above), weekly.

i. Check for oil leaks around pump and replace gaskets or seals immediately if necessary.

Refer to engine, motor, gear reducer, etc. manufacturer's instructions for storage procedures.

2. Long Term Storage
Recommended for a period exceeding 6 months.

a. The pump and parts should be stored in a clean, dry location, free from temperature extremes, in an approximately level position and without distortion.

b. It is desirable to completely fill the power end (crankcase cavity, etc.) of the pump with the approved lubricant. Such a fluid level would ensure all parts within the power end are protected. To fill the power end, remove the hand cover and fill with lubricant. Be sure to replace the hand cover. Service the remainder of the pump as directed above in steps d-h.

c. Inspect periodically for possible oil leakage and replace gaskets or seals immediately if necessary.

Caution: *Before placing the pump, engine, gear reducer, etc. into service, fill the crankcase with the manufacturer recommended lubricant and check for proper lubricant level.*

Caution: *Double acting piston pumps, having steel or file hard steel piston rods, require special attention.*
Remove liquid end stuffing box packing in pumps having piston rods in these materials as they are subject to chemical and atmospheric corrosion, if left in contact with the packing.
Exposed sections of these rod materials must protected by a coating of heavy grease or rust preventative.

Each time the pump is rotated during storage, this coating must be checked and reapplied, if needed.
A heavy coating of lubricant is also recommended immediately after servicing or recommissioning the liquid end.

E. Sizing a *WGO* Pump for High Inlet Pressure Conditions

1. For double acting piston pumps, no derating is required.

2. For a single acting plunger pump, where the *rod load* due to inlet pressure is less than 20% of full plunger/rod load, no derating required.

3. For a single acting plunger pump (with <u>standard</u> bronze wrist pin bushings), where the *rod load* due to inlet pressure is greater than 20% of full plunger/rod load, derating is as follows:

   a. For triplex pumps, add two-thirds the inlet/suction pressure to the discharge pressure in selecting the pump maximum plunger pressure.

   b. For quintuplex pump, add two-thirds the inlet/suction pressure to the discharge pressure in selecting the pump maximum plunger pressure.

4. For a single acting plunger pump (with <u>optional</u> wrist pin roller/needle bearings), where the *rod load* due to inlet pressure is greater than 20% of full plunger/rod load, no derating is required if <u>both</u> of the following requirements are met;

   a.) *inlet/suction pressure is less than 50% of the maximum plunger pressure rating*

   b.) *discharge pressure is less than the maximum plunger pressure rating.*

V. SUPPLY SYSTEM CONSIDERATIONS

A. Pressure

Liquid pressure is defined as the normal component of force per unit area. In common practice and general function, pressures are frequently measured in pounds force per square inch (lbf/in^2). Gauge pressure (psig) is the difference between absolute pressure (psia) and the atmospheric pressure (P_a). Appendix A, Table 7 page A38 shows the relationship between atmospheric pressure and elevation. Vapor pressure is the absolute pressure exerted by the liquid and its vapor to maintain an equilibrium condition at a given temperature of the liquid.

Example 7:

Find vapor pressures of water at 76° F and 212° F at sea level.

From Appendix E - *Table of Vapor Pressure of Water.*
The vapor pressure of 76°F water is 0.4443 psia at sea level.
The vapor pressure of 212°F water is 14.696 psia at sea level.

B. Head

The English unit for measuring head is feet. The equation, expressing pressure (psi) in units of feet, is:

$$\text{Head} = \frac{\text{psi} \times 2.31}{\text{S.G.}}$$

Where,
S.G. = Specific Gravity @ pumping temperature.

Example 8:

Find the head in units of feet (ft.) of crude oil, with a S.G. = 0.8 @ pumping temperature, at 20 psi pressure.

$$\text{Head} = \frac{20 \times 2.31}{0.8} = 57.75 \text{ ft.}$$

Example 9:

Find the head in units of feet (ft.) of mercury with a S.G. = 13.6 @ pumping temperature), at 20 psi pressure.

$$\text{Head} = \frac{20 \times 2.31}{13.6} = 3.39 \text{ ft.}$$

C. Viscosity

Basic metric viscosity units are the *poise* (absolute/dynamic viscosity) and the *stokes* (kinematic viscosity). More customary expression of these units are *centipoise* and *centistokes* respectively, each equal to 1/100th the of basic metric viscosity unit. The relationship between the English units for medium viscosity liquids, SSU (Saybolt Universal Seconds), and metric absolute viscosity is:

$$n \text{ (absolute viscosity, } \textit{centistokes}) = 0.22\,(SSU) - \frac{180}{(SSU)}$$

Introducing the mass density of the liquid (r) allows the expression of the relationship between absolute viscosity to Kinematic viscosity as follows:

$$m \text{ (Kinematic viscosity, } \textit{centipoise}) = rn$$

<u>Example 10:</u>
Find viscosity in *centistokes* (n) and *centipoise* (m) of a liquid with S.G. = 0.8 with a viscosity of 500 SSU.

$$n = 0.22\,(500) - \frac{180}{500} = 109.64 \textit{ centistokes}$$

Then,

$$m = 0.8 \times 109.64 = 87.71 \textit{ centipoise}$$

The basic pump speed, and its relationship with various ranges of liquid viscosity, is discussed in further detail in Appendix A.

D. Frictional Head Losses

Pipe, valves, fittings, hoses, and meters installed in the liquid supply piping system generate resistance to the liquid flow. The friction head is the hydraulic pressure required to overcome frictional resistance of a piping system. The Table in Appendix C shows an equivalent length in feet, of 100 percent opening valves and fittings. Pressure drop in liquid lines versus liquid flow rates is shown in Appendix D.

E. Reynolds Number

The Reynolds Number (R_e) is used in closed conduit/pipe flow, deals with the viscous force in a liquid, and is defined by the following equation:

$$R_e = \frac{r_1\, d_f\, n_1}{m_1}$$

Where,

R_e	=	Reynolds Number
r_1	=	liquid density at flowing temperature, lbm/ft^3
d_f	=	pipe inside diameter, feet
n_1	=	liquid flow velocity, ft/sec
m_1	=	liquid viscosity (*centipoise* divided by 1488 or *centistokes* multiplied by S.G. then divided by divided by 1488)

Customarily; *turbulent flow* occurs when the R_e is greater than 3000, *laminar flow* occurs when the R_e is less than 2000. The *transition period* is when the R_e is between 2000 and 3000.

Example 11:
A 14" schedule 30 piping system is designed to deliver 18,970 BPD (553.3 GPM) of crude oil with a Kinematic viscosity of 50 *centistokes* and S.G. = 0.8 @ 100°F.

Find the Reynolds Number (R_e).

$$R_e = \frac{r_1 d_f n_1}{m_1} = \frac{49.92 \times 1.104 \times 1.28}{0.02688}$$

$$R_e = 2624.2$$

Where,

$$r_1 = 62.4 \times 0.8 = 49.92 \text{ lbm/ft}^3$$

$$d_f = \frac{13.25}{12} = 1.104 \text{ feet}$$

$$n_1 = \frac{553.3}{2.45\,(13.25)^2} = 1.28 \text{ ft/sec}$$

$$m_1 = \frac{50 \times 0.8}{1488} = 0.02688$$

F. Acceleration Head

Whenever a column of liquid is either accelerated or decelerated, pressure surges exist. This condition is found on the suction/inlet side, as well as discharge side, of a reciprocating pump. Not only can the surges cause vibration in the inlet line, but they can restrict and impede the flow of liquid and cause incomplete filling of the inlet valve chamber. The magnitude of the surges, and how they will react in the system, is impossible to predict without an extremely complex and costly analysis of the system. Since the behavior of the natural frequencies in the system is not easily predicted, as much of the surge as possible must be eliminated at the source. Proper sizing, installation and charging of a dampening device will absorb a large percentage of the surge before it travels into and through the system and cause trouble. The function of the device is to absorb the *peak* of the surge and feed it back at the low part of the cycle. The preferred position for the device is in the liquid supply line, as close as physically possible to the reciprocating pump, or alternately attached to the blind side of the pump inlet. In either location, the surges will be significantly dampened and the possibility of harmful vibrations considerably reduced.

The experimental formula for calculating acceleration head is:

$$h_a = \frac{L\,V\,n\,C}{K\,g} \quad \text{and} \quad V = \frac{GPM}{(2.45)\,(ID)^2}$$

Where,

h_a	=	acceleration head (ft.)
L	=	length of liquid supply line (ft.)
V	=	average velocity in liquid supply line (fps)
n	=	Pump speed (rpm)
C	=	Constant depending on the type of pump (see page A35)
K	=	liquid compressibility factor: K = 2.5 for relatively compressible liquids (ethane, hot oil) K = 2.0 for most other hydrocarbons K = 1.5 for amine, glycol and water K = 1.4 for liquids with almost no compressibility (hot water)
g	=	standard gravity = 32.2 ft/sec^2
ID	=	inside diameter of pipe (in.)

Example 12:

Find the acceleration head (h_a) for a single acting 3-1/2" x 4" triplex plunger pump operating at 350 rpm and 175.35 GPM capacity. Supply to the pump is through 50 feet of 4" Schedule 40 pipe fed from an open tank. Assume the liquid is water @ 80°F.

$$h_a = \frac{(50)\,(175.35)\,(350)\,(0.066)}{(1.4)\,(32.2)\,(4.026)^2\,(2.45)}$$

$$h_a = 113.13 \text{ feet}$$

The acceleration head would be reduced to 49.85 feet if 6" Schedule 40 pipe were used, but this is still too high. An 8" schedule 40 pipe would reduce h_a to 28.78 feet. Proper application of dampener device could reduce this value to approximately 10 feet of 4" schedule 40 pipe. This would appreciably help the net positive suction head available (NPSHA) - in addition to the benefits in avoiding harmful vibrations.

G. NPSHR (Net Positive Suction Head Required)

The NPSHR is the head of liquid, in feet, required at the centerline of the liquid end inlet/suction connection to completely fill each cylinder on the reverse/suction stroke. It is the feet of liquid necessary to;

a) overcome the frictional losses through inlet manifold, valves and liquid chamber,
b) overcome the valve weight and spring force acting on the valve,
c) overcome the valve velocity head losses,
d) accelerate the liquid from rest to required velocity.

NPSHR is a function of the liquid and pump characteristics (stroke, plunger size, liquid end design, and operating speed).

NPSHR is usually determined by test or estimated by computation. Figures 55 and 56, on pages A36 and A37 respectively, show the liquid supply system relationships for *open* and *closed* supply tanks.

H. NPSHA (Net Positive Suction Head Available)

NPSHA is the total inlet head from the system at the pump inlet connection minus the vapor pressure of the liquid at the pumping temperature. It is a function of the supply system **and** the liquid and pump characteristics (stroke, plunger size, liquid end design, operating speed).

The pump characteristic is important to both supply system and NPSHR. ***A liquid supply system designed without consideration of the pump characteristics has not been properly designed.***

I. Pump Cavitation

A *WGO* reciprocating power pump is a device to move liquid under specified operating conditions. If the liquid is not arriving at the inlet side of the pump promptly, evenly, and with the least amount of resistance; the pump cannot operate efficiently, nor can it move the liquid through the discharge system smoothly. A poor liquid supply system to the inlet side of the pump will create pump *cavitation* (where liquid moving through the pump vaporizes rapidly wherever the local absolute pressure falls to, or attempts to fall below, the liquid vapor pressure). *Cavitation* can cause premature failure of the pump valve, piston or plunger packing, pitting of the cylinder walls, and damage to the pump and system by subjecting all parts to undue stresses. Consider the interactions between a piston and the liquid it pumps. The piston must stay in contact with the liquid through its entire stroke.

For reciprocating power pumps, plunger/piston velocity varies sinusoidally with crankshaft position. Maximum velocity occurs approximately at mid-stroke and zero velocity occurs at both ends (full forward position and full reverse position) of the stroke. Under certain inlet conditions, liquid looses contact with the piston and creates a *cavity*. When the piston slows, and the liquid catches up with the piston, collapsing of the cavity will occur. This *cavitation* creates shock waves that travel throughout the pump and pumping system, generating noise, vibration and wear.

Even if velocity-matching requirements are met, pressure at the pump inlet must be high enough to prevent gas formation/separation.

J. Liquid Supply System Relationships

1. Open Supply - Elevated Inlet Situation (also see page A36).

If atmospheric pressure available at site is greater than the equation ($NPSHR + P_v + h_f + h_a$), it is possible to install the pump inlet above the level of the liquid. The maximum distance ($l_{e(max)}$) the pump inlet can be placed above the liquid level is determined by the following equation converted to units of feet:

$$l_{e(max)} = P_a - (NPSHR + P_v + h_f + h_a)$$

Where,

l_e	=	elevation distance from the center line of the pump inlet connection to the liquid level, feet
$l_{e(max)}$	=	maximum inlet elevation distance, feet
P_a	=	atmospheric pressure available at site
NPSHR	=	net positive suction head required (specified by pump manufacturer)
P_v	=	absolute vapor pressure (@ pumping temperature) **plus 7 feet**
h_f	=	friction losses through pipe and fittings, feet
h_a	=	acceleration head, feet

To convert psi (pounds per square inch) to feet of head, multiply psi by 2.31 then divide by specific gravity (S.G.).

$$\text{feet of head} = \frac{\text{psi x 2.31}}{\text{S.G.}}$$

If it is found that the actual distance which the pump may be installed above the liquid level is less than the maximum calculated distance, then NPSHA is equal to the equation NPSHA - P_a - (P_v + h_f + h_a + l_e), converted to feet.

In order to find whether the pump can be installed above the liquid level, a comparison of P_a and the evaluation of the equation (NPSHR + P_v + h_f + h_a) is necessary. If P_a is found to be the greater, the pump may be placed above the liquid level.

Example 13:

A single acting 1-3/4" x 1-1/2" triplex plunger pump is operating under the following conditions:

a. the pump speed (n) is 200 rpm
b. 9.18 GPM pump capacity (Q), barrels per day (BPD) = GPM x 34.3
c. the volumetric efficiency (E_v) is 100%
d. pumping fresh water @ 120 °F, specific gravity (S.G.) = 1.0
e. NPSHR specified by manufacturer is 6 feet
f. the entire system installation is at 9000 ft. altitude
g. the supply tank is open to atmosphere
h. the pump inlet connection is 1-1/2" diameter
i. inlet supply system consists of 10 feet of 2" schedule pipe and a single 2" x 1-1/2" eccentric pipe reducer.

Problem: 1. Can this pump be installed above the liquid level?
2. If yes, find $l_{e(max)}$?

Finding values for the above mentioned equations, we ascertain the following:

P_a = atmospheric pressure = 10.5 psia (from Table 7 Appendix A page A36). To convert to feet; multiply by 2.31 then divide by S.G.
P_a = 24.25 feet
P_v = water vapor pressure = 1.6924 psia (from Appendix E). To convert to feet; multiply by 2.31 then divide by S.G.
= 3.9 feet + **7 feet**
P_v = 10.9 feet

h_f = friction losses through pipe and fittings
2" schedule pipe = 0.155 psi /100 feet (from Appendix D).
eccentric reducer = creates a friction loss in an equivalent length of 1 foot of 2" pipe (see Appendix C)

Therefore,

$$h_f\ (2) = \frac{0.155}{100}\ (10 + 1)\ \text{psi}$$

$$= 0.017\ (2.31)\ \text{feet}$$

$$h_f\ (2) = 0.039\ \text{feet}$$

$$h_a = \text{acceleration head} = \frac{L\,V\,n\,C}{K\,g}$$

Where,

L = length of liquid supply system piping = 10 feet

V = average liquid velocity through the pipe

$$= \frac{GPM}{2.45\ (D)^2} = \frac{9.18}{2.45\ \ (1.939)^2} = 0.996\ \text{fps}$$

n = pump speed = 200 rpm

C = empirical constant for triplex (single acting or double acting) = 0.066

K = liquid compressibility factor (water) = 1.5

g = standard gravity = 32.2 ft/sec^2

Substituting into the equation for acceleration head,

$$h_a = 2.72\ \text{feet}$$

Substituting these values into the equation
$(NPSHR + P_v + h_f + h_a) = (6 + 10.9 + 0.039 + 2.72)$ feet = 19.66 feet

The atmospheric pressure @ 9000 feet altitude is 24.25 feet and is greater than the summation of the equation above. Therefore, ***the pump can be installed above the liquid level***.

Find $l_{e(max)}$

$l_{e(max)}$ = $P_a - (NPSHR + P_v + h_f + h_a)$ = 24.25 - 19.66

$l_{e(max)}$ = 4.59 feet

Example 14:
Problem: Calculate NPSHA for the above conditions, if l_e = 2 feet.

Solution: The equation for Net Positive Suction Head Required is
NPSHA = $P_a - (P_v + h_f + h_a + l_e)$ = 24.25 - (13.66 + 2)

NPSHA = 8.59 feet

2. Open Supply - Submerged Inlet Situation (also see Page A36)

If atmospheric pressure (P_a) is less than the equation ($NPSHR + P_v + h_f + h_a$), *the pump can not be installed above the liquid level.* With this situation, a positive static inlet head or a properly sized charging pump is necessary. The charging pump must have more capacity than the reciprocating pump to avoid cavitation from insufficient liquid available. If a charging pump is the only solution, it is usually best to have *WGO* specify what size and type of charging pump to use.

Keep in mind that, as with any pumping issue, in order for *WGO* to give you the proper information; we need ***all*** details of your specific pumping application.

If a pump inlet static head is possible, the minimum static head required ($H_{i(min)}$) is calculated as follows:

$H_{i(min)}$ = $(NPSHR + P_v + h_f + h_a) - P_a$

Where,

$H_{i(min)}$	=	minimum static head required
NPSHR	=	net positive suction head required (specified by pump manufacturer)
P_a	=	atmospheric pressure available at site elevation
P_v	=	absolute vapor pressure **plus 7 feet** (at pumping temperature)
h_f	=	friction losses through pipe, fittings, etc., feet
h_a	=	acceleration head, feet

Example 15:

A single acting 3" x 5" quintuplex plunger pump operating under the following conditions:

a. pump speed (n) = 300 rpm
b. 229.5 GPM pump capacity (Q) = 229.5 GPM, BPD = 7,872
c. pump volumetric efficiency (E_v) = 100%
d. pumping fresh water @ 70° F with S.G. = 1
e. NPSHR specified by pump manufacturer is 15 feet
f. the entire system is at 5000 ft. altitude
g. the supply tank is open to atmosphere
h. the pump inlet connection is 6" diameter
i. liquid supply piping system is composed of:
   - 10 ft. 8" schedule 40 pipe
   - 1 ea. 8" gate valve
   - 2 ea. 8" long radius ells
   - 15 ft. 6" schedule 40 pipe
   - 1 ea. 8" x 6" eccentric reducer

Problem: 1. Can this pump be installed above the liquid level?
2. If not, find minimum static head required.

Values for the equation, $H_{i(min)} = (NPSHR + P_v + h_f + h_a) - P_a$, are found as follows;

P_a = atmospheric pressure (12.2 psia, Table 7, Appendix A)
To convert to feet, multiply by 2.31 then divide by S.G.

P_a = 28.18 feet

P_v = water vapor pressure (0.3631 psia) **plus 7 feet**
= to convert to feet, multiply by 2.31 then divide by S.G. + **7 feet** = (0.83 + 7) feet

P_v = 7.83 feet

NPSHR = 15 feet (specified by *WGO*)

h_f = $h_f(8) + h_f(6)$

Description	equivalent length (Appendix C)
8" gate valve	6 feet
8" long radius ell	18 feet (9 ft./ea. x 2 ea.)
8" pipe	10 feet
total equivalent feet	34 feet

The pressure drop through 8" pipe is 0.05 psi/100 feet (see Appendix D) and for 34 equivalent feet is

$$h_f(8) = \frac{0.05}{100} \times 34 = 0.017 \text{ psi}$$

To convert $h_f(8)$ to feet, multiply by 2.31 then divide by S.G.

$$h_f(8) = \frac{0.017 \times 2.31}{1} = 0.039 \text{ feet}$$

description	equivalent length (Appendix C)
6" gate valve	15 feet
8" x 6" eccentric reducer	4 feet
total equivalent feet	19 feet

The pressure drop through 6" pipe is 0.17 psi/100 feet (see Appendix D) and for 19 equivalent feet is

$$h_f(6) = \frac{0.17}{100} \times 19 = 0.032 \text{ psi}$$

To convert $h_f(6)$ to feet, multiply by 2.31 then divide by S.G.:

$$h_f(6) = \frac{0.032 \times 2.31}{1} = 0.074 \text{ feet}$$

Therefore,

$$h_f = h_f(8) + h_f(6) = (0.039 + 0.074) \text{ feet}$$

$$h_f = 0.133 \text{ feet}$$

$$h_a = \text{acceleration head} = \frac{L\,V\,n\,C}{K\,g}$$

In this case, we have two different h_a values; h_a (8) for 8" pipe and h_a (6) for 6" pipe.

Where:

g	=	32.2 ft./sec^2		
L	=	actual length of liquid supply system piping, feet		
L(6)	=	15 feet		
L(8)	=	10 feet		
V	=	avg. fluid velocity through pipe, fps	=	$\frac{GPM}{(ID)^2 \times 2.45}$
n	=	pump speed	=	300 rpm
C	=	empirical constant for quintuplex	=	0.040
K	=	liquid compressibility factor (water)	=	1.5

Find h_a (8),

$$V(8) = \frac{229.5}{(7.981)^2\ 2.45} = 1.47 \text{ fps}$$

$$h_a(8) = \frac{L(8)\ V(8)\ n\ C}{K\ g}$$

$$h_a(8) = \frac{(10) \times (1.47) \times (300) \times (0.04)}{(1.5) \times (32.2)} = 3.65 \text{ feet}$$

Find h_a (6),

$$V(6) = \frac{229.5}{(6.065)^2\ 2.45} = 2.55 \text{ fps}$$

$$h_a(6) = \frac{L(6)\ V(6)\ n\ C}{K\ g}$$

$$h_a(6) = \frac{(15) \times (2.55) \times (300) \times (0.04)}{(1.5) \times (32.2)} = 9.50 \text{ feet}$$

Therefore,

$$h_a = h_a(8) + h_a(6) = 13.15 \text{ feet}$$

Using these values, we can now solve the equation
$(NPSHR + P_v + h_f + h_a) = (15 + 7.83 + 0.11 + 13.15) = 36.09$

Solution: 1. Since P_a (28.18 feet) is less than the above equation, ***the pump cannot be installed above the liquid level***.

Solution: 2. The minimum positive static inlet head required is
$H_{i(min)} = 36.09 - 28.18 = 7.91$ feet

If actual inlet static head ($H_{i(act)}$) is greater than the minimum inlet static head required ($H_{i(min)}$), the formula used for calculating NPSHA is

$$NPSHA = (H_{i(act)} + P_a) - (P_v + h_f + h_a)$$

Problem: For above conditions, if $H_{i(act)}$ is 30 feet, find NPSHA

Solution:
$NPSHA = (30 + 28.18) - (7.83 + 0.11 + 13.15)$
$NPSHA = 37.09$ feet

3. Closed Supply Vessel (also see page A37)

When absolute pressure at source is equal to the absolute liquid vapor pressure (P_{source} = P_v), the minimum inlet static head required ($H_{i(min)}$) must equal or exceed the sum or all the losses per the following equation:

$$H_{i(min)} = h_f + h_a + NPSHR$$

Example 16:

A double acting 7" x 10" duplex piston pump is operating under the following conditions:

a. pump crankshaft speed (n) = 95 rpm
b. pump capacity (Q) = 553.4 GPM (BPD = 18,980)
c. pump volumetric efficiency (%)
d. pumping crude oil, S.G. = 0.8 @ 100 *centistokes* viscosity
e. NPSHR specified by the pump manufacturer is 12 feet of crude oil
f. the entire system is at sea level
g. the supply tank is a **closed supply vessel** with P_{source} = P_v
h. pump inlet connection is 6"
i. the liquid supply system piping consists of:
   55 ft. 14" schedule 30 pipe
   1 ea. 14" gate valve
   3 ea. 14" long radius ells
   1 ea. 14" x 6" eccentric reducer

Problem: Find the minimum inlet static head required, $H_{i(min)}$,

$$H = h_f + h_a + NPSHR$$

Where,

NPSHR = Net Positive Suction Head Required (specified by pump manufacturer) is 12 feet

h_f = friction losses through liquid supply system pipe and fittings (for this condition there are two different h_f values; h_f (14) for 14" pipe and h_f(6) for 6" pipe.

Therefore, $h_f = h_f(14) + h_f(6)$

Description	equivalent feet (Appendix C)
14" gate valve	10
14" long radius ell (16 ft/ea x 3 ea)	48
14" schedule 30 pipe	55
total equivalent length	113

The pressure drop through 14" pipe is 0.02 psi/100 feet (see Appendix D) and for 113 equivalent feet is

$$h_f(14) = \frac{0.02}{100} \times 113 = 0.023 \text{ psi}$$

Correcting for the 100 *centistokes* Kinematic viscosity (see Appendix D)

$$h_f(14) = 0.023 \times 3.5 = 0.08 \text{ psi}$$

To convert h_f (14) to feet, multiply by 2.31 then divide by S.G:

$$h_f(14) = \frac{0.08 \times 2.31}{0.8} = 0.23 \text{ feet}$$

$h_f(6)$ = equivalent length of the 14" x 6" eccentric reducer in 6" pipe (see Appendix C) is

$h_f(6)$ = 7 feet

Pressure drop through 6" pipe is 0.83 psi / 100 feet (see Appendix D) and for 7 equivalent feet is

$$h_f(6) = \frac{0.83}{100} \times 7 = 0.058 \text{ psi}$$

Correcting for the 100 *centistokes* Kinematic viscosity (see Appendix D)

$$h_f(6) = 0.058 \times 3.5 = 0.20 \text{ psi}$$

To convert h_f (6) to feet, multiply by 2.31 then divide by S.G:

$$h_f(6) = \frac{0.20 \times 2.31}{0.8} = 0.57 \text{ feet}$$

Substituting in for $h_f(14)$ and $h_f(6)$

$$h_f = h_f(14) + h_f(6) = (0.23 + 0.57) \text{ feet}$$

$$h_f = 0.80 \text{ feet}$$

$$V = \frac{GPM}{(ID)^2\ 2.45} = \frac{53.4}{(13.25)^2\ 2.45}$$

$$V = 1.29 \text{ fps}$$

$$h_a = \text{acceleration head} = \frac{L\ V\ n\ C}{K\ g}$$

Where,

g	=	standard gravity	=	32.2 ft/sec^2
K	=	liquid compressibility factor (oil)	=	2
C	=	empirical constant for duplex	=	0.115
n	=	pump speed	=	85 rpm
L	=	actual pipe length	=	55 feet

$$h_a = \frac{(55)\ (1.29)\ (85)\ (0.115)}{(2)\ (32.2)} = 10.77 \text{ feet}$$

Substituting these values into the equation for minimum static head

$$H_{i(min)} = h_f + h_a + NPSHR = 0.8 + 10.77 + 12$$

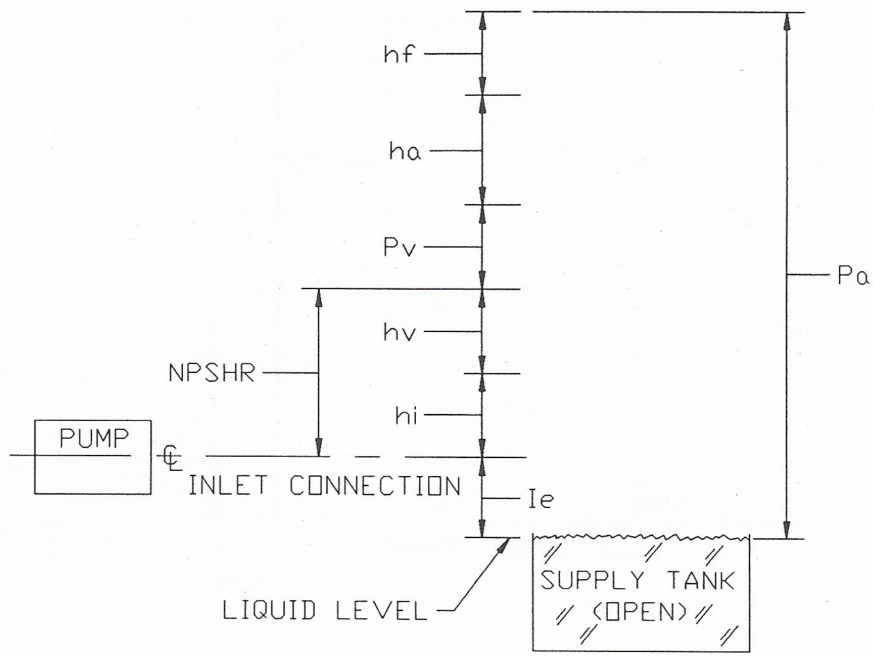
$$H_{i(min)} = 23.57 \text{ feet}$$

At the above conditions, with actual inlet static head ($H_{i(act)}$) of 25 feet, find the Net Positive Suction Head Available

$$NPSHA = H_{i(act)} - (h_f + h_a) = 25 - (0.8 + 10.77)$$

$$NPSHA = 13.43 \text{ feet}$$

Open Supply - Elevated Inlet System

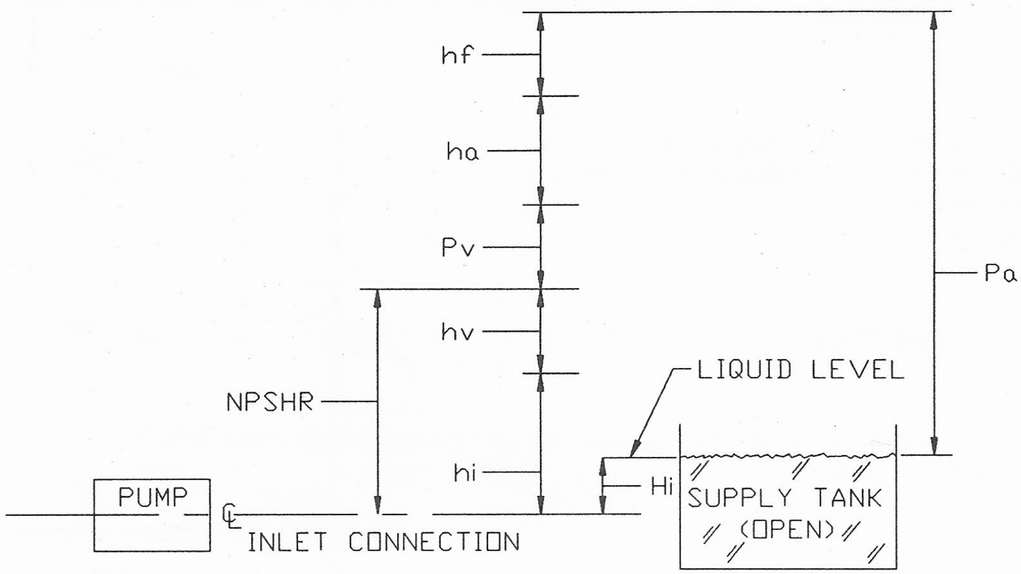


P_a	=	atmospheric pressure available at site
h_f	=	supply system frictional losses (pipe and fittings)
h_a	=	supply system acceleration head
P_v	=	absolute vapor pressure **plus 7 feet** (at pumping temperature)
h_v	=	velocity head through valve
h_i	=	head of valve spring, valve weight, liquid friction due to viscosity, elevation from inlet center line to center line of piston or plunger
NPSHR	=	Net Positive Suction Head Required (specified by pump manufacturer)
I_e	=	inlet elevation

1. If P_a > (NPSHR + P_v + h_a + h_f) then, the pump can be installed above the liquid level and maximum elevated inlet distance
   $I_{e(max)} = P_a$ - (NPSHR + P_v + h_f + h_a)

2. If actual elevated inlet distance $I_e < I_{e(max)}$, then the available net positive inlet head is NPSHA = P_a - (P_v + h_f + h_a + I_e)

Open Supply - Submerged Inlet System

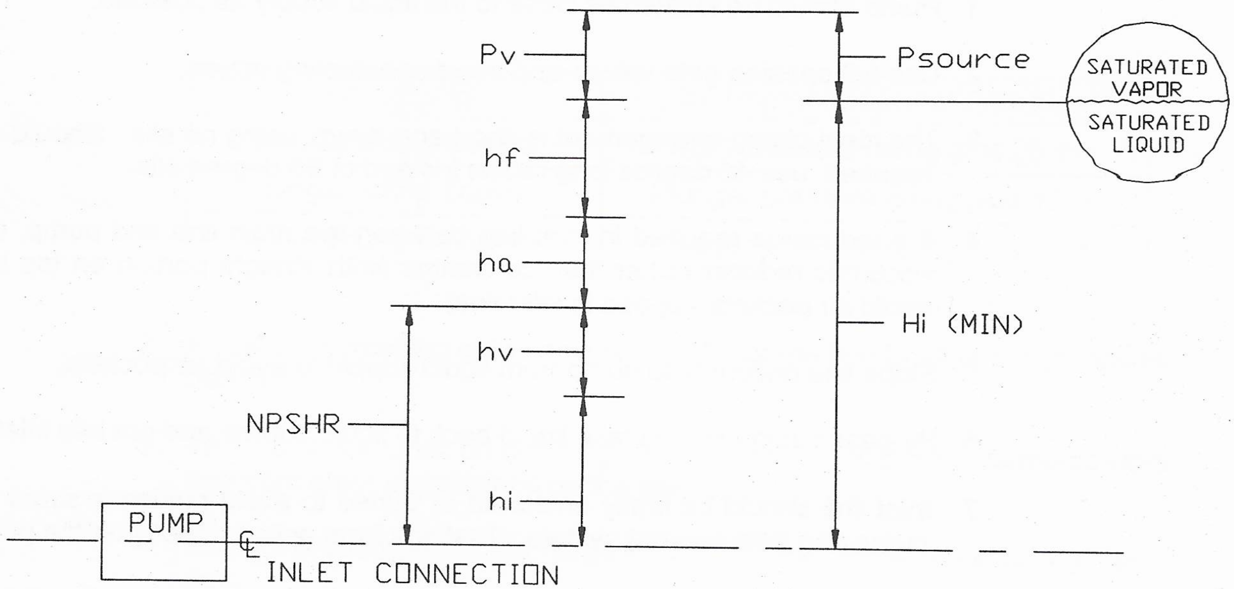


P_a	=	atmospheric pressure available at site.
h_f	=	supply system frictional losses (pipe and fittings)
h_a	=	supply system acceleration head
P_v	=	absolute vapor pressure **plus 7 feet** (at pumping temperature)
h_v	=	velocity head (through valve)
h_i	=	head of valve spring, valve weight, liquid friction due to viscosity, elevation from inlet center line to center line of piston or plunger
NPSHR	=	Net Positive Suction Head Required (specified by pump manufacturer)
H_i	=	static inlet head required

1. If P_a < (NPSHR + P_v + h_a + h_f) then, the pump can be installed above the liquid level and the static inlet head required is H_i = (NPSHR + P_v + h_f + h_a) - P_a

2. If actual static inlet head exceeds minimum required static inlet head ($H_{i(act)} > H_{i(min)}$) then NPSHA = ($H_{i(act)}$ + P_a) - (P_v + h_f + h_a)

Closed Supply System



P_{source}	=	pressure of closed vessel from liquid source
$H_{i(min)}$	=	minimum static head required
h_a	=	supply system acceleration head
P_v	=	absolute vapor pressure **plus 7 feet** (at pumping temperature)
h_v	=	velocity head through valve
h_i	=	head of valve spring, valve weight, liquid friction due to viscosity, elevation from inlet center line to center line of piston or plunger
NPSHR	=	Net Positive Suction Head Required (specified by pump manufacturer)

1. When absolute pressure at source equals the absolute liquid vapor pressure
   $P_{source} = P_v$
   then minimum static inlet head must equal, or exceed, the sum of losses
   $H_{i(min)} = h_f + h_a + \text{NPSHR}$

2. If actual static inlet head exceeds the minimum inlet head $H_{i(act)} > H_{i(min)}$ then
   $\text{NPSHA} = (H_{i(act)} - (h_f + h_a)$

VI. INSTALLATION SUGGESTIONS (see Figure 52 Appendix A page A29, *Suggested Piping Systems for WGO Power Pumps*)

A. Supply System Piping Design

1. Pump should be located as close to the liquid supply as possible.

2. Use full opening gate valves and avoid constricting valves.

3. The ideal piping arrangement is short and direct, using no ells. Should ells be required, use 45 degree long radius instead of 90 degree ells.

4. If a reducer is required in inlet line between the main line and pump, use an eccentric reducer rather than concentric (with straight portion on top to help avoid air pockets trapped in inlet line).

5. Slope line uniformly to pump from liquid supply to avoid air pockets.

6. By-pass design should take liquid back to liquid source and not into inlet line.

7. Inlet line should be firmly anchored or buried to avoid putting a strain on the pump and help prevent system vibrations from acting directly on the pump.

8. For inlet lines leading directly to the pump, select a size line so that the velocity of the liquid will not exceed one foot per second, or one that is two sizes larger than the pump inlet connection, whichever gives the lower line velocity.

9. Install a properly sized charging pump in the inlet line as close to the pump as possible.

10. Install a full opening gate valve on the inlet line to allow for pump maintenance.

11. Inlet PCT (pulsation control tool).

B. Discharge Piping Design

1. The insertion of a *pressure relief valve* (PRV) of suitably sized for the full pump capacity and set fully open at a pressure *above* the required operating discharge pressure of the pump, is mandatory for safe operation.

The PRV should be placed in the discharge line, as close as possible to the pump and before any other valves. The PRV outlet should be piped back to the liquid *source*.

Suggested *trial* set pressures of pump PRV's:

Type Pump	Recommended Relief Pressure
Double acting Duplex	Piston pressure rating plus 25 %
Single acting Triplex	Plunger pressure rating plus 10 %
Single acting Quintuplex	Plunger pressure rating plus 10 %

2. Install a full opening gate valve on the discharge line to allow pump maintenance.

3. Install a check valve on discharge line to protect pump from harmonic vibration from other installations on the line.

4. Install a properly sized discharge PCT to reduce vibration and piping damage.

5. Securely anchor discharge lines.

6. When possible, run discharge lines straight for at least ten feet before bending.

7. Install a by-pass valve to aid in pump start-up.

8. Use 45 degree long radius ells.

C. Recommended Safety Devices

1. Hi-Lo oil level control & oil temperature gauge.

2. High temperature control.

3. Oil pressure control.

4. Low liquid level control (to control the NPSHA).

5. Low discharge pressure shut-down.

6. High discharge pressure shut-down.

7. Pre/Post lube of force feed power end lubrication system.

VII. TROUBLE LOCATING OF RECIPROCATING PUMPS

Trouble	Possible Causes
Low Volumetric Efficiency (*failure to deliver rated capacity and pressure*)	1) Air or vapor pocket in inlet line 2) Capacity of charge pump less than capacity of power pump 3) Air or vapor trapped in or above inlet manifold 4) Air leak in liquid supply piping system 5) Loose bolts in pump inlet manifold 6) Air or gases entrained in liquid 7) Foreign object holding pump inlet or discharge valve(s) open 8) Incorrect drive ratio 9) Loose belts 10) Incorrect motor or engine speed 11) Loose valve covers or cylinder head 12) Worn valves and seats 13) Safety relief valve partially open, or not holding pressure 14) Worn liners, piston rings or plungers 15) Bypass valve open, or not holding pressure 16) Blown liner gasket 17) NPSHA not sufficient 18) Liquid bypassing internally 19) Foreign object blocking liquid passage 20) Vortex in supply tank 21) Insufficient power delivered by motor
NPSHA Too Low	1) Inlet line partially clogged 2) Liquid vapor pressure too high 3) Liquid pumping temperature too high 4) Restricted inlet pipe fittings 5) Inlet line too long 6) Too many pipe fittings 7) Too small inlet line 8) Too low static inlet head 9) Too low atmospheric pressure
Liquid Not Delivered	1) Pump not primed 2) Air or vapor pocket in inlet line 3) Clogged inlet line 4) All inlet valves propped open 5) All discharge valves propped open 6) Loose bolts in pump inlet manifold 7) Too high valve velocities

Trouble	Possible Causes
Cavitation	1) NPSHA too low 2) Liquid NOT Delivered to Pump Inlet Connection 3) Excessive Stuffing Box Leakage 4) NPSHR too high
Cylinder Head or Valve Cover Leaks	1) Over Recommended Pressure 2) Loose Cylinder Head/Valve Cover 3) Damaged Gasket/O-ring
Water in Crankcase/Oil	1) Water Condensation 2) Worn seals 3) Clogged Air Breather(s) 4) Worn Crankcase Packing 5) Loose Covers
Crankcase Oil Leaks	1) Oil Level/Temperature Too High 2) Worn seals 3) Worn Crankcase Packing 4) Loose Crankcase Cover
Power End Overheating	1) Pump Running Backward/RPM too low 2) Insufficient Oil in Power End 3) Excessive Oil in Power End 4) Incorrect Oil Viscosity 5) Operating Pump above Recommended Pressure 6) Main Bearings too Tight 7) Drive Misaligned 8) Belts too Tight 9) Discharge Valve, one or more, Stuck Open 10) Insufficient Cooling 11) Pump RPM too Low 12) Inadequate Ventilation 13) Liquid End Packing Adjusted too Tight (adjustable style packing only)
Pump Driver Overload	1) Pump RPM too High 2) Low Voltage or other Electrical Trouble 3) Trouble with Engine, Turbine, Gear Reducer or other Related Equipment 4) Excessive Discharge Line Pressure 5) Clogged Discharge Line 6) Closed/Throttled Valve in Discharge Line 7) Incorrect Plunger/Piston Size for Application 8) Improper Bypass Conditions 9) Over-tightened Stuffing Box Glands on Adjustable Packing

Trouble	Possible Causes
Stuffing Box Leaks	1) Worn Packing 2) Worn rods or plunger 3) Worn stuffing boxes 4) Wrong size packing 5) Worn O-ring seal (replaceable boxes)
Stud Failure	1) Excessive discharge pressure 2) Improper torquing of nuts 3) Shock overload caused by pump cavitation
Excessive Valve Noise	1) Broken or weak valve spring 2) Pump cavitation 3) Air leak in inlet piping or loose bolts in pump inlet manifold 4) Air trapped above inlet valve
Suction Line or Discharge Line Vibration	1) Piping inadequately supported 2) Inlet line too long or too small in diameter 3) Too many bends in inlet line 4) Multiple pump installations operating in phase 5) Obstruction Under Valve(s) 6) Packing Worn 7) Operating Above Recommended Pressure or RPM 8) Low NPSHA
Noisy Operation *(Be sure to differentiate between liquid knock and mechanical knock - very few knocks are mechanical on new installations.)*	1) Piston or plunger loose 2) Valve noise amplified through power end 3) Pump cavitation 4) Liquid knock 5) Air leak in inlet piping 6) Loose bolts in pump inlet manifold 7) Hydraulic noise in liquid end 8) Loose or worn crosshead pins and bushings 9) Loose connecting rod cap bolt 10) Worn connecting rod bearings 11) Worn crosshead 12) Main bearing end play excessive 13) Worn gears or chains 14) Gears or chains out of line 15) Pump running backward 16) Partial loss of prime 17) Shocks in piping system 18) Water in power end crankcase 19) Poorly supported piping, abrupt turns in piping, piping misaligned, pipe size too small

Trouble	Possible Causes
Broken shafts, bent or stripped threads and *other catastrophic failures*	1) Start-up against closed gate valve in discharge line. If valve seats are discovered driven too deeply after operation of the pump, look for the following pattern of driven seats, indicative of start-up or run against a closed discharge line valve: *Triplex* single-acting plunger pump: 2 inlets and 1 discharge valve seat, or 1 inlet and 2 discharge valve seats *Quintuplex* single-acting plunger pump: 3 inlets and 2 discharge valve seats, or 2 inlets and 3 discharge valve seats *Duplex* double-acting piston pumps: 2 inlets and 2 discharge valve seats 2) Low oil level 3) Contaminated oil 4) Main bearing failure 5) Piston or plunger striking cylinder head 6) Disintegration of worn valves 7) Frozen liquid in liquid body 8) Air leak in liquid supply system 9) Loose bolts in pump inlet manifold
Packing Failure	1) Normal wear 2) Improper material 3) Improper lubrication 4) Adjustable packing - gland tightened excessively 5) Dirty liquid 6) Plunger or piston rod misalignment 7) Dirty environment
Valve Failure	1) Normal wear 2) Pump cavitation 3) Abrasives in fluid 4) Incompatibility of valve components to corrosive liquid 5) Electrolysis 6) Incorrect installation – driving on the valve stem, improper torque on jam nut, valve seat and valve deck not thoroughly clean and dry when seat installed.
Plunger Failure	1) Thermal shock (cold liquid hitting hot ceramic plunger) 2) Packing too tight 3) Inlet valve dislocated/disassembled during pump operation 4) Stuffing box gland rubbing on plunger due to improper tightening procedure 5) Dirty liquid 6) Dirty environment

Appendix A

*1

reciprocating pumps

POWER PUMPS

reciprocating pumps, power
types

POWER PUMP

Power Pump

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

Horizontal Pump

The axial centerline of the cylinder is horizontal.

Vertical Pump

The axial centerline of the cylinder is vertical.

Single-Acting Pump

Liquid is discharged only during the forward stroke of the plunger or piston, that is, during one half of the revolution.

Double-Acting Pump

Liquid is discharged during both the forward and return strokes of the plungers or piston, that is, discharge takes place during the entire revolution.

Piston Pump

The liquid end contains pistons.

Plunger Pump

The liquid end contains plungers.

Simplex Pump

Contains one piston or its equivalent, that is, single or double-acting plunger(s).

Duplex Pump

Contains two pistons or their equivalent, that is, single or double-acting plungers.

Multiplex Pump

Contains more than two pistons or their equivalent, that is, single or double-acting plungers.

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

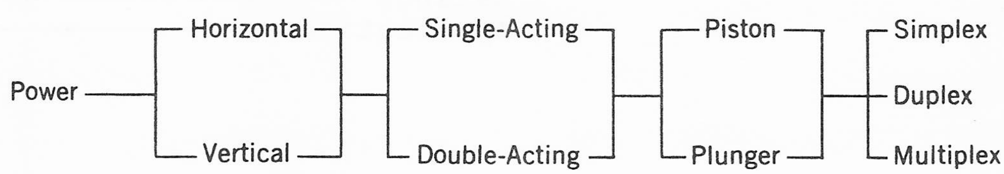


Fig. 2 TYPES OF POWER PUMPS

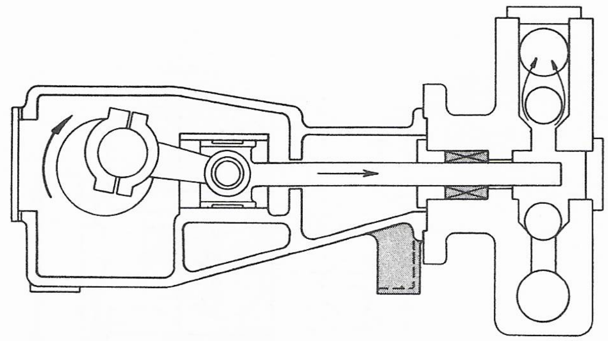


Fig. 3 HORIZONTAL SINGLE-ACTING PLUNGER POWER PUMP

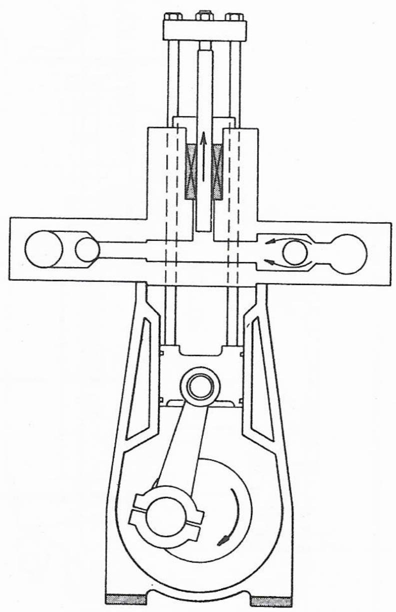


Fig. 4 VERTICAL SINGLE-ACTING PLUNGER POWER PUMP

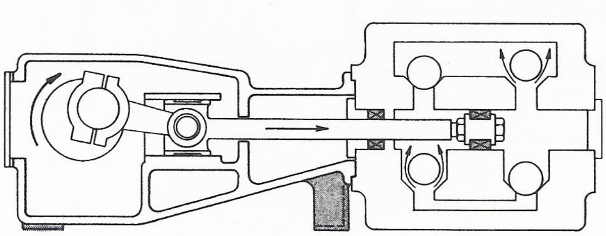


Fig. 5 HORIZONTAL DOUBLE-ACTING PISTON POWER PUMP

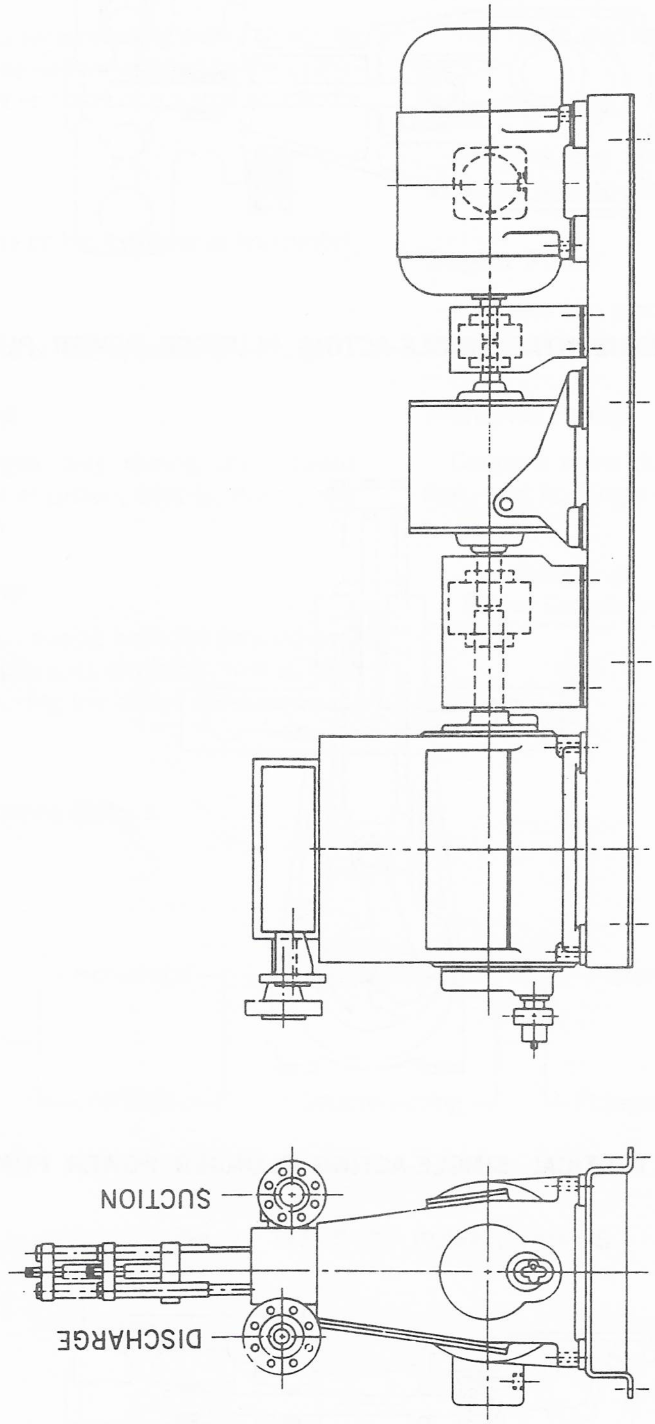


Fig. 6 VERTICAL TRIPLEX PLUNGER PUMP, ON BASE, GEAR REDUCTION

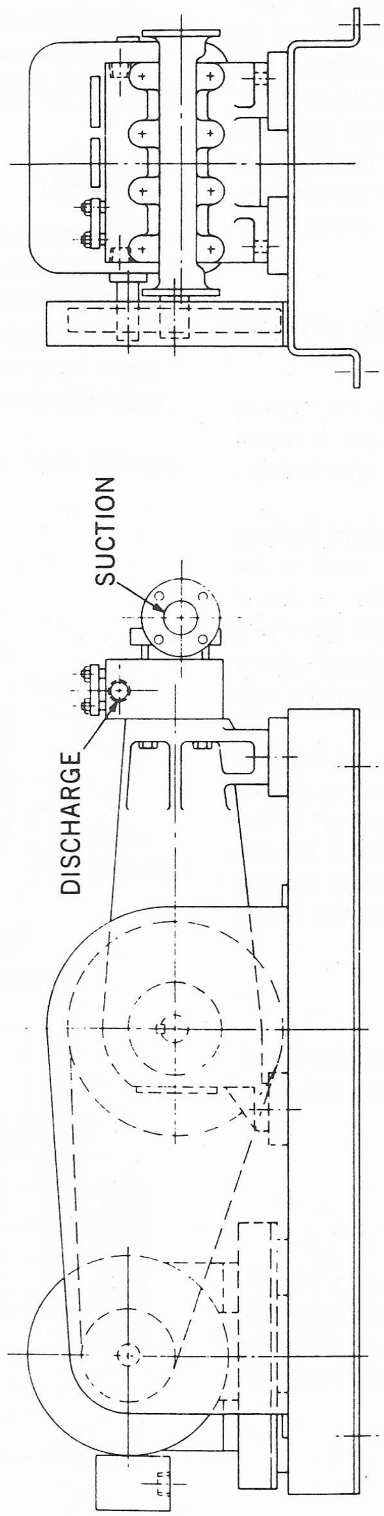


Fig. 7 HORIZONTAL TRIPLEX PLUNGER PUMP, ON BASE, BELT DRIVE

reciprocating pumps, power
nomenclature

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 power pump components but do not necessarily represent recommended designs. Variations in design may exist without violating the intent of these Standards.

Right and Left Hand Shaft Extension of Power Pumps

"Right " or "left hand" designates the side of the power end from which the crankshaft or 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.

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.

LIQUID END

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

LIQUID END PARTS

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. (Fig. 8.)

Cylinder Liner

A replaceable liner which is placed in the cylinder of a piston pump. The piston reciprocates within the liner. (See Figs. 27 and 28.)

Manifolds

A suction manifold is a chamber which accepts liquid from the suction port(s) and distributes it to the suction valves. (Fig. 9.)

A discharge manifold is a chamber which accepts liquid from the individual discharge valves and directs it to the discharge port(s). (Fig. 9.)

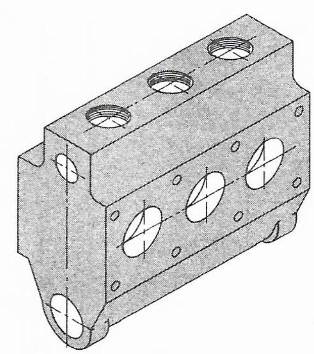


Fig. 8 LIQUID CYLINDER

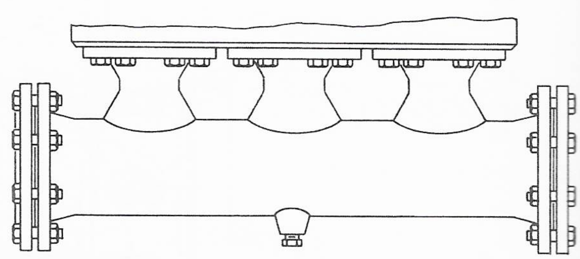


Fig. 9 MANIFOLD

Valve Chest Cover

A cover for the valves within the cylinder. (Fig. 28.)

Valve Plate (Valve Deck)

A plate that contains the suction or discharge valves. (Fig. 28.)

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.

A piston in a reciprocating pump is usually double-acting.

The pistons in Figs. 10 and 11 have followers which retain the packing rings.

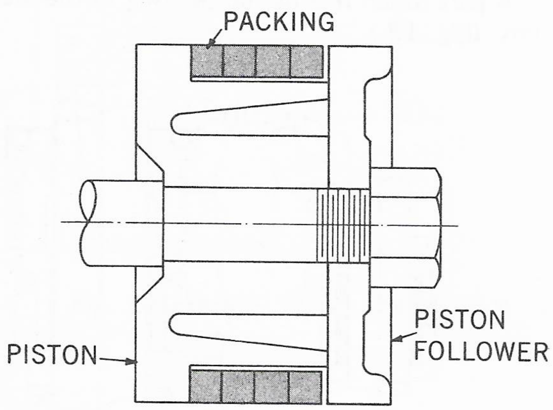


Fig. 10 PISTON ASSEMBLY

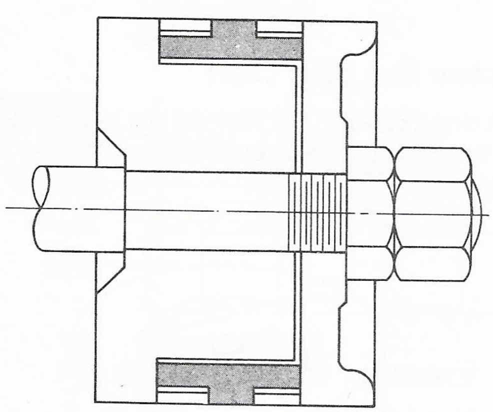


Fig. 11 BULL & SNAP RING PISTON

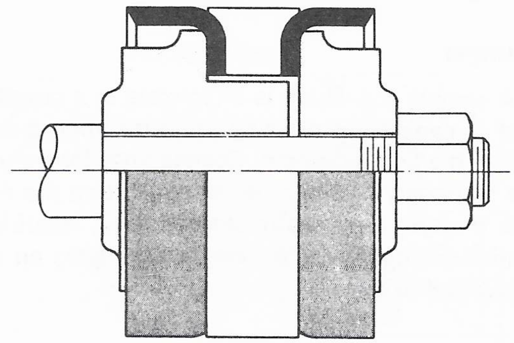


Fig. 12 CUP TYPE PISTON

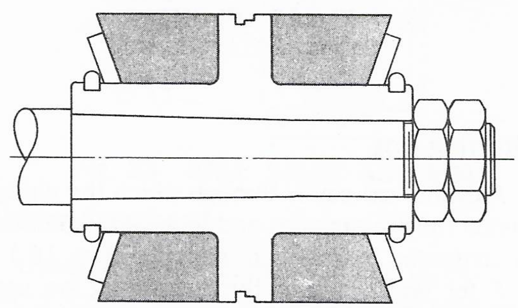


Fig. 13 SLUSH PISTON

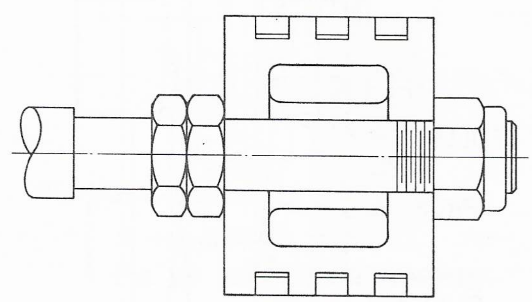


Fig. 14 INDIVIDUAL RING PISTON

Plunger

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

A plunger is normally single-acting, requiring a double-acting pump to have two plungers on each crosshead axis.

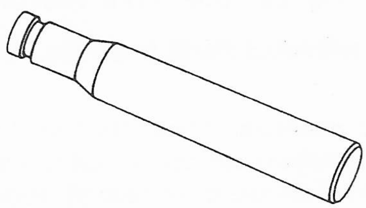


Fig. 15 PLUNGER

Stuffing Box

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

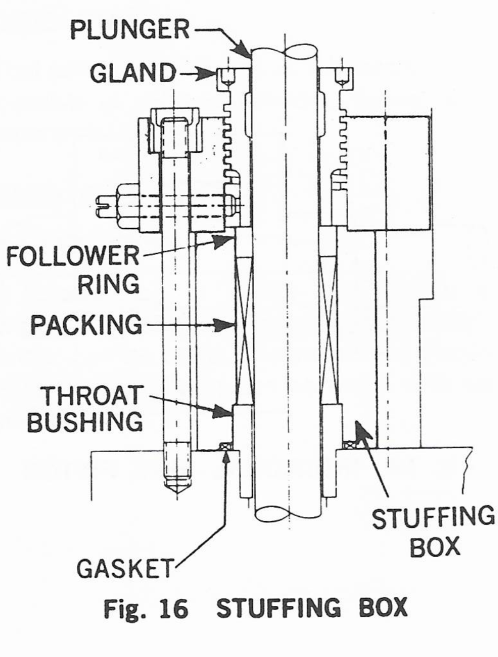


Fig. 16 STUFFING BOX

Packing

A material used to provide a seal around the plunger, piston rod, or piston. (Fig. 17.)

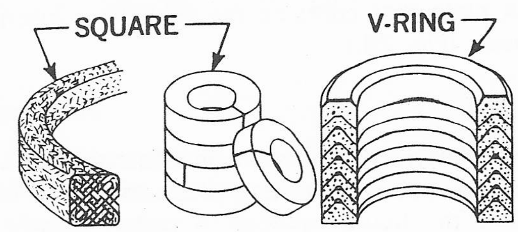


Fig. 17 PACKING

Gland

A part which retains the packing in the stuffing box. (Fig. 18.)

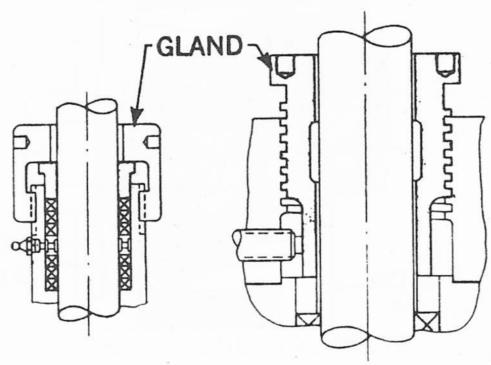


Fig. 18 GLAND

Lantern Ring (Seal Cage)

A ring located in the stuffing box to provide space for the introduction of a lubricant or a barrier liquid. (Fig. 19.)

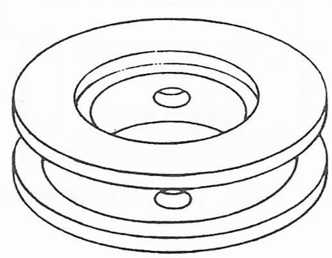


Fig. 19 LANTERN RING

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). (Figs. 20 to 23.)

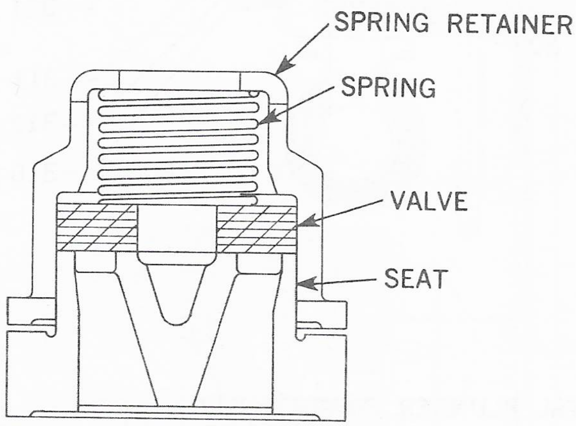


Fig. 20 PLATE VALVE ASSEMBLY

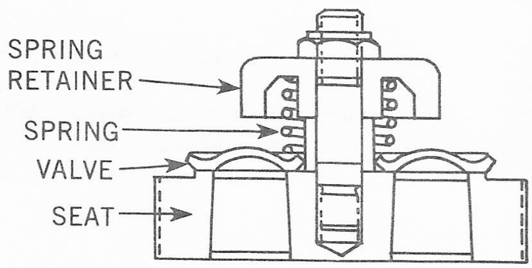


Fig. 21 DISC VALVE ASSEMBLY

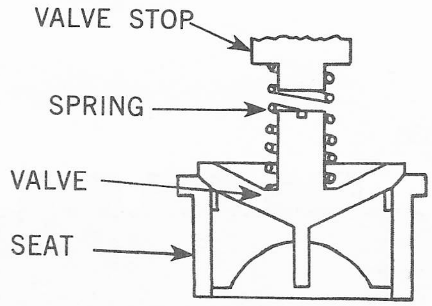


Fig. 22 WING VALVE ASSEMBLY

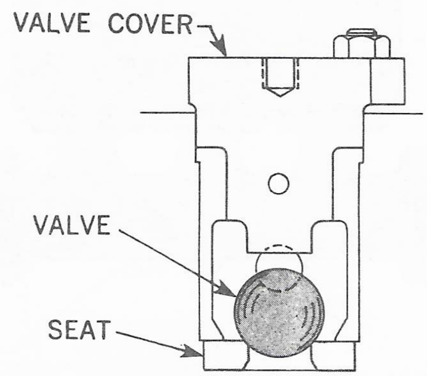


Fig. 23 BALL VALVE ASSEMBLY

Upper Crosshead

Used in vertical plunger pumps to transmit the reciprocating motion of the side rod to the plunger. (Fig. 24.)

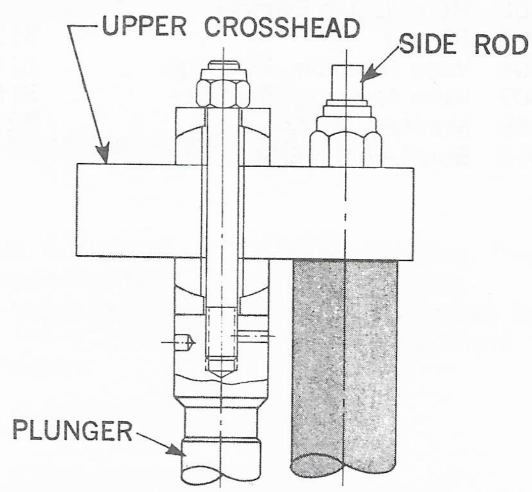


Fig. 24 UPPER CROSSHEAD

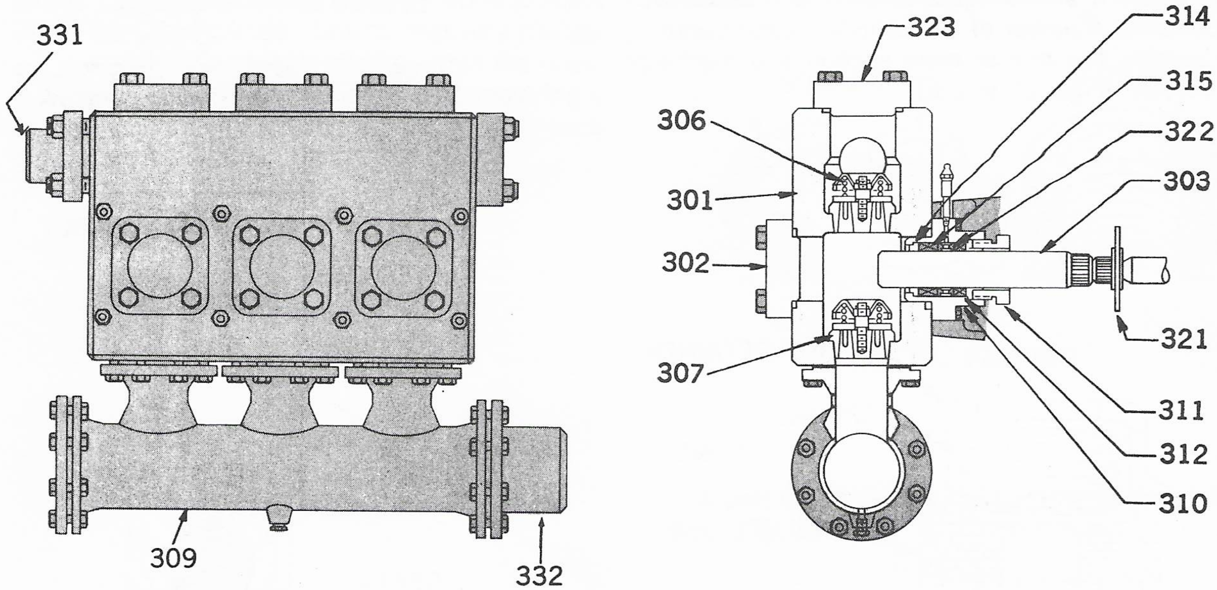


Fig. 25 LIQUID END, HORIZONTAL PLUNGER POWER PUMP

301 Cylinder, Liquid
302 Head, Liquid Cylinder
303 Plunger
306 Valve Assembly, Discharge
307 Valve Assembly, Suction
309 Manifold, Suction
310 Box, Liquid Stuffing
311 Gland, Liquid Stuffing Box
312 Ring, Follower
314 Bushing, Throat
315 Packing
321 Deflector
322 Ring, Lantern
323 Cover, Valve
331 Flange, Discharge Companion
332 Flange, Suction Companion

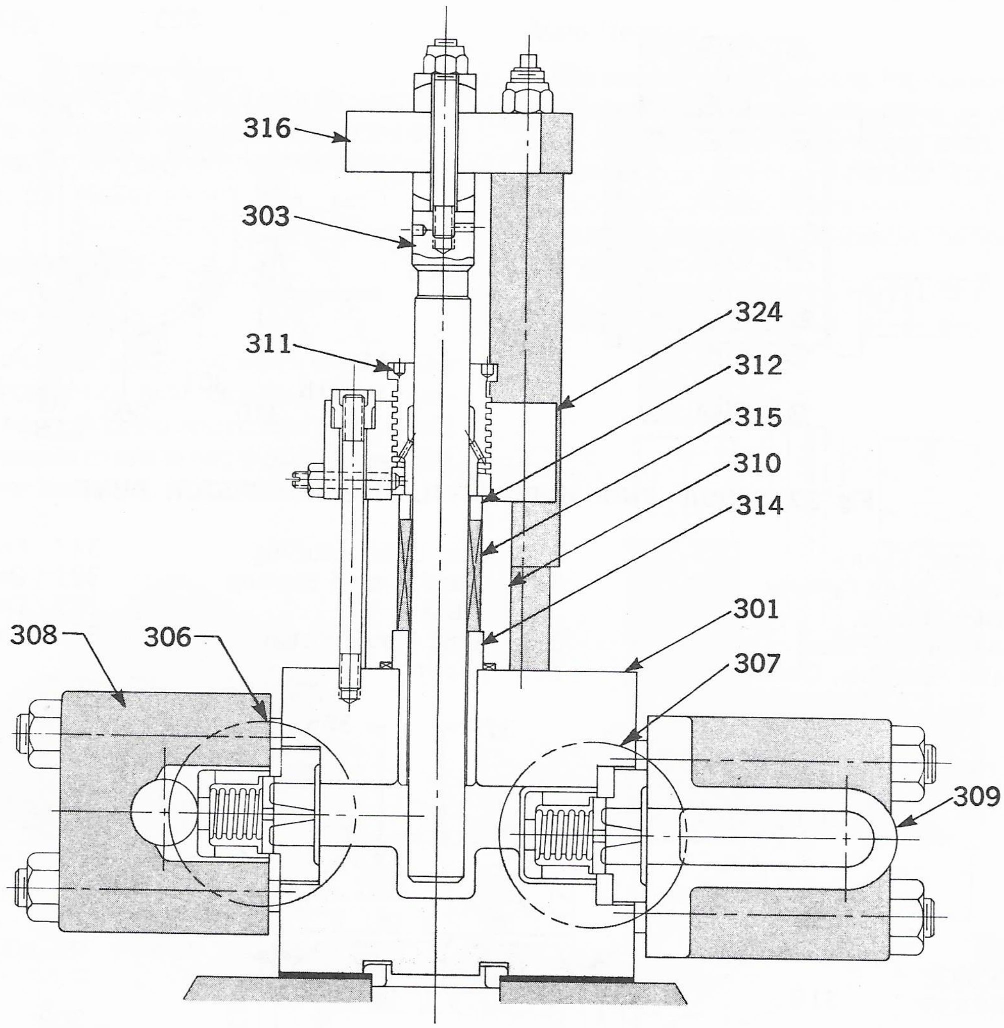


Fig. 26 LIQUID END, VERTICAL PLUNGER POWER PUMP

301 Cylinder, Liquid
303 Plunger
306 Valve Assembly, Discharge
307 Valve Assembly, Suction
308 Manifold, Discharge
309 Manifold, Suction
310 Box, Liquid Stuffing
311 Gland, Liquid Stuffing Box
312 Ring, Follower
314 Bushing, Throat
315 Packing
316 Crosshead, Upper
324 Ring, Gland

reciprocating pumps, power
nomenclature

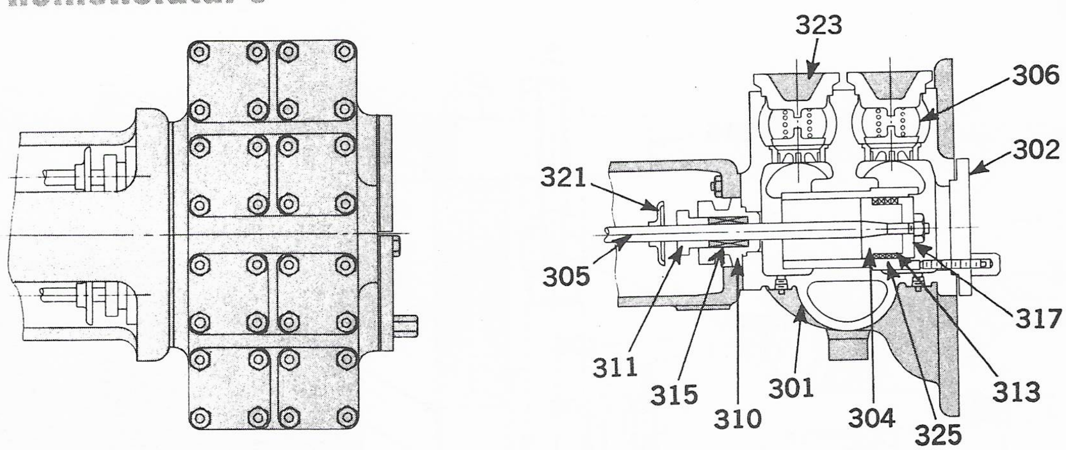


Fig. 27 LIQUID END, HORIZONTAL SIDE POT PISTON PUMP

301 Cylinder, Liquid
302 Head, Liquid Cylinder
304 Piston, Liquid
305 Rod, Liquid Piston
306 Valve Assembly, Discharge
310 Box, Liquid Stuffing
311 Gland, Liquid Stuffing Box
313 Ring, Liquid Piston
315 Packing
317 Follower, Piston
321 Deflector
323 Cover, Valve
325 Liner, Cylinder

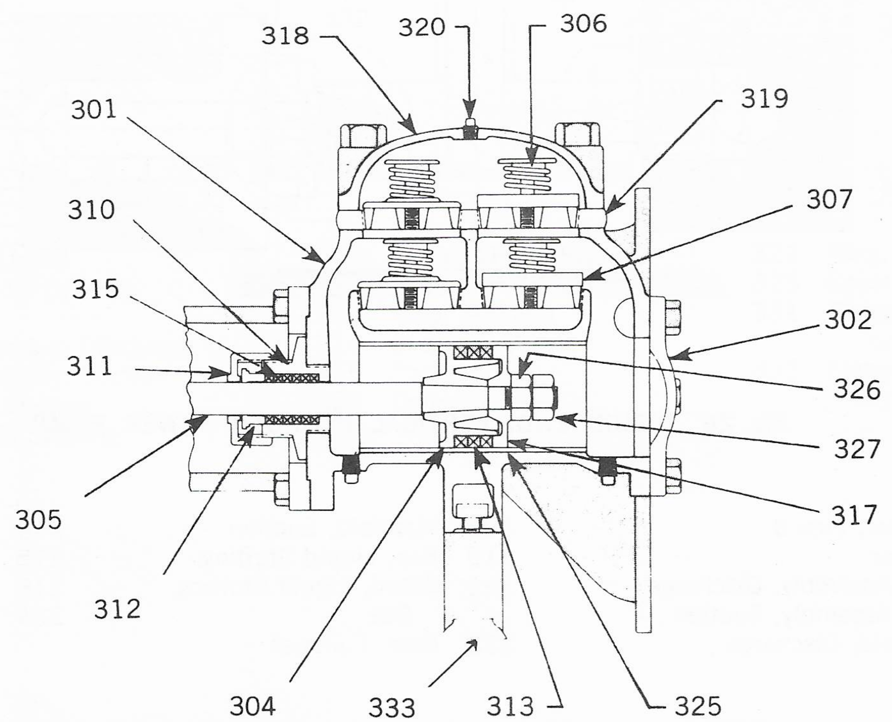


Fig. 28 LIQUID END, HORIZONTAL VALVE PLATE PISTON PUMP

301 Cylinder, Liquid
302 Head, Liquid Cylinder
304 Piston, Liquid
305 Rod, Liquid Piston
306 Valve Assembly, Discharge
307 Valve Assembly, Suction
310 Box, Liquid Stuffing
311 Gland, Liquid Stuffing Box
312 Ring, Follower
313 Ring, Liquid Piston
315 Packing
317 Follower, Piston
318 Cover, Liquid Valve Chest
319 Plate, Valve
320 Plug
325 Liner, Cylinder
326 Nut, Piston
327 Nut, Piston Jam
333 Foot, Liquid Cylinder

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. (Figs. 39, 40, 41.)

POWER END PARTS

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, one upper and one lower half. (Fig. 29.)

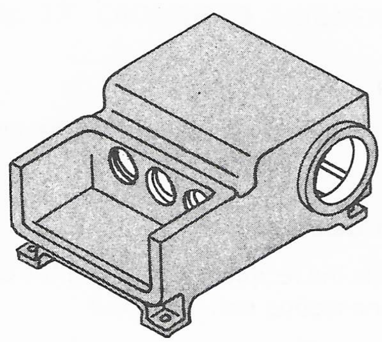


Fig. 29 POWER FRAME

Crankshaft

The stepped shaft which transmits power and motion to the connecting rods. Main bearings and connecting rods are fitted on this member. (Fig. 30.)

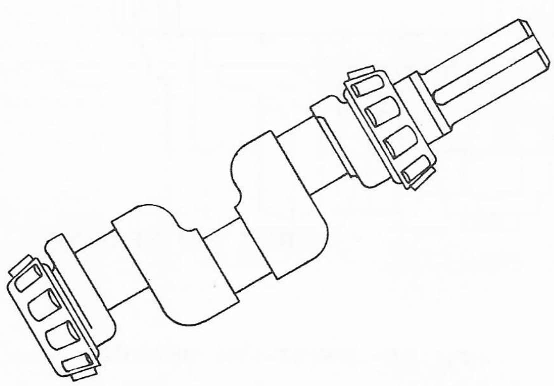


Fig. 30 CRANKSHAFT

Main Bearing

The bearing which supports the crankshaft. Main crankshaft bearings may be sleeve or antifriction type, mounted at each end of the shaft or located elsewhere to provide proper support. These bearings absorb the liquid and inertia loads which are developed by the plunger as it displaces the liquid. (Figs. 31, 32.)

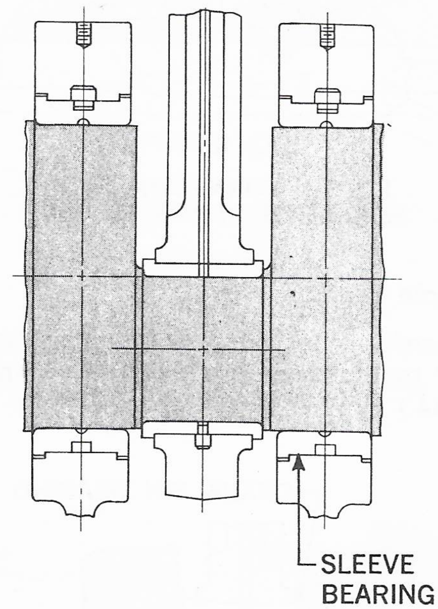


Fig. 31 SLEEVE BEARING

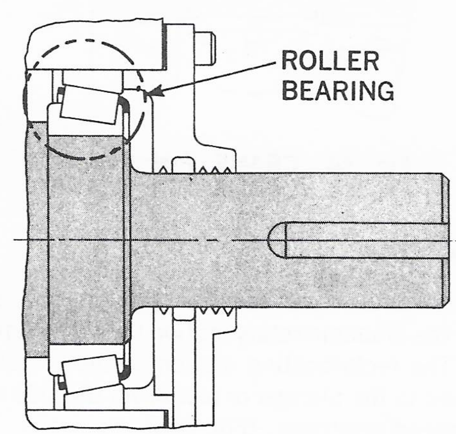


Fig. 32 ROLLER BEARING

Connecting Rod

Articulates the motion of the crankshaft to the crosshead. Power is transmitted thru compression and/or tension. (Fig. 33.)

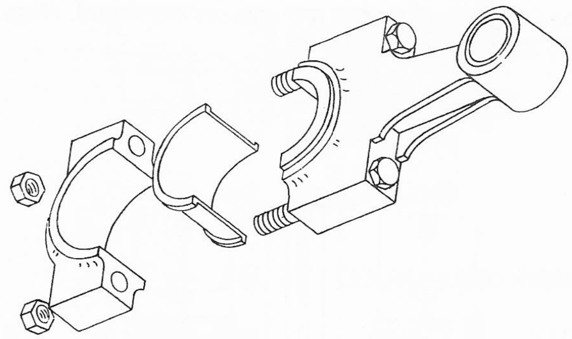


Fig. 33 CONNECTING ROD

Crankpin Bearing

Transmits the oscillating reciprocating load transmitted by the connecting rod to the crankshaft. (Fig. 34.)

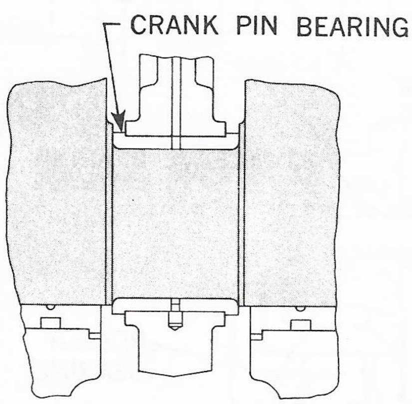


Fig. 34 CRANK PIN BEARING

Power Crosshead

Creates a linear reciprocating motion derived from the crankpin rotary motion thru the connecting rod. The reciprocating motion of the crosshead is applied to the plunger or piston via the side rods, or crosshead extension. (Fig. 35.)

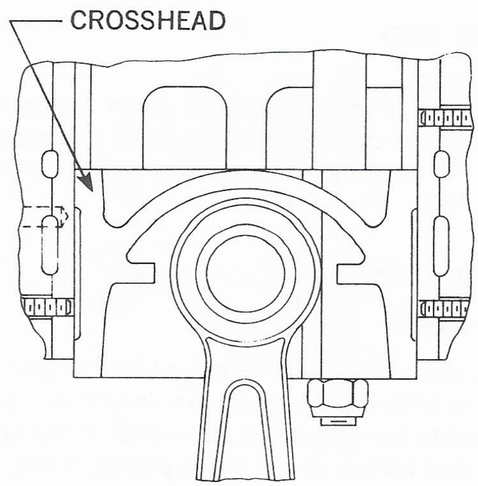


Fig. 35 POWER CROSSHEAD

Wrist Pin

Connects the connecting rod to the crosshead. (Fig. 36.)

Wrist Pin Bearing

Transmits the reciprocating load of the crosshead into the connecting rod. (Fig. 36.)

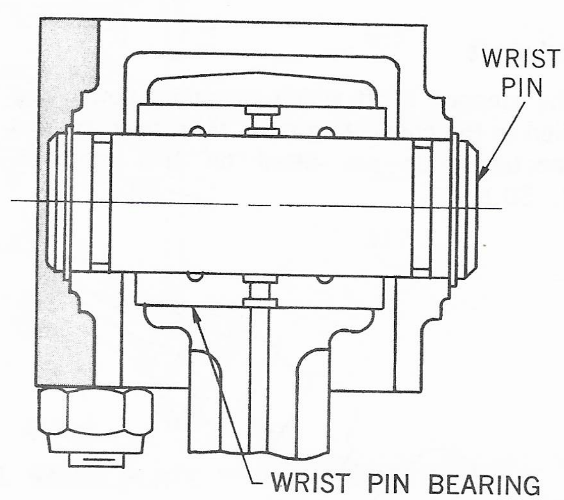


Fig. 36 WRIST PIN BEARING

Crosshead Extension (Plunger Extension)

Connects the crosshead to the plunger. (Fig. 37.)

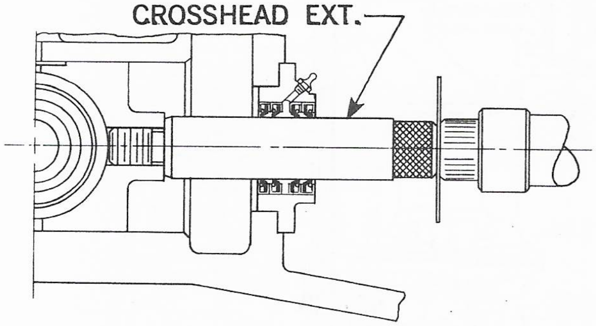


Fig. 37 CROSSHEAD EXTENSION

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. (Fig. 38.)

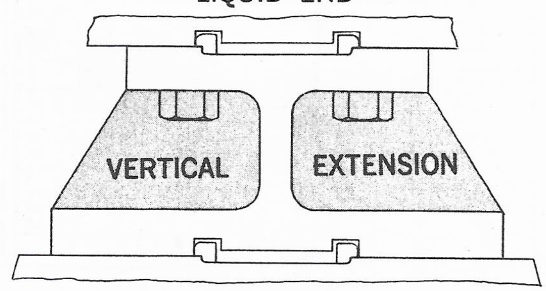


Fig. 38 FRAME EXTENSION

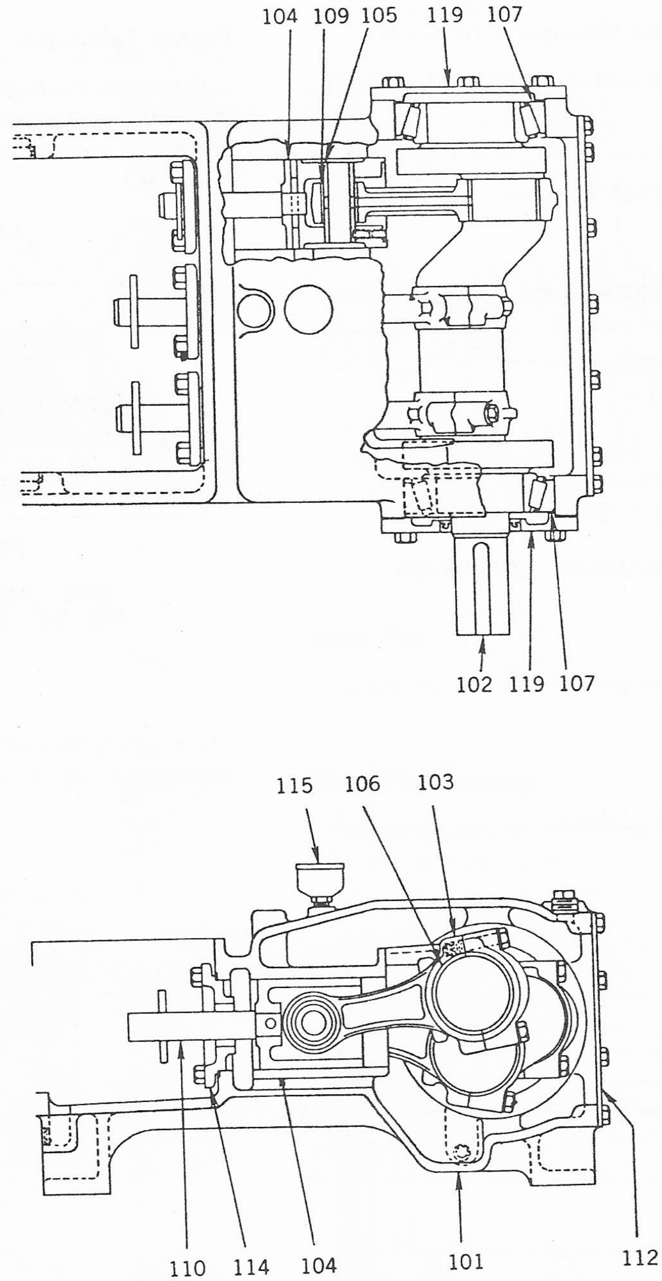


Fig. 39 POWER END, HORIZONTAL PLUNGER POWER PUMP

101 Frame, Power
102 Crankshaft
103 Rod, Connecting
104 Crosshead, Power
105 Pin, Wrist
106 Bearing, Crankpin
107 Bearing, Main Crankshaft
109 Bearing, Wrist Pin
110 Extension, Crosshead
112 Cover, Crankcase
114 Box, Wiper
115 Breather
119 Housing, Crankshaft Bearing

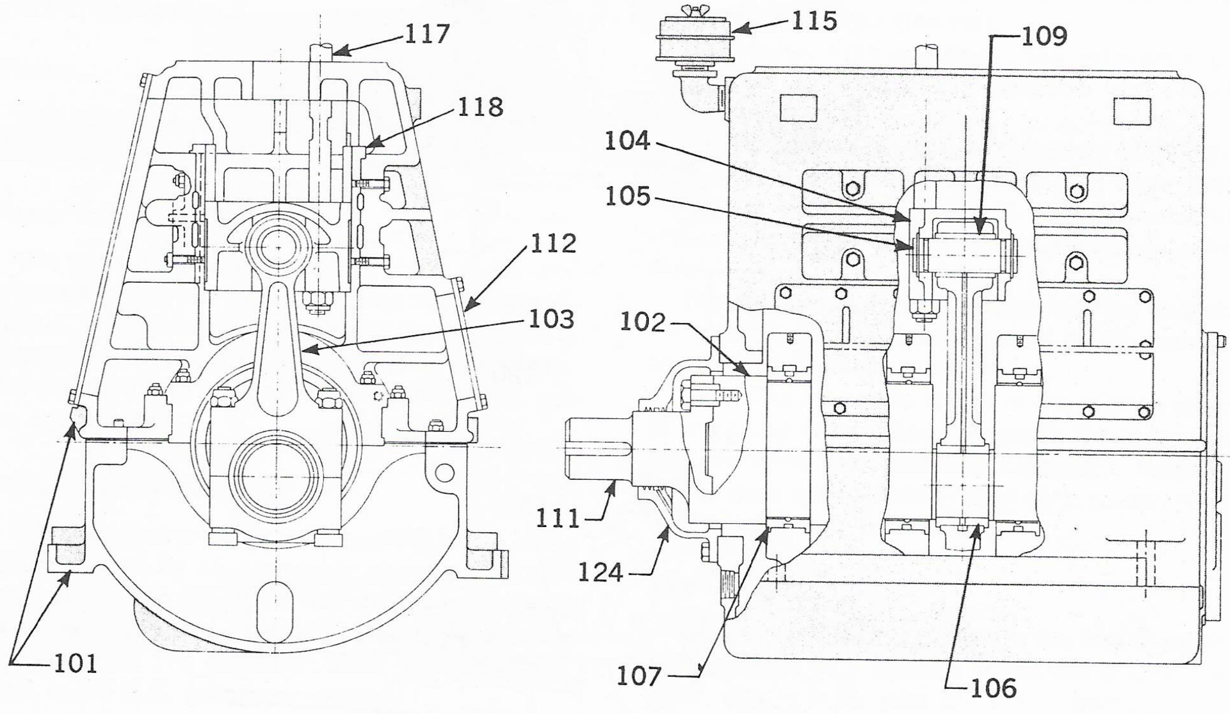
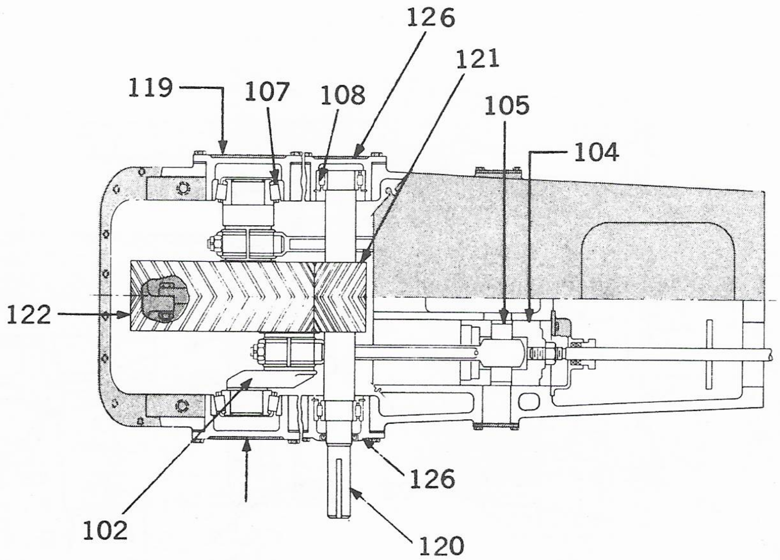


Fig. 40 POWER END, VERTICAL PLUNGER POWER PUMP

101	Frame, Power	107	Bearing, Main, Crankshaft	115	Breather
102	Crankshaft	109	Bearing, Wrist Pin	117	Rod, Side
103	Rod, Connecting	111	Extension Crankshaft	118	Way, Crosshead
104	Crosshead, Power	112	Cover, Crankcase	124	Cover, Crankshaft Extension
105	Pin, Wrist				
106	Bearing, Crankpin				



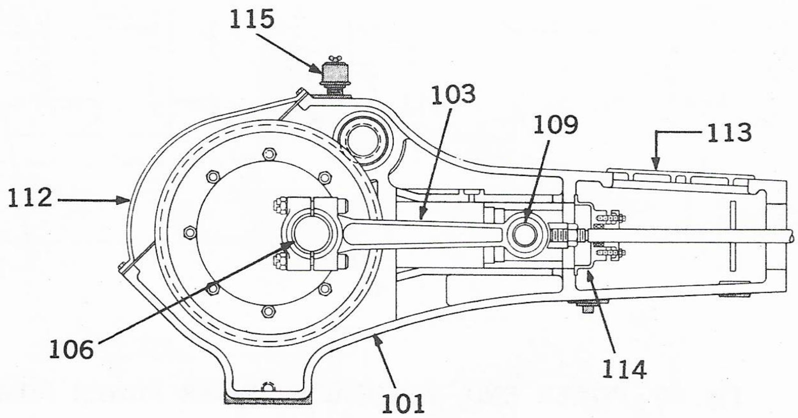


Fig. 41 POWER END, HORIZONTAL DUPLEX POWER PUMP WITH INTEGRAL GEARS

101 Frame, Power
102 Crankshaft
103 Rod, Connecting
104 Crosshead, Power
105 Pin, Wrist
106 Bearing, Crankpin
107 Bearing, Main, Crankshaft
108 Bearing, Pinion Shaft
109 Bearing, Wrist Pin
112 Cover, Crankcase
113 Cover, Cradle
114 Box, Wiper
115 Breather
119 Housing, Bearing Crankshaft
120 Pinion Shaft
121 Pinion
122 Gear
126 Housing, Bearing, Pinion Shaft

Purpose

The purpose of this section is to define terms used in pump ratings. These ratings are characteristics of pump design and not conditions of the specific application.

Stroke

One complete uni-directional motion of piston or plunger. Stroke length is expressed in inches.

Pump Capacity (Q)

The capacity of a reciprocating pump is the total volume through-put per unit of time at suction conditions. It includes both liquid and any dissolved or entrained gases at the stated operating conditions. The standard unit of pump capacity is the U.S. gallon per minute.

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. The standard unit of pump displacement is the U.S. gallon per minute.

For single-acting pumps:

$$D = \frac{Asnm}{231}$$

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

$$D = \frac{(2A - a)\ snm}{231}$$

where

- A = Plunger or piston area, square inch
- a = Piston rod cross-sectional area, square inch (double-acting pumps)
- s = Stroke length, inch
- n = RPM of crankshaft
- m = Number of pistons or plungers

Plunger or Piston Speed (v)

The plunger or piston speed is the average speed of the plunger or piston. It is expressed in feet per minute.

$$v = \frac{ns}{6}$$

Pressures

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

Discharge Pressure (p_d)—The liquid pressure at the centerline of the pump discharge port.

Suction Pressure (p_s)—The liquid pressure at the centerline of the suction port.

Differential Pressure (p_{td})—The difference between the liquid discharge pressure and suction pressure.

Net Positive Suction Head Required (NPSHR)—The amount of suction pressure, over vapor pressure, required by the pump to obtain satisfactory volumetric efficiency and prevent excessive cavitation.

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. Required NPSH does not include system acceleration head, which is a system-related factor.

Slip (S)

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

Power (P)

Pump Power Input (P_i)—The mechanical power delivered to a pump input shaft, at the specified operating conditions. Input horsepower may be calculated as follows:

$$P_i = \frac{Q \times p_{td}}{1714 \times \eta_p}$$

Pump Power Output (P_o)—The hydraulic power imparted to the liquid by the pump, at the specified operating conditions. Output horsepower may be calculated as follows:

$$P_o = \frac{Q \times p_{td}}{1714}$$

The standard unit for power is the horsepower.

Efficiencies (η)

Pump Efficiency (η_p) (also called pump mechanical efficiency)—The ratio of the pump power output to the pump power input.

$$\eta_p = \frac{P_o}{P_i}$$

Volumetric Efficiency (η_v)—The ratio of the pump capacity to displacement.

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

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 pounds force.

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 pounds force.

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 inches.

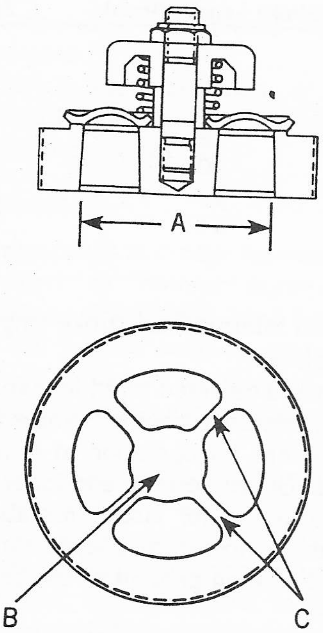


Fig. 42 DISC VALVE

In Figs. 42 and 43 the seat area = A minus B minus C; in Fig. 44 the seat area = A minus D; in Fig. 45 the seat area = A.

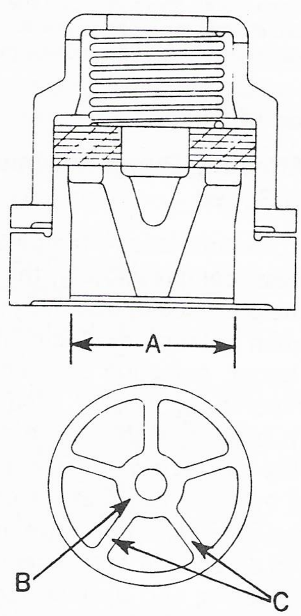


Fig. 43 PLATE VALVE

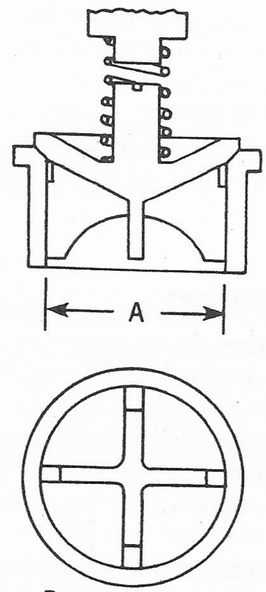


Fig. 44 WING VALVE

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.

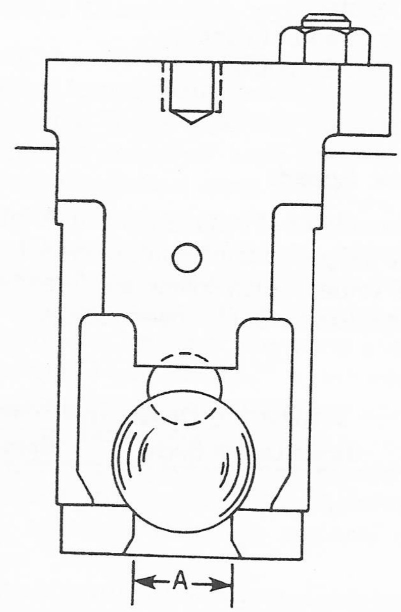


Fig. 45 BALL VALVE

reciprocating pumps, power
applications

Purpose

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

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 on page 255.
Spray Drying
Standby Liquid Control (nuclear power plant)
Transfer
Wash-water Injection
Waste Disposal

Basic Speeds

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

Single-Acting Plunger-Type Power Pumps

Stroke Length (Inch)	Speed (RPM)
2	750
3	530
4	420
5	360
6	315
7	290
8	262

Double-Acting Piston-Type Power Pumps

Stroke Length (Inch)	Speed (RPM)
2	140
4	116
6	100
8	90
10	83
12	78
14	74
16	70

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.

As a guide, for viscosities above 300 SSU at pumping temperature, speeds are normally reduced to the following percent of the basic speeds:

Viscosity (SSU)	1000	2000	4000	6000	8000	10,000
% of Basic Speed	90	80	70	62	55	50

Note: The 12th Edition of the Standards contained "basic speeds" for power pumps, and these numbers became a "yard stick" or "bench mark" for various user specifications. The 13th Edition did not contain the "basic speeds." So that these numbers may be retained as reference points, they are stated above.

Discussion of Speeds

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—available NPSH, piping design and layout, pulsation dampeners (if any), 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.

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.

Medium Speeds

Power pump speeds at or near the manufacturer's published "rated" or "normal" curve will include those applications when clean, cold liquids are involved, and will provide 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, in some cases, automatic safety shut-down devices, suction and discharge dampeners, and may also require plunger or piston rod packing lubrication.

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

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 down-time 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.

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 only optimum application factors are present, and that 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.

STARTING POWER PUMPS

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 on 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 will 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 smoother is the torque. 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 which reduces line voltage disturbances which Design "B" motors would cause.

Large double-acting duplex power pumps particularly 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, or other preventive steps are taken.

Pump Torque Criteria

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, i.e. the twisting effort (moment) applied to pump shaft or reducer shaft, falls usually into one or two general applications, as follows.

Starting With Liquid By-Pass

Operating personnel manually opens by-pass valve. Or, a power-actuated dump valve (programmed to open automatically) by-passes 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 by-pass (dump) valve remains fully or partly open. (See Fig. 50.)

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

With liquid by-pass, the total starting torque requirement is mainly related to the mechanical inertia of the pump, couplings, gears and motor rotor. These items are heavy, and substantial starting torque may result. All the liquid in the pump suction line and in the by-pass line must be accelerated from stand-still 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 liquid) and the rate of acceleration, plus total system friction.

The rate of acceleration is very important. The starting torque is directly proportional to it. (Stated another way: Accelerating 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 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.

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 3) 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.

Starting Without Liquid By-Pass

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 capacity). This practice should therefore be avoided except in emergency situations.

Starting without liquid by-pass 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 by-passing, or suction valve unloading is provided, and it may be termed "full-load" starting.

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 over-heating (or damage) may result unless the motor and its starting equipment have been generously selected and sized to meet this. 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 only to the applied voltage, and to the size and design of the motor, an across-the-line starter will provide the maximum starting torque. And, a given induction motor will develop 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) will simply require longer accelerating time than will 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 by-passing provided. Then, pump discharge pressure will be related very largely to pump speed (discharge rate) and acceleration rate.

Consider a single pump which forces fluid through a long piping system which is nearly level. 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.

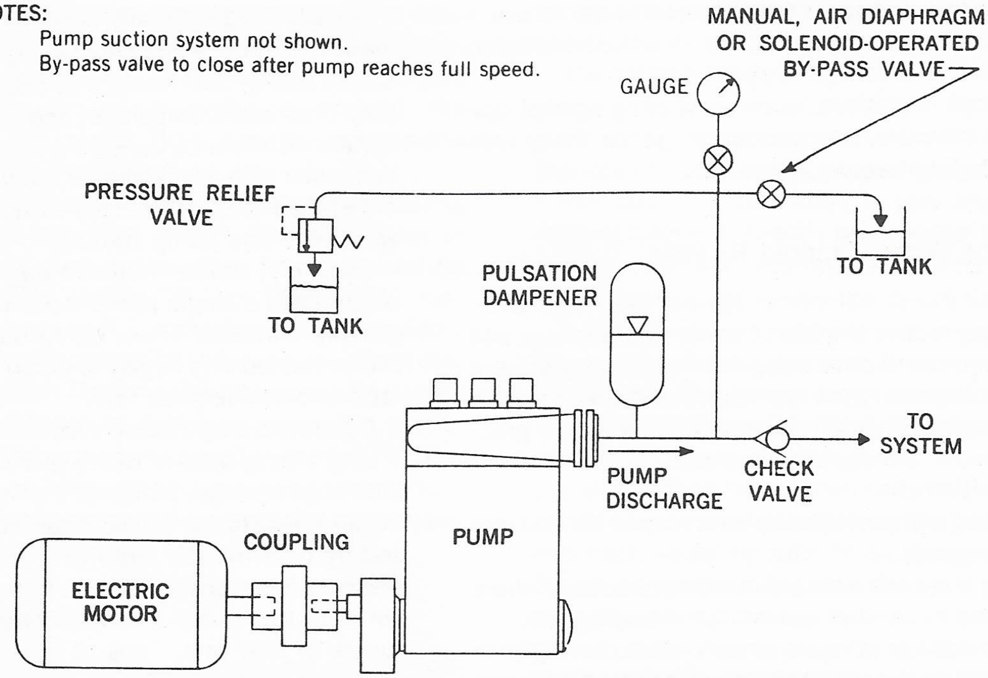
Analysis of the starting torque requirement is complex, and depends on the inertia of the accelerating liquid, the size and length of piping, liquid viscosity and density, and on the elasticity of the piping. If the mass of liquid is very large, a discharge dampener and a check valve may be advantageous, 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, since 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 will usually be adequate. An alternate is the automatic by-pass valve and check valve arrangement, with dampener as shown in Fig. 50.

The arrangement shown in Fig. 50 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.

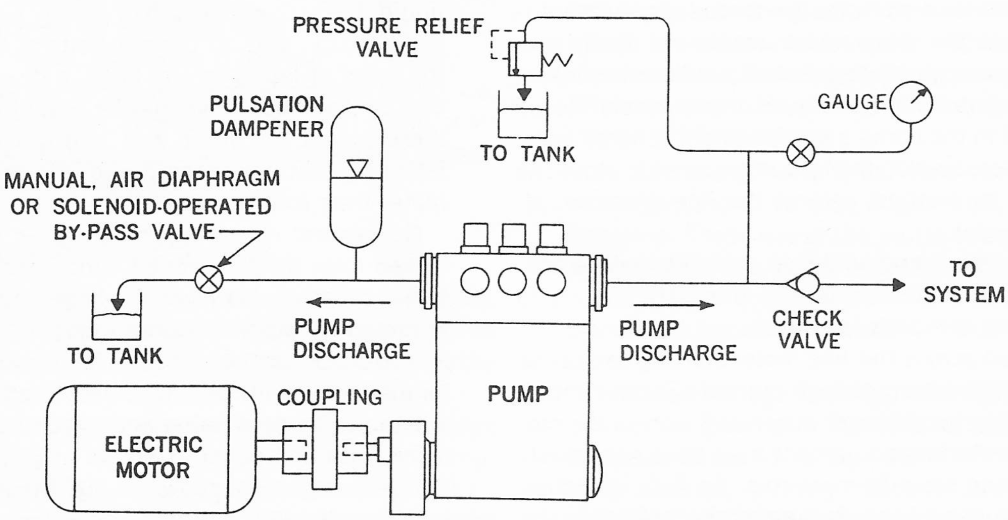
The arrangement affords a convenient means of expelling any air trapped in the pump cylinders before placing pressure load on the pump. This is desirable, especially for multicylinder pumps which sometimes become rough and "air bound" after

NOTES:
Pump suction system not shown.
By-pass valve to close after pump reaches full speed.



SCHEMATIC: FOR PUMPS WITH SINGLE DISCHARGE CONNECTION

Pump suction system not shown.
By-pass valve to close after pump reaches full speed.



SCHEMATIC: FOR PUMPS WITH DUAL DISCHARGE CONNECTIONS

Fig. 50 SCHEMATICS OF LIQUID BY-PASS SYSTEMS

servicing or prolonged idleness. Simply open the liquid by-pass valve to allow the liquid to discharge back to the tank thereby expelling the air. When running smoothly, close by-pass valve and thus load the pump.

Electric Motor Locked-Rotor Torques

The following table summarizes minimum locked rotor torque ratings for standard NEMA Design "B" 60 Hertz squirrel-cage induction motors expressed as percent of full-load torque:

TABLE 3
Minimum Locked-Rotor Torque Ratings

Rating (HP)	1800 RPM Motors	1200 RPM Motors
1	275%	170%
1½	250%	165%
2	235%	160%
3	215%	155%
5	185%	150%
7½	175%	150%
10	165%	150%
15	160%	140%
20	150%	135%
25	150%	135%
30	150%	135%
40	140%	135%
50	140%	135%
60	140%	135%
75	140%	135%
100	125%	125%
125	110%	125%
150	110%	120%
200	100%	120%
250	80%	100%
300	80%	100%
350	80%	100%
400	80%	
450	80%	
500	80%	

Note: In the range from 1 through 75 horsepower, the 1800 RPM motors show higher locked-rotor torque ratings than do the 1200 RPM motors. However, from 125 through 350 horsepower, the 1200 RPM motors have larger NEMA ratings.
Locked rotor torques of large motors must be carefully evaluated before final selection.

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 sufficiently above vapor pressure to prevent flashing as the liquid enters the pump chambers. If gas bubbles are entrained in the liquid, or if flashing occurs in the pump, damaging vibrations may occur in both inlet and outlet lines, volumetric efficiency will drop, and various pump and system components may fail. Small amounts of gas or cavitation can reduce life of packing, valve springs, valves, seats and gaskets. Larger quantities of gas, or more severe cavitation, can cause pitting of liquid end components and catastrophic failure of the liquid cylinder, crankshaft, bearings, and drive train components.

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 Fig. 51, should be designed with the following features:
   a. Sufficient size to allow entrained gas bubbles to rise to the surface.
   b. Lines which feed liquid into tank below minimum liquid level.
   c. Completely submerged baffle plate separating incoming from outgoing liquid.
   d. Vortex breaker at outlet connection (to pump).
2. Each pump should be provided with a separate inlet line from liquid source to pump, rather than connecting two or more pumps to a common manifold. Mutually reinforcing pulsations are thus avoided.
3. Inlet pipe diameter should be at least equal to, and preferably larger than, pump inlet connection.
4. Inlet pipe should be as short and direct as possible with a minimum of turns, bends, and restrictions. All turns should be made with long-radius elbows or laterals. Pulsations resulting from a long inlet line can sometimes be partially reduced by a pulsation dampener and sometimes by raising the liquid level at the source, but these changes seldom provide results as satisfactory as a short, direct, large-diameter line.
5. The inlet system must provide NPSH that exceeds the sum of the NPSHR of the pump, all friction losses, *and acceleration head*. (See separate discussion on page 252.) Additional head must be provided if the liquid contains dissolved gases. It is recommended that a margin of at least 7 feet be provided.
6. The inlet system should contain no high points that would collect gas. All "horizontal" runs should slope up toward the pump. The pipe reducer at the pump inlet should be of the eccentric type, installed with the *flat side up*.
7. A strainer, if used, should have a free flow area at least three times the flow area of the inlet line. If there is doubt about its regular mainte-

nance, a strainer should not be used. (A plugged 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 a system will not provide sufficient NPSH, and cannot be redesigned, it shall be necessary to do one or more of the following:

1. Install pulsation dampener in inlet line adjacent to power pump liquid cylinder. A dampener, properly installed and charged, may significantly reduce the length of pipe used in the acceleration head equation (see Pulsation Dampener, following).
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.

A booster pump for a power pump is normally a centrifugal pump, but may be a positive displacement pump under special conditions (see paragraph 4). 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 Fig. 52).
2. The booster pump must add enough pressure to the system to provide sufficient NPSH to the power pump allowing for the acceleration head and friction losses.
3. Install pulsation dampener in inlet line adjacent to power pump liquid cylinder (or if of proper construction, on the opposite side of cylinder, see Fig. 52). The dampener is often omitted, though, between a centrifugal booster pump and a low-speed power pump under the following conditions:

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

   b. Diameters of all piping between liquid source and power pump are equal to, or larger than, inlet connection of power pump.

   c. The booster pump is sized for maximum instantaneous capacity of the power pump. The following tabulation gives the percentage that the maximum instantaneous capacity exceeds the mean capacity for each type of power pump.

Type of Power Pump		% Over Mean Capacity
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%

   d. 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 by-pass valve is required between pumps. The booster pump must be sufficiently over-sized to provide the minimum flow required through the by-pass valve.

Suction Tank

It is recommended that a suction tank, if used, be arranged as shown in Fig. 51 for return and make-up lines.

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	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.

Where the suction or discharge line, or both, are of considerable length, or if the suction is under a static head of poor design, or where the liquid handled is hot, a desurging device of suitable size for the suction or discharge lines, or both, may sometimes be necessary to insure smooth, quiet operation

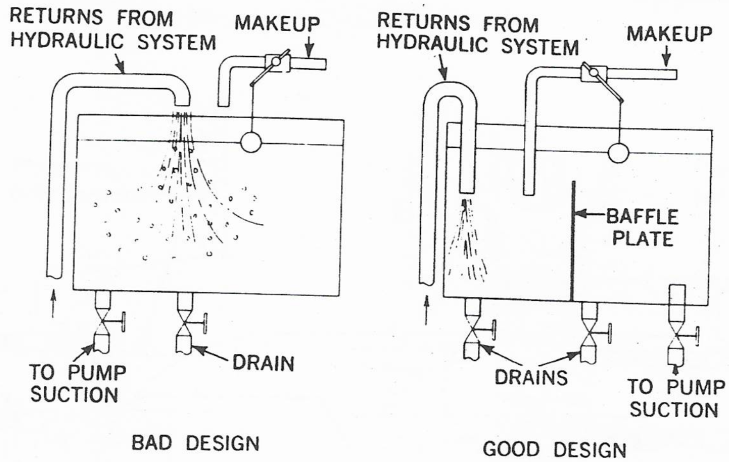


Fig. 51 SUCTION TANKS

of the system. The size of the pulsation dampener will depend upon the type, size, and speed of pump, the liquid, and the layout of the piping systems. Recommendations as to size and type of pulsation dampener should be obtained from the pulsation dampener manufacturer.

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 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 Steam Pump	— Double Acting
Duplex Power Pump	— Single and Double Acting
Simplex Power Pump	— Double Acting
Simplex Steam 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.

Pulsation dampeners, particularly on the suction, should be located as close as possible to the pump and in such 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 can be at least 1 to 1½ times the pump displacement per revolution.

Discharge Piping

To facilitate starting and eliminate air, a by-pass valve should be installed close to the pump. Also, to protect the pump, a stop valve and a check valve should be employed. See Fig. 52. If an increaser is used to increase the size of the piping, it should be placed between check valve and pump.

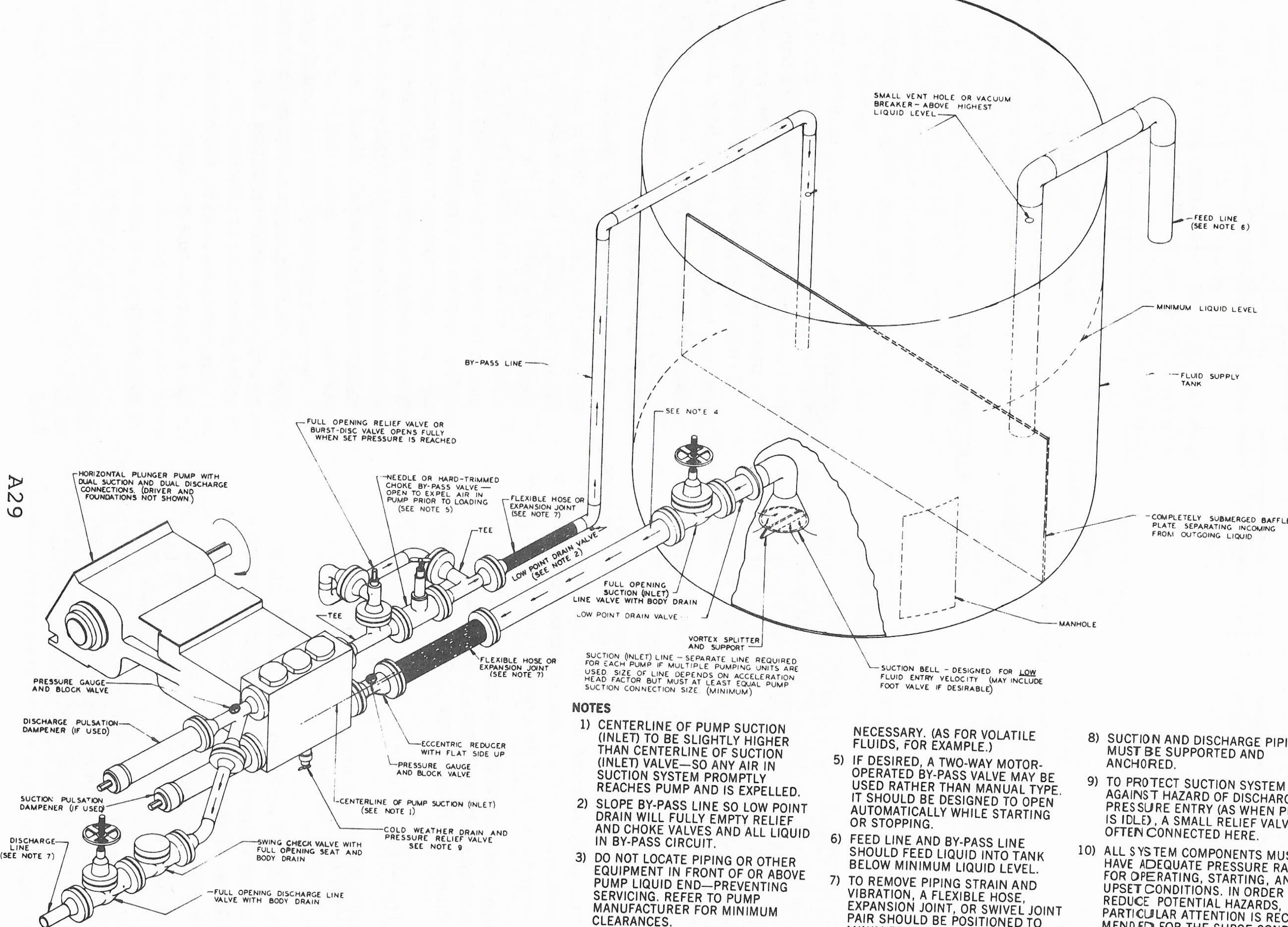
The vibration in pipe lines must be minimized, with lines as short and as direct as practicable. When a change in direction is required, long radius elbows and tees or a 45 degree bend in the pipe itself should be utilized. "Dead ends" must be avoided.

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.

Relief Valve

The insertion of a discharge relief valve of suitable size for the capacity of the pump, set to open at a



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.

Fig. 52 SUGGESTED PIPING SYSTEM FOR POWER PUMPS

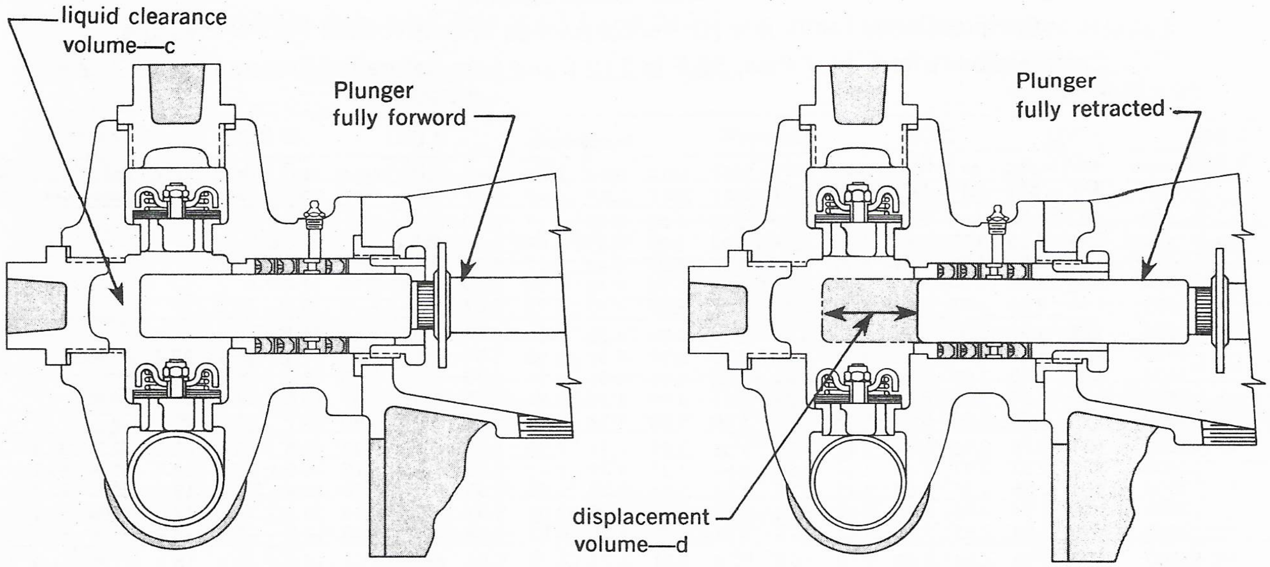


Fig. 53 PLUNGER MOVEMENT WHEN CALCULATING VOLUMETRIC EFFICIENCY

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 capacity so it will relieve the full capacity 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.

Calculating Volumetric Efficiency For Water

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

$$\text{Vol. Eff.} = \frac{1 - \left[P_{td}\beta_t \left(1 + \frac{c}{d} \right) \right]}{1 - P_{td}\beta_t} - S$$

where

β_t = Compressibility factor at temperature t (degrees Fahrenheit or centigrade). (See Tables 4 and 5).

c = Liquid chamber volume in the passages of chamber between valves when plunger is at the end of discharge stroke in cu in

d = Volume displacement per plunger in cu in

P_{td} = Discharge pressure minus suction pressure in psi

S = Slip, expressed in decimal value

reciprocating pumps, power
applications

TABLE 4
Water Compressibility

Compressibility Factor $\beta_t \times 10^{-6}$ = Contraction in Unit Volume Per Psi Pressure
Compressibility from 14.7 Psia, 32 F to 212 F and from Saturation Pressure Above 212 F

Pressure Psia	Temperature 0 C 32 F	20 C 63 F	40 C 104 F	60 C 140 F	80 C 176 F	100 C 212 F	120 C 248 F	140 C 284 F	160 C 320 F	180 C 356 F	200 C 392 F	220 C 428 F	240 C 464 F	260 C 500 F	280 C 536 F	300 C 572 F	320 C 608 F	340 C 644 F	360 C 680 F
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	2.83	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

EXAMPLE: Find the volumetric efficiency of a reciprocating pump with the following conditions:

Type of pump	3 in diam plunger × 5 in stroke triplex
Liquid pumped	Water
Suction pressure	Zero psig
Discharge pressure	1785 psig
Pumping temperature	140 F
c	127.42 cu in
d	35.343 cu in
S	.02

Find β_t from Table of Water Compressibility (Table 4).

$\beta_t = .00000305$ at 140 F and 1800 psia

Calculate volumetric efficiency:

$$\text{Vol. Eff.} = \frac{1 - \left[P_{td}\beta_t \left(1 + \frac{c}{d} \right) \right]}{1 - P_{td}\beta_t} - S =$$

$$\frac{1 - \left[(1785 - 0)(.00000305) \left[1 + \frac{127.42}{35.343} \right] \right]}{1 - (1785 - 0)(.00000305)} - .02$$

$= .96026$

$= 96$ per cent

Calculating Volumetric Efficiency For Hydrocarbons

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

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

where

c	= Fluid chamber volume in the passages of chamber between valves, when plunger is at the end of discharge stroke, in cubic inches
d	= Volume displacement per plunger, in cubic inches
P	= pressure in psia (P_s = suction pressure in psia; P_d = discharge pressure in psia)

TABLE 5

Water Compressibility

Compressibility Factor $\beta_t \times 10^{-6}$ = Contraction in Unit Volume Per Psi Pressure
Compressibility from 14.7 Psia at 68 F and 212 F and from Saturation Pressure at 392 F

Pressure Psia	Temperature 20 C 68 F	100 C 212 F	200 C 392 F	Pressure Psia	Temperature 20 C 68 F	100 C 212 F	200 C 392 F
6000	2.84	3.14	5.20	22000	2.61	2.42	3.75
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				

P_c = Critical pressure of liquid in psia (See Table 6).

P_r = Reduced pressure $= \dfrac{\text{Actual pressure in psia}}{\text{Critical pressure in psia}} = \dfrac{P}{P_c}$

P_{rs} = Reduced suction pressure $= \dfrac{P_s}{P_c}$

P_{rd} = Reduced discharge pressure $= \dfrac{P_d}{P_c}$

S = Slip expressed in decimal value

t = Temperature, in degrees Rankine = Degrees F + 460 (t_s = suction temperature in degrees Rankine; t_d = discharge temperature in degrees Rankine)

T_c = Critical temperature of liquid, in degrees Rankine (See Table 6)

T_r = Reduced temperature $= \dfrac{\text{actual temp. in degrees Rankine}}{\text{critical temp. in degrees Rankine}} = \dfrac{t}{T_c}$ (See Fig. 54)

T_{rs} = Reduced suction temperature $= \dfrac{t_s}{T_c}$

T_{rd} = Reduced discharge temperature $= \dfrac{t_d}{T_c}$

Vol. Eff. = Volumetric efficiency expressed in decimal value.

ρ $= \dfrac{\rho_1}{\omega_1} \times \omega \times 62.4$ = density of liquid in lb per cu ft

ρ_s = Density in lb per cu ft at suction pressure

ρ_d = Density in lb per cu ft at discharge pressure

ω = Expansion factor of liquid (See Fig. 54)

$\dfrac{\rho_1}{\omega_1}$ = Characteristic constant in grams per cubic centimeter for any one liquid which is established by density measurements and the corresponding values of ω (see Table 6)

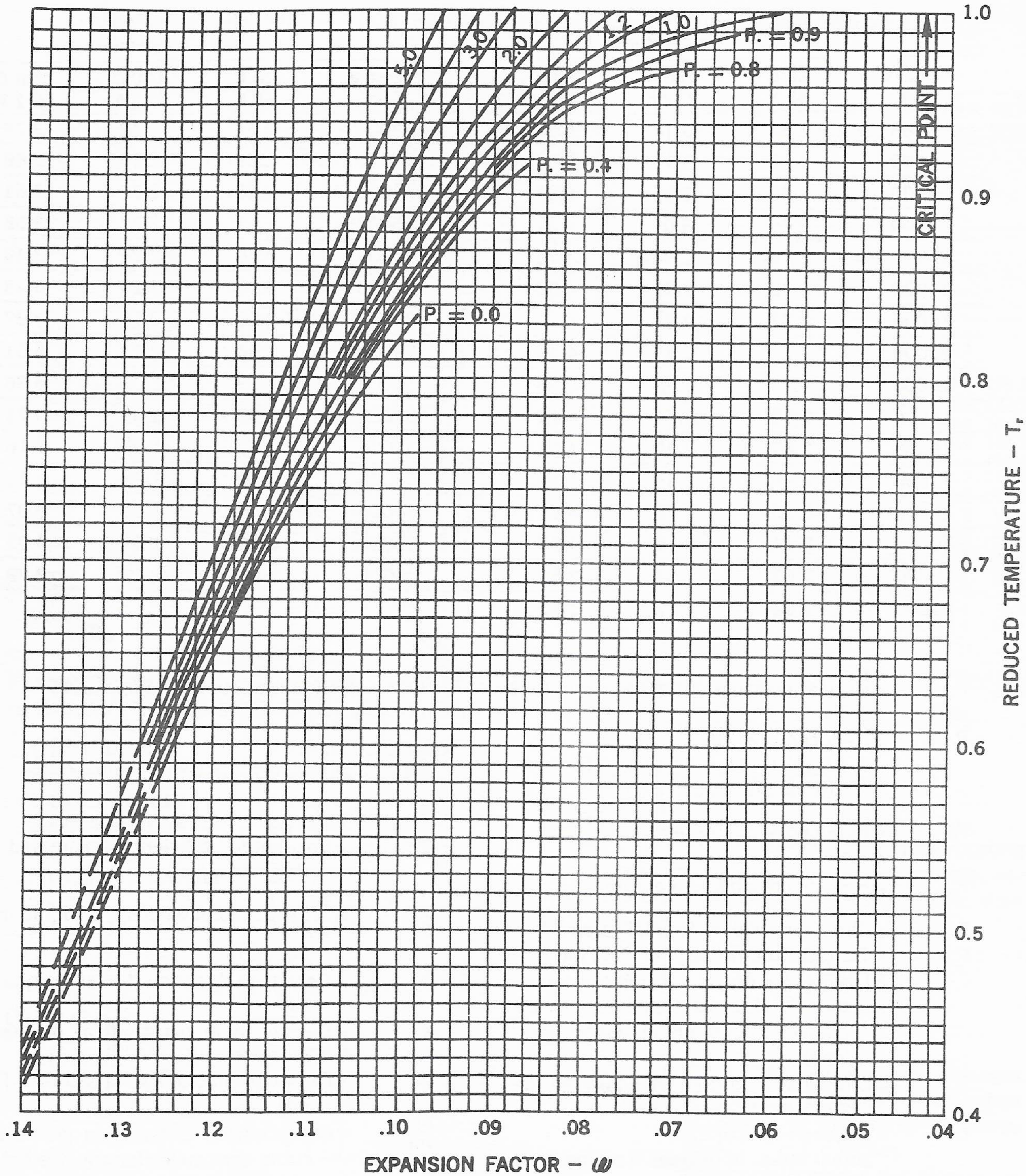


Fig. 54 THERMAL EXPANSION AND COMPRESSIBILITY OF LIQUIDS

TABLE 6
Physical Properties of Hydrocarbons

Carbon Atoms	Name	T_c degrees Rankine	P_c lb per sq in.	ρ_1/ω_1 grams per cc
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
18	Octadecane	1345	202	*5.49
20	Eicosane	1380	187	*5.5
25	Pentacosane	1465	156	*5.5
30	Triacontane	1545	133	*5.5
35	Pentatriacontane	1610	120	*5.5
40	Tetracontane	1675	108	*5.5
45	Pentatetracontane	1740	100	*5.4

*Based on experimental density, questionable because near melting point.

EXAMPLE: Find volumetric efficiency of the previous reciprocating pump example with the following new conditions:

Type of Pump	3 inch dia. plunger × 5 inch stroke triplex
Liquid pumped	Propane
Suction temperature	70 F
Discharge temperature	80 F
Suction pressure	242 psig
Discharge pressure	1911 psig

Find density at suction pressure:

$$T_{rs} = \frac{t_s}{T_c} = \frac{460+70}{666} = .795$$

$$P_{rs} = \frac{P_s}{P_c} = \frac{257}{642} = .4$$

$$\frac{\rho_1}{\omega_1} = 4.803 \text{ (From Table 6, propane)}$$

$$\omega = .1048 \text{ (From Fig. 54)}$$

$$\rho_s = \frac{\rho_1}{\omega_1} \times \omega \times 62.4$$

$$= 4.803 \times .1048 \times 62.4$$

$$= 31.4 \text{ lb per cu ft}$$

Find density at discharge pressure:

$$T_{rd} = \frac{t_d}{T_c} = \frac{460+80}{666} = .81$$

$$P_{rd} = \frac{P_d}{P_c} = \frac{1926}{642} = 3.0$$

$$\omega = .1089 \text{ (From Fig. 54)}$$

$$\rho_d = \frac{\rho_1}{\omega_1} \times \omega \times 62.4$$

$$= 4.803 \times .1089 \times 62.4$$

$$= 32.64 \text{ lb per cu ft}$$

Therefore

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

$$= 1 - \left[.02 - \frac{127.42}{35.343}\left(1 - \frac{32.64}{31.4}\right) \right]$$

$$= .8376$$

$$= 83.76 \text{ per cent}$$

Friction Head (h_f)

Friction head is the hydraulic pressure required to overcome frictional resistance of a piping system to liquid flow.

Static Head (h_s)

Static suction head is the vertical distance from the centerline of the pump inlet port to the liquid level at the source of supply.

Velocity Head (h_v)

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

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 will occur whenever friction head exceeds the static suction head (submergence) of the pump.

Flooded Suction

"Flooded" suction implies that the liquid will flow 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 capacity.

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

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 in inches of mercury vacuum (in Hg vac). To convert, use the formula, psi = 0.49 × inches Hg. Suction lift may be thought of as "negative" static suction head.

Net Positive Suction Head Available (NPSHA)

Net positive suction head available is the total suction pressure, including allowance for acceleration head available from the system at the pump suction connection, minus the vapor pressure of the liquid at the pumping temperature. NPSHA for a reciprocating pump is normally expressed in pounds per square inch (psi) or feet.

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.

Acceleration Head (h_a)

Total suction lift, as defined in the preceding paragraphs, represents the average without reference to the fluctuation above and below this average due to the inertia effect of the fluid mass in the suction line. With the higher rotative speed of present-day pumps 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 separation and pounding or vibration of the suction line.

With the slider-crank drive 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 head required to accelerate the fluid 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 fluid and the pipe and may be calculated as follows:

$$h_a = \frac{LVnC}{Kg}$$

where

h_a = Acceleration head in feet
L = Length of suction line in feet
V = Velocity in suction line in fps
n = Pump speed in rpm
C = 0.200 for simplex double-acting
= 0.200 for duplex single-acting
= 0.115 for duplex double-acting
= 0.066 for triplex single or double-acting
= 0.040 for quintuplex single or double-acting
= 0.028 for septuplex, single or double-acting
= 0.022 for nonuplex, single or double-acting
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 (32.2 ft/sec^2)

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

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 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 great influence of liquid and piping elasticity and 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 pressure-fluctuation or piping vibration is a problem.

EXAMPLE: Given a 2″ × 5″ triplex pump running at 360 rpm and displacing 73 gpm of water with a suction pipe made up of 4 feet of 4-inch pipe and 20 feet of 6-inch pipe:

Average velocity in 4-inch pipe

$$V_4 = \frac{.321 \times 73}{12.73} = 1.84 \text{ fps}$$

Average velocity in 6-inch pipe

$$V_6 = \frac{.321 \times 73}{28.89} = 0.811 \text{ fps}$$

Acceleration head in 4-inch pipe

$$h_{a_4} = \frac{4 \times 1.84 \times 360 \times .066}{1.4 \times 32.2} = 3.88 \text{ ft}$$

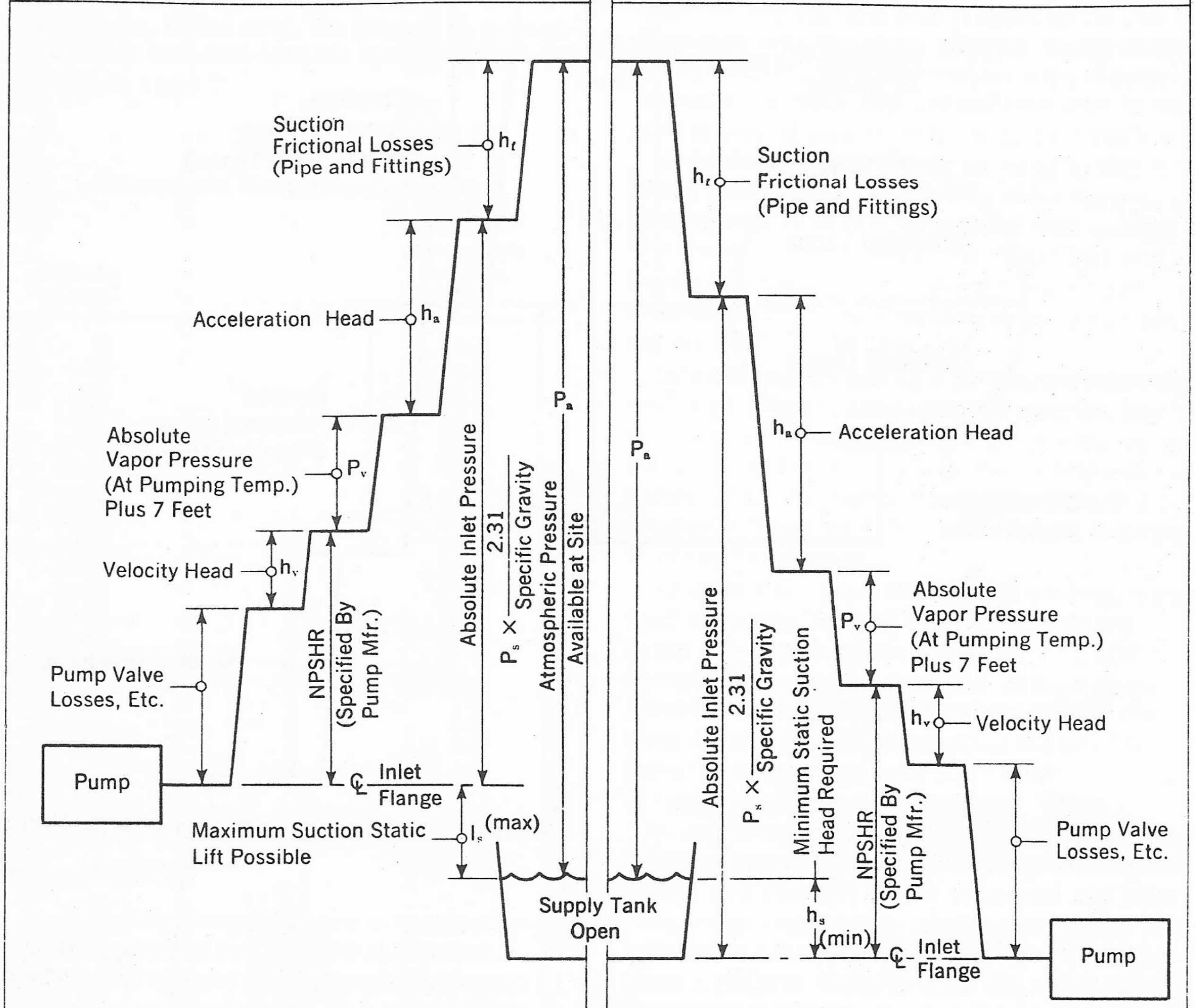
Acceleration head in 6-inch pipe

$$h_{a_6} = \frac{20 \times .811 \times 360 \times .066}{1.4 \times 32.2} = 8.55 \text{ ft}$$

Total acceleration head

$$h_a = 3.88 + 8.55 = 12.43 \text{ ft}$$

SUCTION LIFT SITUATION	SUBMERGED SUCTION SITUATION



SUCTION LIFT SITUATION

I. If $P_a > (NPSHR + P_v + h_f + h_a)$

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

Max. $l_s = P_a - (NPSHR + P_v + h_f + h_a)$

II. If Actual Suction Lift $l_s < l_s$max. then available net positive suction head is—

$NPSHA = P_a - (P_v + h_f + h_a + l_s)$

SUBMERGED SUCTION SITUATION

I. If $P_a < (NPSHR + P_v + h_f + h_a)$

Then, a suction lift is impossible and a positive static suction head is necessary—

h_s minimum $= (NPSHR + P_v + h_f + h_a) - P_a$

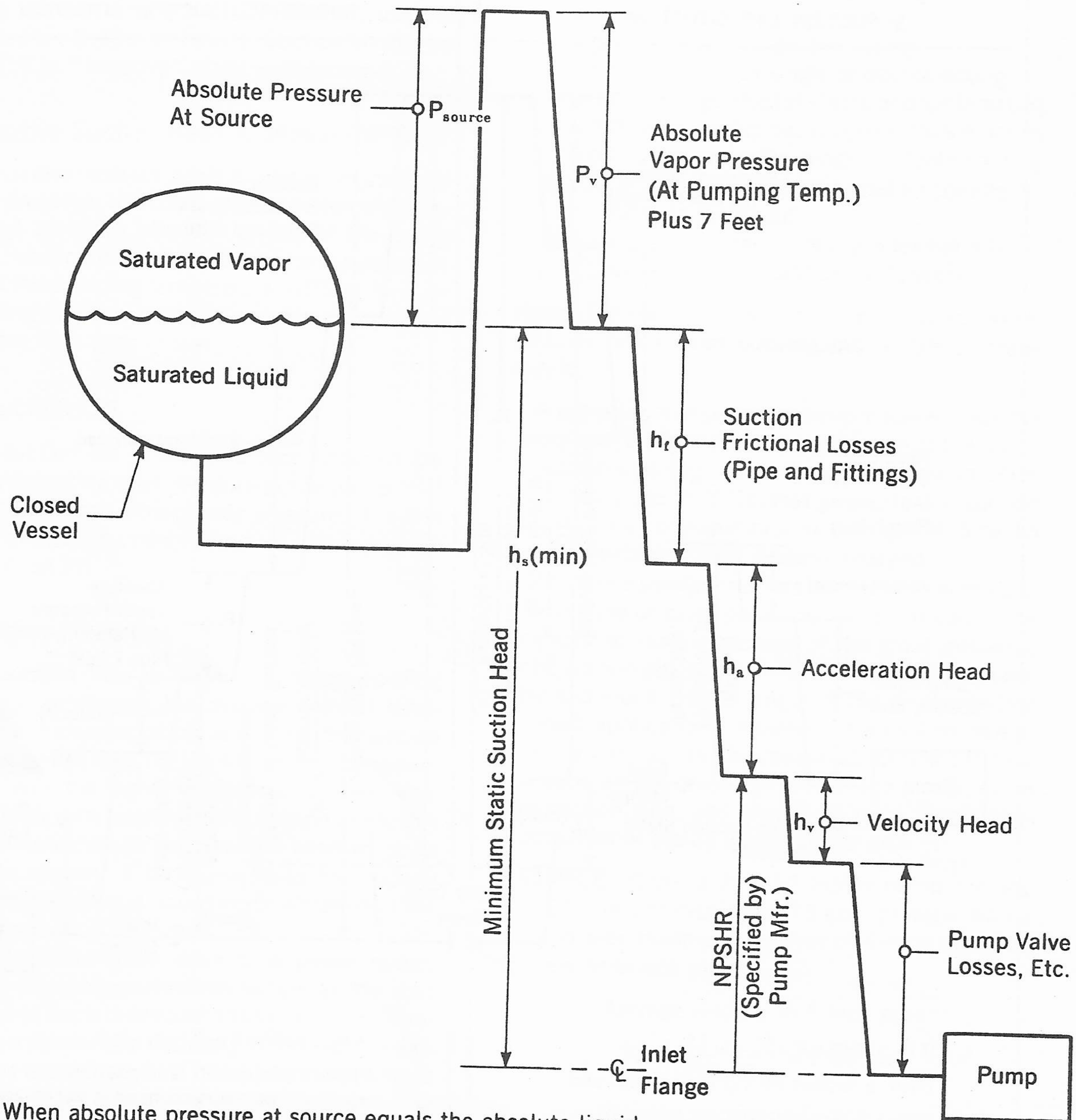
II. If Actual Suction Static Head Exceeds Minimum Req'd. Static Head

h_s actual $> h_s$ minimum

And then—

$NPSHA = (h_s + P_a) - (P_v + h_f + h_a)$

Fig. 55 SUCTION SYSTEM RELATIONSHIPS, OPEN SUPPLY



I. When absolute pressure at source equals the absolute liquid vapor pressure—

$$P_{source} = P_v$$

Then minimum static suction head must equal (or exceed) the sum of all the losses and deductions—

$$H_{s_{min}} = h_f + h_a + NPSHR$$

II. If actual static suction head exceeds the required minimum—

$$h_s > h_{s_{min}}$$

Then—

$$NPSHA = h_s - (h_f + h_a)$$

Fig. 56 SUCTION SYSTEM RELATIONSHIPS, CLOSED SUPPLY

Atmospheric Pressure (P_a)

The absolute pressure of the atmosphere at the pumping site. At sea level, the value of P_a is taken at 14.7 psia and declines with increasing altitude as shown in Table 7.

TABLE 7
Atmospheric Pressure vs Elevation

Altitude (Feet)	Average Atmospheric Pressure (Psia)
0	14.7
500	14.4
1000	14.2
1500	13.9
2000	13.7
2500	13.4
3000	13.2
4000	12.7
5000	12.2
6000	11.8
7000	11.3
8000	10.9
9000	10.5
10000	10.1
12000	9.3
14000	8.6
16000	8.0

Piston and Plunger Pumps for Slurry Service

Typical Service

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

As pipeline systems grow in length and capacity, they require higher pressures. This is advantageous for a reciprocating pump which has the desirable characteristic of maintaining high efficiency at any desired flow rate and pressure. The type used, be it piston or plunger, single or double acting, will depend on the particular application. Developed pressures range up to 3,000 psi with overall efficiencies of 80 to 90 per cent.

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

Reciprocating slurry pumps are so designed that the fluid 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 will still be performing adequately, and will not have worn to the point of causing failure of other parts of the machine.

Hydraulic passages should be sized so that the lowest velocity of the fluid will be above the critical carrying velocity (4 to 6 feet/second 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 6 to 12 feet/second.

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 fluid pumped from flushing fluid can be tolerated.

To protect the main stuffing box packing, clear fluid 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 fluid source to prevent accidental back flow of slurry into the clear fluid system.

Valves for use in slurry service are designed for velocities between 6 and 12 feet/second 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 fluid 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 when used. 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 fluid. This is to take into account the acceleration head of the solids and gases entrained with the fluid. Likewise, pulsation dampeners are usually larger on the suction and discharge side of the pump for the same reasons.

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

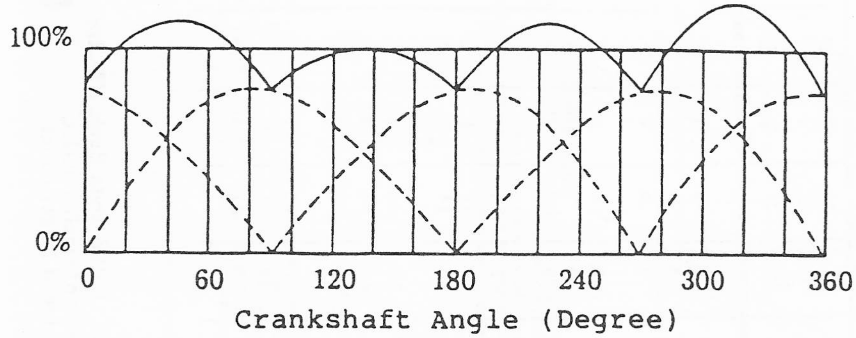
Rod and plunger packing requires special considerations when dealing with abrasive materials. In a piston pump, the piston runs in a renewable metal cylinder or liner. The liners are made of abrasion and corrosion resistant metals to resist wear for each specific slurry. Piston rods and plungers are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

Where the abrasion of the slurry is not great and the pressures are below 2,000 psi, large volume piston pumps are more suitable. The transportation of coal slurry will fall into this class of service.

Appendix B

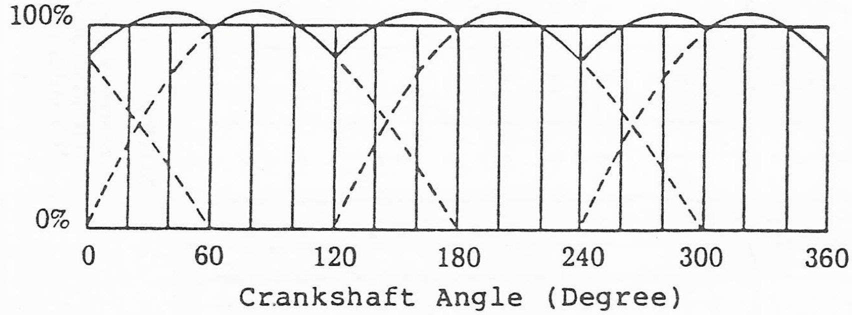
Reciprocating Pumps Flow Characteristics

*2



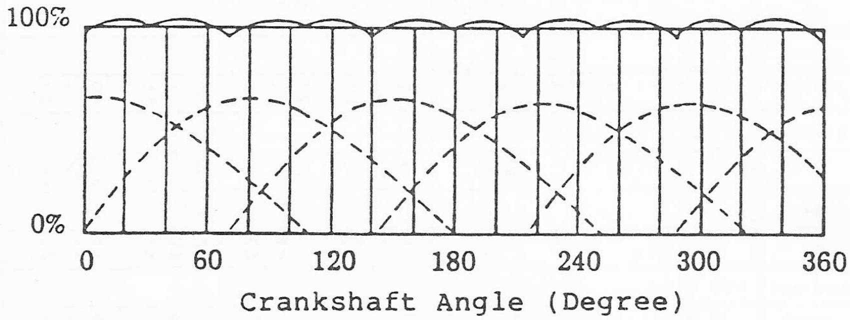
DUPLEX DOUBLE-ACTING

Average Flow -- 100%
Maximum Flow -- 100% + 24%
Minimum Flow -- 100% - 22%
Total Flow Var.-- 46%



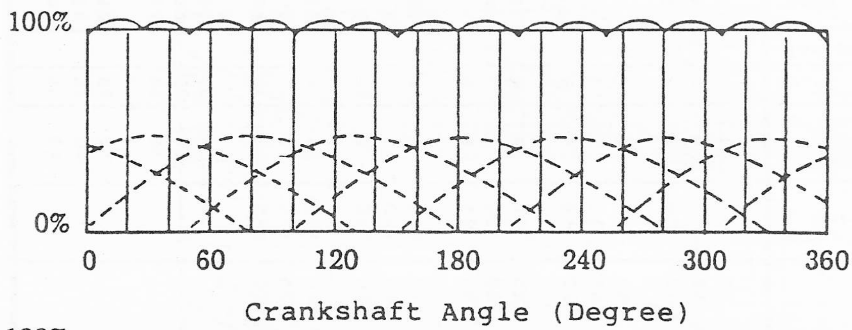
TRIPLEX SINGLE-ACTING

Average Flow -- 100%
Maximum Flow -- 100% + 6%
Minimum Flow -- 100% - 17%
Total Flow Var.-- 23%



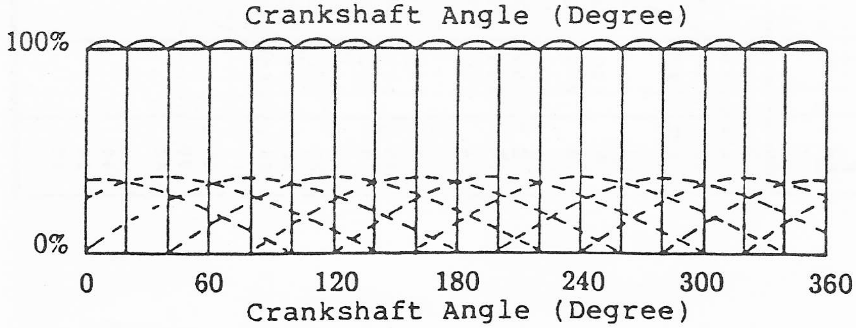
QUINTUPLEX SINGLE-ACTING

Average Flow -- 100%
Maximum Flow -- 100% + 2%
Minimum Flow -- 100% - 5%
Total Flow Var.-- 7%



SEPTUPLEX SINGLE-ACTING

Average Flow -- 100%
Maximum Flow -- 100% + 1.2%
Minimum Flow -- 100% - 2.6%
Total Flow Var.-- 3.8%



NONUPLEX SINGLE-ACTING

Average Flow -- 100%
Maximum Flow -- 100% + 0.6%
Minimum Flow -- 100% - 1.5%
Total Flow Var.-- 2.1%

EQUIVALENT LENGTH OF 100 PERCENT OPENING VALVES AND FITTINGS IN FEET

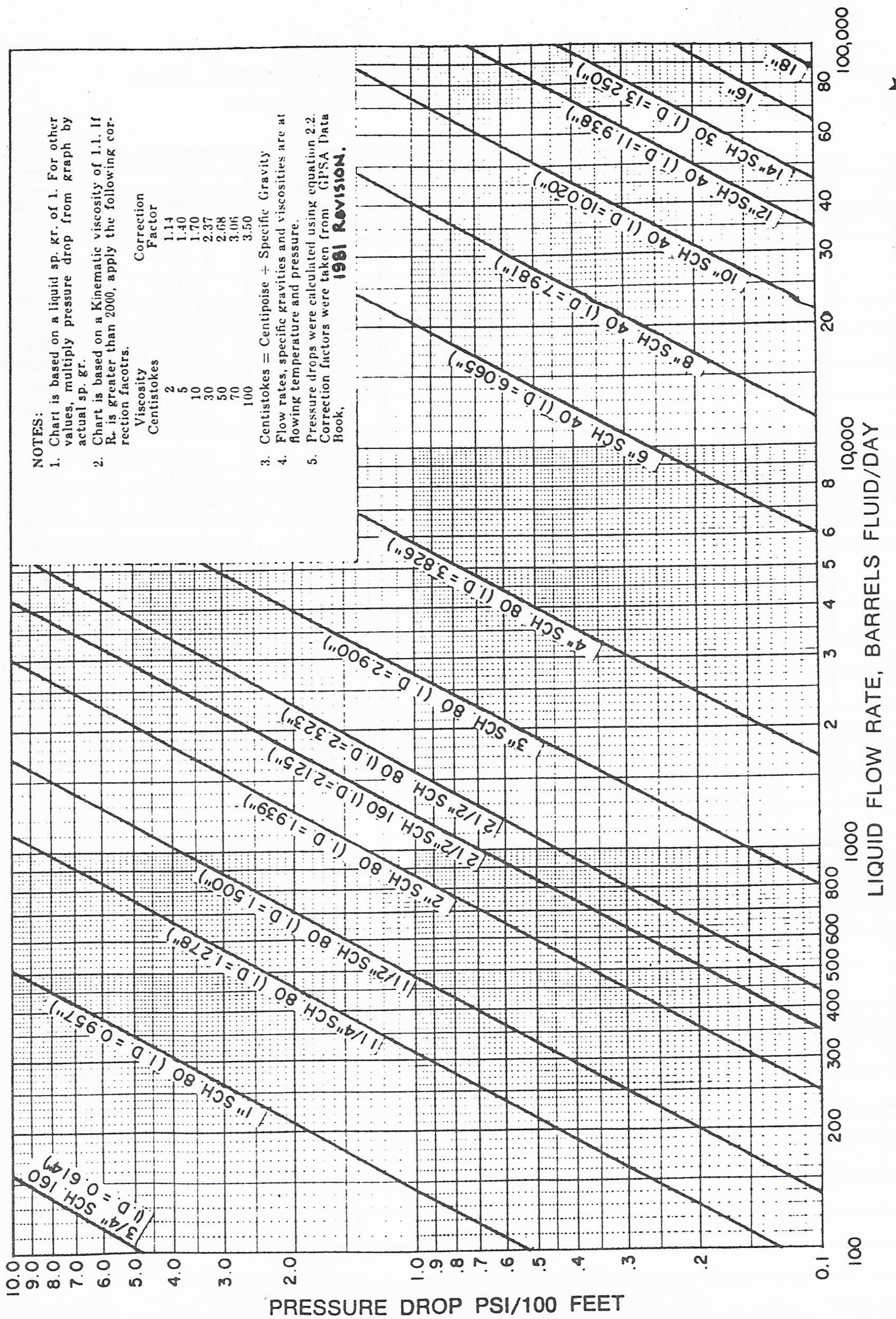
Nominal Pipe Size (Inches)	Globe Valve or Ball Check Valve	Angle Valve	Swing Check Valve	Plug Valve, Gate Valve or Ball Valve	45° Ell		Short Radius Ell		Long Radius Ell		Branch of Tee		Run of Tee		Enlargement					Contraction				
															Sudden			Standard Reducer		Sudden			Standard Reducer	
															Equivalent length in terms of small diameter									
					Weld	Thread	Weld	Thread	Weld	Thread	Weld	Thread	Weld	Thread	d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4	d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4
1 1/2	55	26	13	1	1	2	3	5	2	3	8	9	2	3	5	3	1	4	1	3	2	1	1	
2	70	33	17	2	2	3	4	5	3	4	10	11	3	4	7	4	1	5	1	3	3	1	1	
2 1/2	80	40	20	2	2		5		3		12		3		8	5	2	6	2	4	3	2	2	
3	100	50	25	2	2		6		4		14		4		10	6	2	8	2	5	4	2	2	
4	130	65	32	3	3		7		5		19		5		12	8	3	10	3	6	5	3	3	
6	200	100	48	4	4		11		8		28		8		18	12	4	14	4	9	7	4	4	1
8	260	125	64	6	6		15		9		37		9		25	16	5	19	5	12	9	5	5	2
10	330	160	80	7	7		18		12		47		12		31	20	7	24	7	15	12	6	6	2
12	400	190	95	9	9		22		14		55		14		37	24	8	28	8	18	14	7	7	2
14	450	210	105	10	10		26		16		62		16		42	26	9			20	16	8		
16	500	240	120	11	11		29		18		72		18		47	30	10			24	18	9		
18	550	280	140	12	12		33		20		82		20		53	35	11			26	20	10		
20	650	300	155	14	14		36		23		90		23		60	38	13			30	23	11		
22	688	335	170	15	15		40		25		100		25		65	42	14			32	25	12		
24	750	370	185	16	16		44		27		110		27		70	46	15			35	27	13		
30				21	21		55		40		140		40											
36				25	25		66		47		170		47											
42				30	30		77		55		200		55											
48				35	35		88		65		220		65											
54				40	40		99		70		250		70											
60				45	45		110		80		260		80											

NOTES:

1. Source of data is GPSA Data Book, 1981 Rev.
2. d is inside diameter of smaller outlet.
   D is inside diameter of larger outlet.

*3

*4



PRESSURE DROP IN LIQUID LINES

*5

VAPOR PRESSURE OR SATURATION PRESSURE OF WATER*

Temp. Deg. F.	psi Absolute	Inches Mercury	Feet Of Head	Temp. Deg. F.	psi Absolute	Inches Mercury	Feet Of Head
32	0.08854	0.1806	.2048	112	1.3504	2.7494	3.117
34	0.09610	0.1957	.2219	114	1.4298	2.9111	3.299
36	0.10412	0.2120	.2404	116	1.5130	3.0806	3.490
38	0.11256	0.2291	.2599	118	1.6006	3.2589	3.695
40	0.12170	0.2478	.2810	120	1.6924	3.4458	3.907
42	0.13149	0.3180	.3036	122	1.7888	3.6420	4.121
44	0.14197	0.2890	.3278	124	1.8897	3.8475	4.354
46	0.15319	0.3119	.3537	126	1.9955	4.0629	4.597
48	0.16523	0.3364	.3815	128	2.1064	4.2887	4.853
50	0.17811	0.3626	.4110	130	2.2225	4.5251	5.128
52	0.19182	0.3906	.4428	132	2.3440	4.7725	5.400
54	0.20642	0.4203	.4763	134	2.4712	5.0314	5.693
56	0.2220	0.4520	.5123	136	2.6042	5.3022	6.000
58	0.2386	0.4858	.5504	138	2.7432	5.5852	6.320
60	0.2563	0.5218	.5913	140	2.8886	5.8812	6.666
62	0.2751	0.5601	.6347	142	3.0440	6.1903	7.013
64	0.2951	0.6009	.6809	144	3.1990	6.5132	7.307
66	0.3164	0.6442	.7301	146	3.365	6.850	7.753
68	0.3390	0.6903	.7822	148	3.537	7.202	8.149
70	0.3631	0.7392	.8377	150	3.718	7.569	8.566
72	0.3886	0.7912	.8965	160	4.741	9.653	10.923
74	0.4156	0.8462	.9589	170	5.992	12.200	13.805
76	0.4443	0.9046	1.025	180	7.510	15.290	17.302
78	0.4747	0.9666	1.095	190	9.339	19.014	21.516
80	0.5069	1.0321	1.170	200	11.526	23.467	26.555
82	0.5410	1.1016	1.249	210	14.123	28.755	32.538
84	0.5771	1.1750	1.332	212	14.696	29.921	33.858
86	0.6152	1.2527	1.421	220	17.186	34.991	39.595
88	0.6556	1.3347	1.514	240	24.969	50.837	57.526
90	0.6982	1.4215	1.611	260	35.429	72.134	81.625
92	0.7432	1.5131	1.716	280	49.203	100.178	113.359
94	0.7906	1.6097	1.825	300	67.013	136.439	154.391
96	0.8407	1.7117	1.940	350	134.63	274.108	310.147
98	0.8935	1.8192	2.062	400	247.31	503.526	569.778
100	0.9492	1.9325	2.191	450	422.6	860.418	973.628
102	1.0078	2.0519	2.326	500	680.8	1386.156	1568.495
104	1.0695	2.1775	2.468	550	1045.2	2128.038	2408.036
106	1.1345	2.3099	2.617	600	1542.9	3141.360	3554.687
108	1.2029	2.4491	2.775	700	3093.7	6298.804	7127.575
110	1.2748	2.5955	2.942				

*Specific Gravity of Water at 60 deg. F. = 1.00

materials *6

DATA REQUIRED BY PUMP MANUFACTURERS FOR PROPER SELECTION OF MATERIAL

1. SOLUTION TO BE PUMPED (Give common name, where possible, such as "spinning bath," "black liquor," "spent pickle," etc.) ________________

____

____

2. PRINCIPAL CORROSIVES (H_2SO_4, HCl, etc.) ________________ % by weight
(In the case of mixtures, state definite percentages by weight. For example: mixture contains 2% acid, in terms of 96.5% H_2SO_4.) ________________

____

____

3. pH (if aqueous solution) ________________ at ________________ F
4. IMPURITIES OR OTHER CONSTITUENTS NOT GIVEN IN "2" (List amounts of any metallic salts, such as chlorides, sulphates, sulphides, chromates, and any organic materials which may be present, even though in percentages as low as .01%. Indicate, where practical, whether they act as accelerators or inhibitors on the pump material.) ________________

____

____

5. SPECIFIC GRAVITY (solution pumped) ________________ at ________________ F
6. TEMPERATURE OF SOLUTION: Maximum ________ F, Minimum ________ F, Normal ________ F
7. VAPOR PRESSURES AT ABOVE TEMPERATURES: Maximum ____ Minimum ____ Normal ____
(Indicate units used, such as pounds, gauge, inches water, millimeters mercury.) ________________

____

____

8. VISCOSITY ________________ SSU; or ________________ centistokes; at ________________ F
9. AERATION: Air-Free ________________ Partial ________________ Saturated ________________
Does liquid have tendency to foam? ________________

____

____

10. OTHER GASES IN SOLUTION ________________ ppm, or ________________ cc per liter

____

____

11. SOLIDS IN SUSPENSION: (state types) ________________

____

____

____

Specific gravity of solids ________________

Quantity of solids ________________ % by weight

Particle size ________________ mesh ________________ % by weight

mesh ________________ % by weight

mesh ________________ % by weight

Character of solids: Pulpy ________ Gritty ________ Hard ________ Soft ________

materials

12. CONTINUOUS OR INTERMITTENT SERVICE

Will pump be used for circulation in closed system or for transfer? ______

Will pump be operated at times against closed discharge? ______

If intermittent, how often is pump started? ______ times per ______

Will pump be flushed and drained when not in service? ______

13. TYPE OF MATERIAL IN PIPE LINES TO BE CONNECTED TO PUMP

If desirable, are insulated joints practical? ______

If so, what percentage of element (Fe, Ni, Cu, etc.) is objectionable? ______

14. IS METAL CONTAMINATION UNDESIRABLE? ______

15. PREVIOUS EXPERIENCE. Have you pumped this solution previously? ______

If so, of what material or materials was pump made? ______

Service life in months? ______

In case of trouble, what parts were affected? ______

Was trouble primarily due to corrosion? ______ erosion? ______

galvanic action? ______ stray current? ______

Was attack uniform? ______ If localized, what parts were involved? ______

If galvanic action, name materials involved ______

If pitted, describe size, shape and location (A sketch will be helpful in an analysis of problem.)

16. WHAT IS CONSIDERED AN ECONOMIC LIFE?

(If replacement does not become too frequent, the use of inexpensive pump materials may be the most economical.) ______

Appendix G
Conversion Factors

To convert	Multiply by	To obtain
A		
acre	4047	sq meters
atmospheres	76	cms of mercury (at 0 degree C.)
atmospheres	33.9	ft of water (at 4° C.)
atmospheres	29.92	in. of mercury (at 0° C.)
atmospheres	1.0333	kgs/sq. cm
atmospheres	14.696	pounds/sq. in. (psi)
B		
barrels (U.S. liquid)	31.5	gallons
barrels (oil)	42	gallons
barrels/day (oil)	0.00184	liters/second
barrels/day (oil)	0.029154	gallons/minutes
barrels/day (oil)	0.00662	cubic meter/hour
barrels/hr (oil)	0.7	gallons/minutes
bars	0.9869	atmospheres
bars	14.504	pounds/sq in
bar	1.0197	kg/cm ^{2}
btu	778.02	foot-pounds
C		
centigrade (degrees)	(° c x 9/5) +32	farenheit (degrees)
centiliters	0.3382	ounce (fluid) U.S.
centiliters	0.6103	cubic in.
centimeters	0.03281	feet
centimeters	0.3937	inches
centimeters	0.000006214	miles
centimeters of mercury	0.01316	atmospheres
centimeters of mercury	0.4461	ft. of water
centimeters of mercury	0.1934	pounds/sq. in.
centipoise	0.01	gr/cm-sec
centipoise	0.000672	pound/ft-sec
cubic cms	0.0002642	gallons (U.S.)
cubic foot	7.48052	gallons (U.S.)
cubic foot	28.31702	liters
cubic foot	0.028317	cubic meter
D		
degrees (angle)	0.01745	radians

Appendix G
Conversion Factors

To convert	Multiply by	To obtain
F		
foot	30.48	centimeters
foot	0.3048	meter
foot	0.0001894	mile (stat.)
foot of water	0.0295	atmospheres
foot of water	0.4335	pounds/sq. in (psi)
foot of water	0.03048	kg/cm^2
G		
gallons	378.5	cu. cms.
gallons	0.1337	cu. feet
gallons	231	cu. inches
gallons	0.003785	cu. meters
gallons	0.004951	cu. yards
gallons	3.78533	liters
gallons (liq br imp)	1.20095	gallons (U.S.liquid)
gallons (U.S.)	0.83267	gallons (imp)
gallons of water	8.337	pounds of water
gallons/min	1.42857	barrels/hour
gallons/min	34.3	barrels/day
gallons/min	0.0631	liters/sec
H		
horsepower	42.44	btu/min
horsepower	33,000	foot-lbs/min
horsepower (metric)	0.9863	horsepower
horsepower	1.014	horsepower (metric)
horsepower	10.68	kg-calories/ min
horsepower	0.7457	kilowatts
horsepower	745.7	watts
horsepower (boiler)	33,520	btu/hr
horsepower (boiler)	9.803	kilowatts
horsepower-hours	1,980,000	foot-lbs
I		
inches	2.54	centimeters
inches	0.0254	meters
inches	0.00001578	miles
inches	1000	mils
inches of mercury	0.03342	atmospheres

Appendix G
Conversion Factors

To Convert	Multiply by	To obtain
inches of mercury	0.4912	pounds/sq.in
inches of water (at 4°C)	0.002458	atmosphere
inches of water (at 4°C)	0.03613	pounds/sq.in

K

To Convert	Multiply by	To obtain
kilograms	2.2046	pounds
kilograms	0.0009842	tons (long)
kilograms	0.001102	tons (short)
kilograms/ sq. cm.	0.9678	atmospheres
kilograms/ sq. cm	32.81	feet of water
kilograms/ sq. cm	28.96	inches of mercury
kilograms/ sq. cm	14.22	pounds/ sq. in.
kilowatts	44260	foot-lbs/min.
kilowatts	737.6	foot-lbs/sec
kilowatts	1.341	horsepower
knots	6080	feet/hr

L

To Convert	Multiply by	To obtain
liters	1000	cu. cm
liters	61.02	cu inches
liters	0.2642	gallons (U.S. liquid)
liters	2.113	pints (U.S. liquid)
liters	1.057	quarts (U.S. liquid)
$\log_{10}$ N	2.303	Ln N

M

To Convert	Multiply by	To obtain
meters	100	centimeters
meters	3.281	feet
meters	39.37	inches
meters	0.0006214	miles (stat)
microns	0.000001	meters
miles (statute)	5280	feet
miles (statute)	1.609	kilometer
mils	0.00254	centimeters
mils	0.001	inches

O

To Convert	Multiply by	To obtain
ounces	0.0625	pounds
ounce	28.35	grams
ounces (fluid)	0.02957	liters

Appendix G
Conversion Factors

P

To convert	Multiply by	To obtain
pints (liquid)	0.125	gallons
pounds	0.4536	kilograms
pounds (troy)	0.00036735	tons (long)
pounds (troy)	0.00037324	tons (metric)
pounds (troy)	0.00041143	tons (short)
pounds of water	0.1198	gallons
pounds/ sq. in. (psi)	0.06804	atmosphere
pounds/ sq. in. (psi)	2.307	feet of water
pounds/ sq. in. (psi)	2.036	inches of mercury
pounds/ sq. in. (psi)	0.070307	kgs/sq. cm.
pounds/ sq. in. (psi)	6.895	kpa

Q

To convert	Multiply by	To obtain
quarts (liquid)	0.75	gallons

R

To convert	Multiply by	To obtain
radians	57.296	degrees
revolutions	360	degrees
revolutions/ min.	6	degree/ sec.
revolutions/ min.	0.1047	rad/ sec

S

To convert	Multiply by	To obtain
square in.	6.4516	sq. cms.
square ft.	0.0929	sq. meter
square mi.	2.59	sq.kms.

T

To convert	Multiply by	To obtain
temperature (°F) -32	5/9	temperature (°C)
tons (long)	2240	pounds
tons (long)	1.12	ton (short)
tons (metric)	2205	pounds
tons (short)	2000	pounds
tons (short)	907.2	kilograms

APPENDIX H
USEFUL FORMULAS

1. Liquid Pressure (psi) or (lbf/in^2) = $P = F / A_c$ or F / A_p

2. Pipe Area, Inside (in^2) = $A_c = [\pi \underline{(ID)}^2] / 4$

3. Force due to Pressure (lbf) = $F = P \times A_c$ or $P \times A_p$

4. Liquid Velocity, Avg. (ft/sec) = $V = \dfrac{Q}{2.448 \times (ID)^2}$

5. Pump Displacement (USGPM) = $D = \dfrac{rpm \times gpr \times E_v}{100}$

6. Pump HorsePower Required = $\dfrac{(Q \times P_d) \times (100)}{1714 \times (E_m)} - \dfrac{(Q \times P_i) \times (E_m - 5)}{1714 \times (100)}$

7. Torque (ft-lbs) = $\dfrac{hp \times (5252)}{rpm}$

8. Electric Motor Speed (rpm) = $\dfrac{120 \times \text{frequency (hz)}}{\text{number of poles}}$

9. Static Head of Liquid (ft) = $\dfrac{2.31 \times (P)}{S.G.}$

10. Velocity Head of Liquid (ft) = $V^2 / (2 \times g)$

11. Absolute Viscosity = S.G. (cSt)

12. Kinematic Viscosity = $0.22 \times SSU - \dfrac{180}{SSU}$

13. Absolute Pressure (psia) = atmospheric pressure (@ local elevation) + gauge pressure

14. Gallons per Revolution = $\dfrac{A_p \times \text{stroke length (in)} \times \text{no. plungers}}{231}$

17. Barrels per Day (BPD) = $34.3 \times gpr \times rpm$

18. Specific Gravity @ 60°F (S.G.) = $\dfrac{141.5}{131.5 + \text{API gravity (degree)}}$

Symbol Summary

A_c	=	area of pipe, in^2	where ($A_c = [\pi (ID)^2] / 4$)
A_p	=	area of plunger, in^2	where ($A_p = [\pi (d)^2] / 4$)
BPH	=	barrels per hour	
d	=	plunger diameter, inches (in)	
cP	=	absolute viscosity, centipoise	
cSt	=	kinematic viscosity, centistokes	
D	=	pump displacement, gallons per minute (GPM)	
E_m	=	pump mechanical efficiency, percent (at P_d and P_i)	
E_v	=	volumetric efficiency, percent (%)	
g	=	standard gravity, ft/sec^2 (fps)	
gpr	=	US gallons per revolution	
ID	=	pipe inside diameter, inches (in)	
P_d	=	liquid pressure at pump discharge, lbf/in^2 gauge (psig)	
P_i	=	liquid pressure at pump inlet, lbf/in^2 gauge (psig)	
Q	=	pump capacity, GPM	where ($Q = D / E_v$)
rpm	=	rotational speed in revolutions per minute	
S.G.	=	specific gravity	
SSU	=	viscosity, Saybolt Universal Seconds (SSU)	