

Section TECH-A

Centrifugal Pump Fundamentals

TECH-A-1 Head

The pressure at any point in a liquid can be thought of as being caused by a vertical column of the liquid which, due to its weight, exerts a pressure equal to the pressure at the point in question. The height of this column is called the static head and is expressed in terms of feet of liquid.

The static head corresponding to any specific pressure is dependent upon the weight of the liquid according to the following formula.

$$\text{Head in Feet} = \frac{\text{Pressure in psi} \times 2.31}{\text{Specific Gravity}}$$

A Centrifugal pump imparts velocity to a liquid. This velocity energy is then transformed largely into pressure energy as the liquid leaves the pump. Therefore, the head developed is approximately equal to the velocity energy at the periphery of the impeller. This relationship is expressed by the following well-known formula:

$$H = \frac{v^2}{2g}$$

Where H = Total head developed in feet.

v = Velocity at periphery of impeller in feet per sec.

g = 32.2 Feet/Sec.²

We can predict the approximate head of any centrifugal pump by calculating the peripheral velocity of the impeller and substituting into the above formula. A handy formula for peripheral velocity is:

$$v = \frac{\text{RPM} \times D}{229} \quad \text{Where D = Impeller diameter in inches}$$

The above demonstrates why we must always think in terms of feet of liquid rather than pressure when working with centrifugal pumps. A given pump with a given impeller diameter and speed will raise a liquid to a certain height regardless of the weight of the liquid, as shown in Fig. 1.

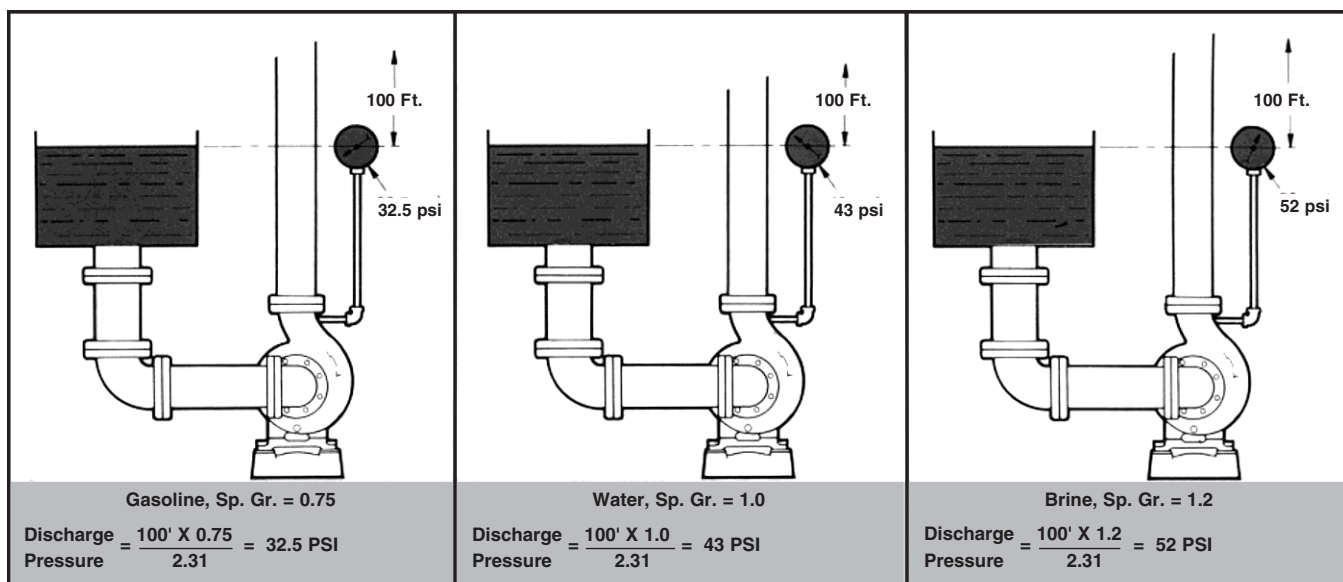


Fig. 1 Identical Pumps Handling Liquids of Different Specific Gravities.

All of the forms of energy involved in a liquid flow system can be expressed in terms of feet of liquid. The total of these various heads determines the total system head or the work which a pump must perform in the system. The various forms of head are defined as follows.

SUCTION LIFT exists when the source of supply is below the centerline of the pump. Thus the **STATIC SUCTION LIFT** is the vertical distance in feet from the centerline of the pump to the free level of the liquid to be pumped.

SUCTION HEAD exists when the source of supply is above the centerline of the pump. Thus the **STATIC SUCTION HEAD** is the vertical distance in feet from the centerline of the pump to the free level of the liquid to be pumped.

STATIC DISCHARGE HEAD is the vertical distance in feet between the pump centerline and the point of free discharge or the surface of the liquid in the discharge tank.

TOTAL STATIC HEAD is the vertical distance in feet between the free level of the source of supply and the point of free discharge or the free surface of the discharge liquid.

The above forms of static head are shown graphically in Fig. 2a & b

FRICTION HEAD (h_f) is the head required to overcome the resistance to flow in the pipe and fittings. It is dependent upon the size and type of pipe, flow rate, and nature of the liquid. Frictional tables are included in section TECH-C.

VELOCITY HEAD (h_v) is the energy of a liquid as a result of its motion at some velocity V . It is the equivalent head in feet through which the water would have to fall to acquire the same velocity, or in other words, the head necessary to accelerate the water. Velocity head can be calculated from the following formula:

$$h_v = \frac{V^2}{2g} \quad \text{where } g = 32.2 \text{ ft/sec.}^2 \\ V = \text{liquid velocity in feet per second}$$

The velocity head is usually insignificant and can be ignored in most high head systems. However, it can be a large factor and must be considered in low head systems.

PRESSURE HEAD must be considered when a pumping system either begins or terminates in a tank which is under some pressure other than atmospheric. The pressure in such a tank must first be converted to feet of liquid. A vacuum in the suction tank or a positive pressure in the discharge tank must be added to the system head, whereas a positive pressure in the suction tank or vacuum in the discharge tank would be subtracted. The following is a handy formula for converting inches of mercury vacuum into feet of liquid.

$$\text{Vacuum, ft. of liquid} = \frac{\text{Vacuum, in. of Hg} \times 1.13}{\text{Sp. Gr.}}$$

The above forms of head, namely static, friction, velocity, and pressure, are combined to make up the total system head at any particular flow rate. Following are definitions of these combined or "Dynamic" head terms as they apply to the pump.

TOTAL DYNAMIC SUCTION LIFT (h_s) is the static suction lift minus the velocity head at the pump suction flange plus the total friction head in the suction line. The total dynamic suction lift, as determined on pump test, is the reading of a gauge on the suction flange, converted to feet of liquid and corrected to the pump centerline*, minus the velocity head at the point of gauge attachment.

TOTAL DYNAMIC SUCTION HEAD (h_s) is the static suction head plus the velocity head at the pump suction flange minus the total friction head in the suction line. The total dynamic suction head, as determined on pump test, is the reading of the gauge on the suction flange, converted to feet of liquid and corrected to the pump centerline*, plus the velocity head at the point of gauge attachment.

TOTAL DYNAMIC DISCHARGE HEAD (h_d) is the static discharge head plus the velocity head at the pump discharge flange plus the total friction head in the discharge line. The total dynamic discharge head, as determined on pump test, is the reading of a gauge at the discharge flange, converted to feet of liquid and corrected to the pump centerline*, plus the velocity head at the point of gauge attachment.

TOTAL HEAD (H) or TOTAL Dynamic HEAD (TDH) is the total dynamic discharge head minus the total dynamic suction head or plus the total dynamic suction lift.

$$\text{TDH} = h_d + h_s \text{ (with a suction lift)}$$

$$\text{TDH} = h_d - h_s \text{ (with a suction head)}$$

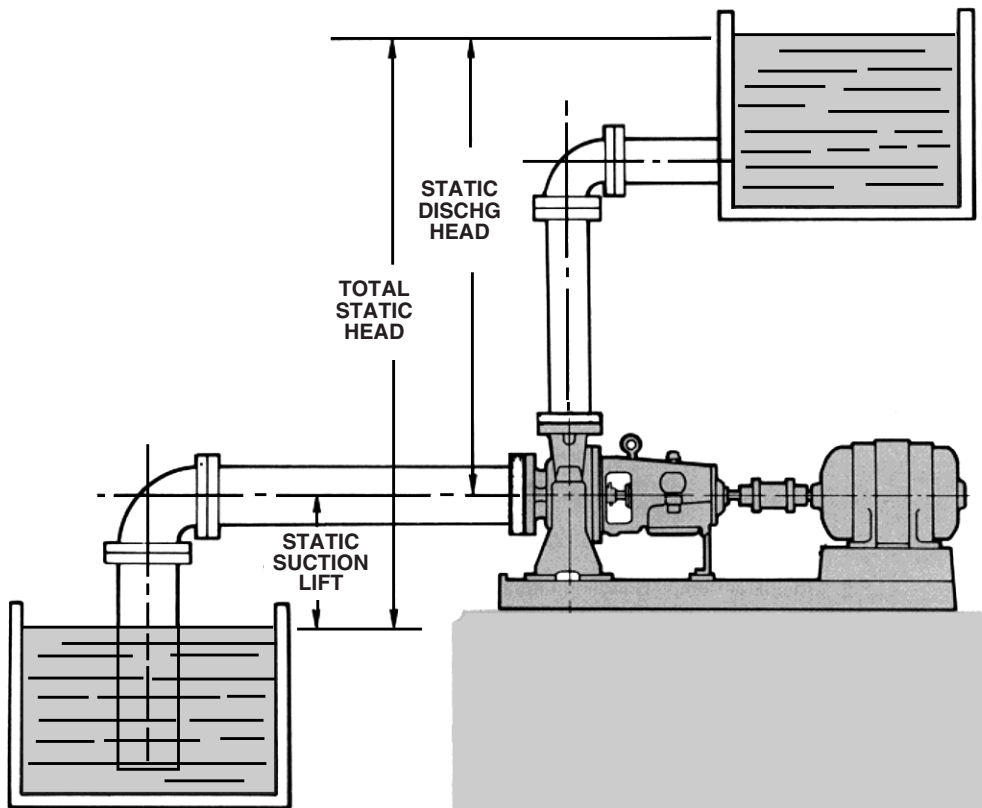


Fig. 2-a Suction Lift –
Showing Static Heads in a Pumping System Where the Pump
is Located Above the Suction Tank. (Static Suction Head)

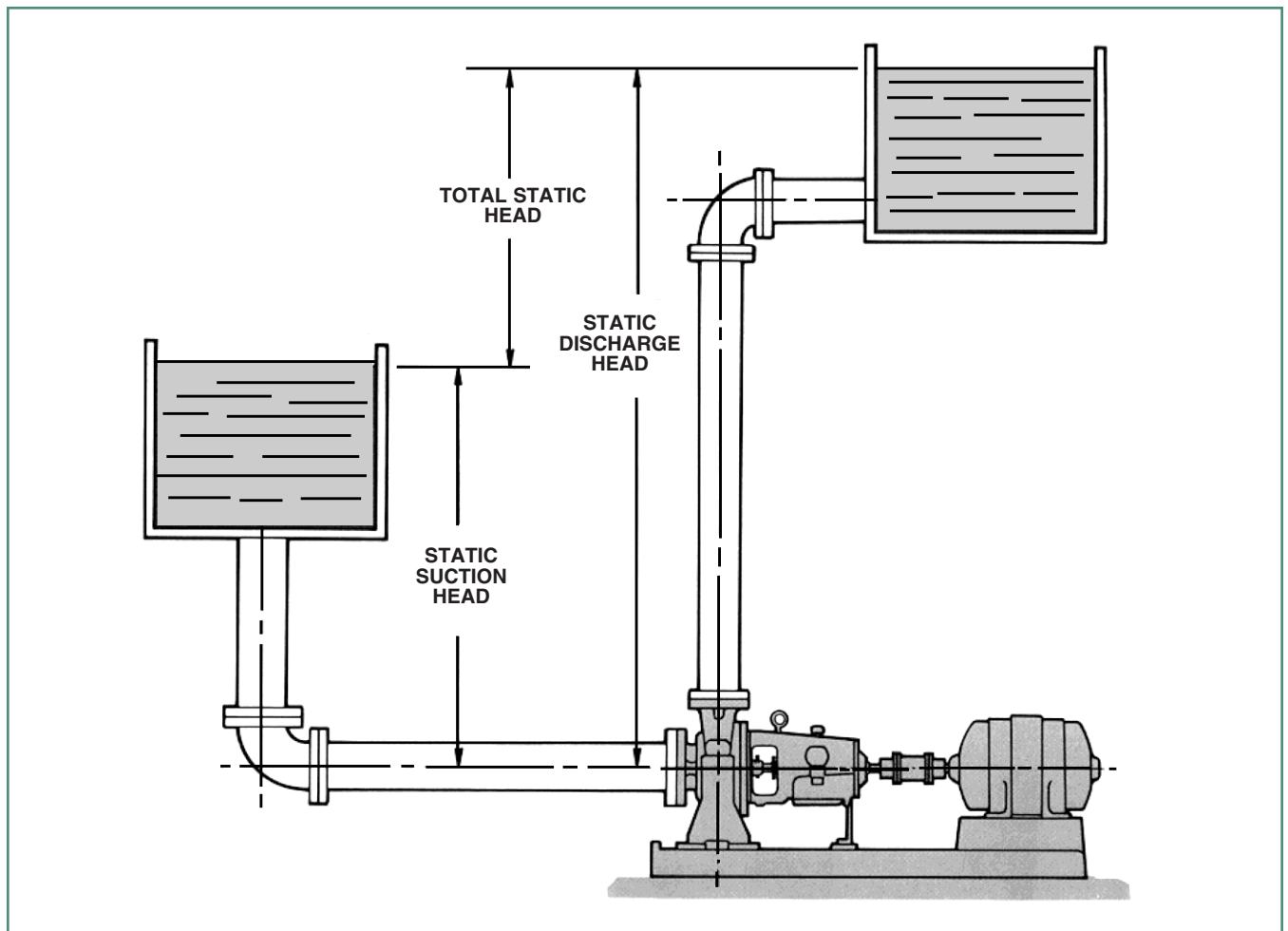


Fig. 2-b Suction Head –
Showing Static Heads in a Pumping System Where the Pump
is Located Below the Suction Tank. (Static Suction Head)

TECH-A-2 Capacity

Capacity (Q) is normally expressed in gallons per minute (gpm). Since liquids are essentially incompressible, there is a direct relationship between the capacity in a pipe and the velocity of flow. This relationship is as follows:

$$Q = A \times V \text{ or } V = \frac{Q}{A}$$

Where A = Area of pipe or conduit in square feet.
V = Velocity of flow in feet per second.

*On vertical pumps the correction should be made to the eye of the suction or lowest impeller.

TECH-A-3 Power and Efficiency

The work performed by a pump is a function of the total head and the weight of the liquid pumped in a given time period. The pump capacity in gpm and the liquid specific gravity are normally used in the formulas rather than the actual weight of the liquid pumped.

Pump input or brake horsepower (bhp) is the actual horsepower delivered to the pump shaft. Pump output or hydraulic horsepower (whp) is the liquid horsepower delivered by the pump. These two terms are defined by the following formulas.

$$\text{whp} = \frac{Q \times \text{TDH} \times \text{Sp. Gr.}}{3960}$$

$$\text{bhp} = \frac{Q \times \text{TDH} \times \text{Sp. Gr.}}{3960 \times \text{Pump Efficiency}}$$

The constant 3960 is obtained by dividing the number of foot pounds for one horsepower (33,000) by the weight of one gallon of water (8.33 pounds.)

The brake horsepower or input to a pump is greater than the hydraulic horsepower or output due to the mechanical and hydraulic losses incurred in the pump. Therefore the pump efficiency is the ratio of these two values.

$$\text{Pump Eff} = \frac{\text{whp}}{\text{bhp}} = \frac{Q \times \text{TDH} \times \text{Sp. Gr.}}{3960 \times \text{bhp}}$$

TECH-A-4 Specific Speed and Pump Type

Specific speed (N_s) is a non-dimensional design index used to classify pump impellers as to their type and proportions. It is defined as the speed in revolutions per minute at which a geometrically similar impeller would operate if it were of such a size as to deliver one gallon per minute against one foot head.

The understanding of this definition is of design engineering significance only, however, and specific speed should be thought of only as an index used to predict certain pump characteristics. The following formula is used to determine specific speed:

$$N_s = \frac{N \sqrt{Q}}{H^{3/4}}$$

Where N = Pump speed in RPM

Q = Capacity in gpm at the best efficiency point

H = Total head per stage at the best efficiency point

The specific speed determines the general shape or class of the impeller as depicted in Fig. 3. As the specific speed increases, the ratio of the impeller outlet diameter, D_2 , to the inlet or eye diameter, D_1 , decreases. This ratio becomes 1.0 for a true axial flow impeller.

Radial flow impellers develop head principally through centrifugal force. Pumps of higher specific speeds develop head partly by centrifugal force and partly by axial force. A higher specific speed indicates a pump design with head generation more by axial forces and less by centrifugal forces. An axial flow or propeller pump with a specific speed of 10,000 or greater generates its head exclusively through axial forces.

Radial impellers are generally low flow high head designs whereas axial flow impellers are high flow low head designs.

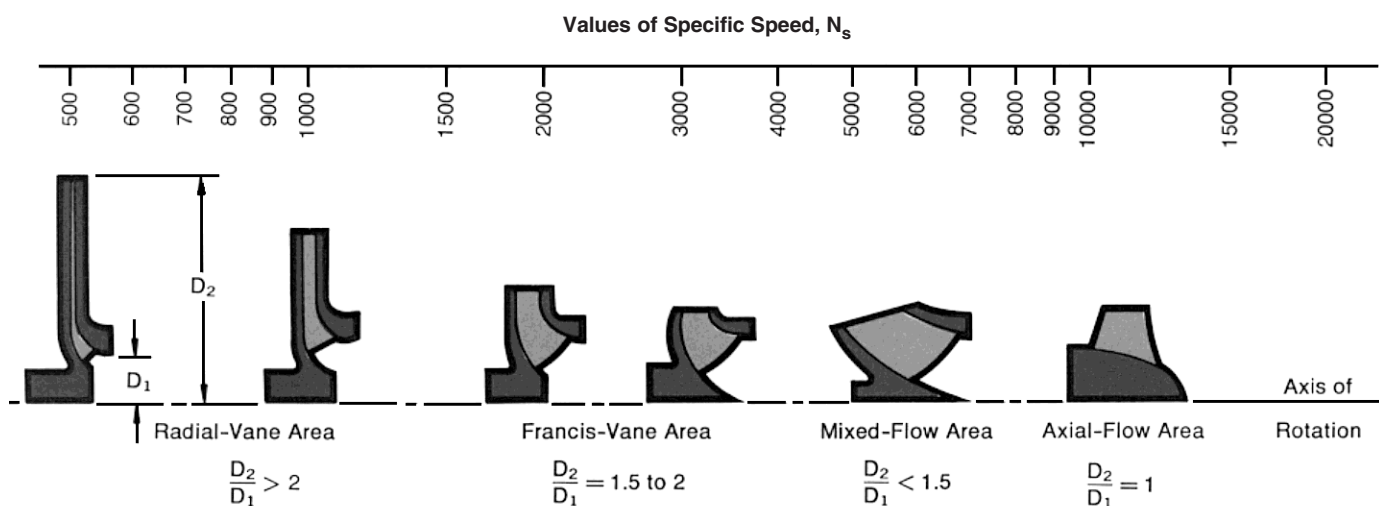


Fig. 3 Impeller Design vs Specific Speed

TECH-A-5 Net Positive Suction Head (NPSH) and Cavitation

The Hydraulic Institute defines NPSH as the total suction head in feet absolute, determined at the suction nozzle and corrected to datum, less the vapor pressure of the liquid in feet absolute. Simply stated, it is an analysis of energy conditions on the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump.

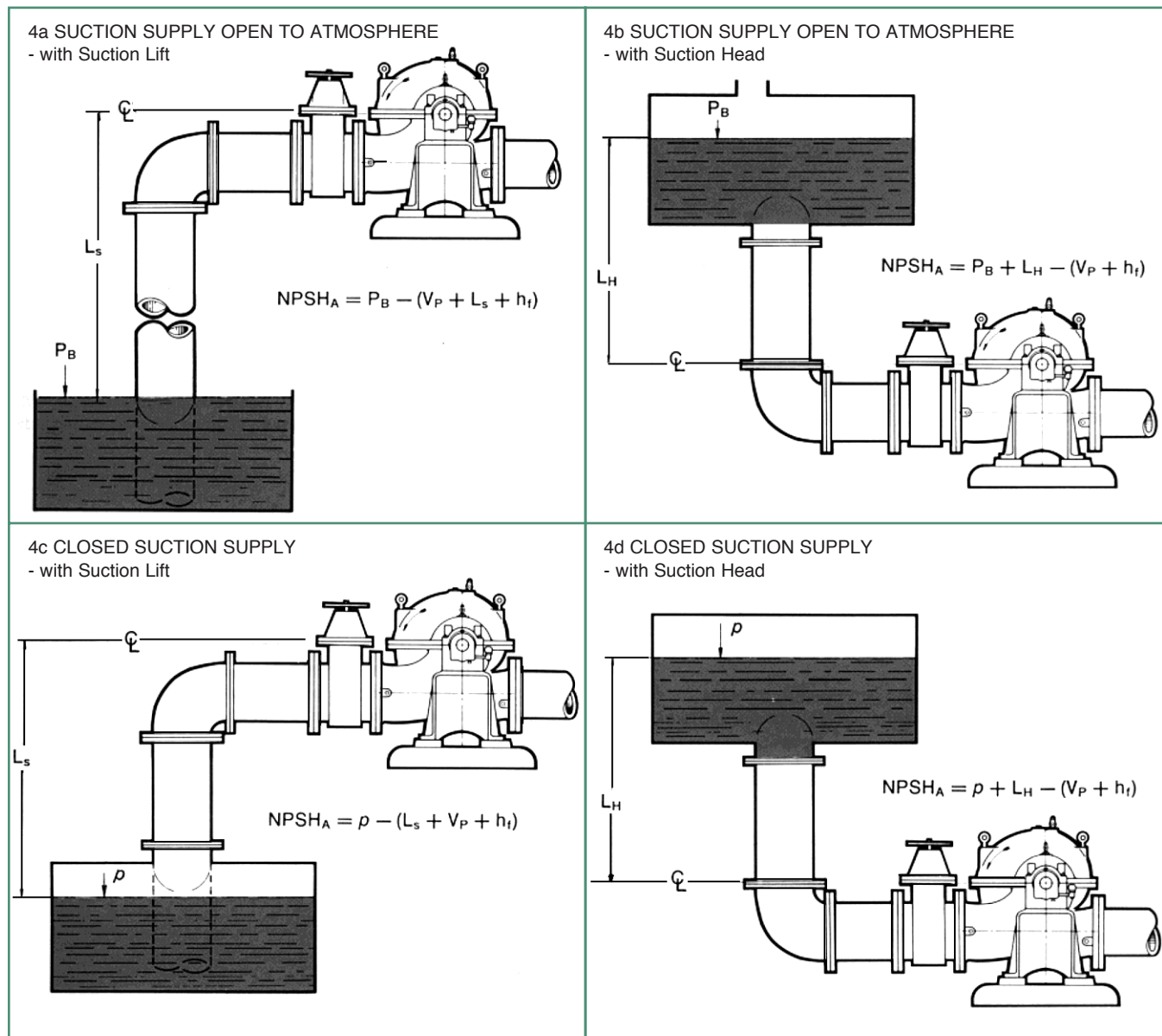
The pressure which a liquid exerts on its surroundings is dependent upon its temperature. This pressure, called vapor pressure, is a unique characteristic of every fluid and increases with increasing temperature. When the vapor pressure within the fluid reaches the pressure of the surrounding medium, the fluid begins to vaporize or boil. The temperature at which this vaporization occurs will decrease as the pressure of the surrounding medium decreases.

A liquid increases greatly in volume when it vaporizes. One cubic foot of water at room temperature becomes 1700 cu. ft. of vapor at the same temperature.

It is obvious from the above that if we are to pump a fluid effectively, we must keep it in liquid form. NPSH is simply a measure of the amount of suction head present to prevent this excess vaporization at the lowest pressure point in the pump.

NPSH Required is a function of the pump design. As the liquid passes from the pump suction to the eye of the impeller, the velocity increases and the pressure decreases. There are also pressure losses due to shock and turbulence as the liquid strikes the impeller. The centrifugal force of the impeller vanes further increases the velocity and decreases the pressure of the liquid. The NPSH Required is the positive head in feet absolute required at the pump suction to overcome these pressure drops in the pump and maintain enough of the liquid above its vapor pressure to limit the head loss, due to the blockage of the cavitation vapor bubble, to 3 percent. The 3% head drop criteria for NPSH Required is used worldwide and is based on the ease of determining the exact head drop off point. Most standard low suction energy pumps can operate with little or no margin above the NPSH Required, without seriously affecting the service life of the pump. The NPSH Required varies with speed and capacity within any particular pump. Pump manufacturer's curves normally provide this information.

NPSH Available is a function of the system in which the pump operates. It is the excess pressure of the liquid in feet absolute over its vapor pressure as it arrives at the pump suction. Fig. 4 shows four typical suction systems with the NPSH Available formulas applicable to each. It is important to correct for the specific gravity of the liquid and to convert all terms to units of "feet absolute" in using the formulas.



P_B = Barometric pressure, in feet absolute.

V_P = Vapor pressure of the liquid at maximum pumping temperature, in feet absolute.

p = Pressure on surface of liquid in closed suction tank, in feet absolute.

L_s = Maximum static suction lift in feet.

L_H = Minimum static suction head in feet.

h_f = Friction loss in feet in suction pipe at required capacity

Fig. 4 Calculation of system Net Positive Suction Head Available for typical suction conditions.

In an existing system, the NPSH Available can be determined by a gauge on the pump suction. The following formula applies:

$$\text{NPSH}_A = P_B - V_p \pm Gr + h_v$$

Where Gr = Gauge reading at the pump suction expressed in feet (plus if above atmospheric, minus if below atmospheric) corrected to the pump centerline.

h_v = Velocity head in the suction pipe at the gauge connection, expressed in feet.

Cavitation is a term used to describe the phenomenon, which occurs in a pump when there is insufficient NPSH Available. The pressure of the liquid is reduced to a value equal to or below its vapor pressure and small vapor bubbles or pockets begin to form. As these vapor bubbles move along the impeller vanes to a higher pressure area, they rapidly collapse.

The collapse, or "implosion" is so rapid that it may be heard as a rumbling noise, as if you were pumping gravel. In high suction energy pumps, the collapses are generally high enough to cause minute

pockets of fatigue failure on the impeller vane surfaces. This action may be progressive, and under severe (very high suction energy) conditions can cause serious pitting damage to the impeller.

The accompanying noise is the easiest way to recognize cavitation. Besides possible impeller damage, excessive cavitation results in reduced capacity due to the vapor present in the pump. Also, the head may be reduced and/or be unstable and the power consumption may be erratic. Vibration and mechanical damage such as bearing failure can also occur as a result of operating in excessive cavitation, with high and very high suction energy pumps.

The way to prevent the undesirable effects of cavitation in standard low suction energy pumps is to insure that the NPSH Available in the system is greater than the NPSH Required by the pump. High suction energy pumps require an additional NPSH margin, above the NPSH Required. Hydraulic Institute Standard (ANSI/HI 9.6.1) suggests NPSH margin ratios of from 1.2 to 2.5 times the NPSH Required, for high and very high suction energy pumps, when operating in the allowable operating range.

TECH-A-6 NPSH Suction Specific Speed and Suction Energy

In designing a pumping system, it is essential to provide adequate NPSH available for proper pump operation. Insufficient NPSH available may seriously restrict pump selection, or even force an expensive system redesign. On the other hand, providing excessive NPSH available may needlessly increase system cost.

Suction specific speed may provide help in this situation.

Suction specific speed (S) is defined as:

$$S = \frac{N (\text{GPM})^{1/2}}{(\text{NPSH}_R)^{3/4}}$$

Where **N** = Pump speed RPM

GPM = Pump flow at best efficiency point at impeller inlet (for double suction impellers divide total pump flow by two).

NPSH = Pump NPSH required at best efficiency point.

For a given pump, the suction specific speed is generally a constant - it does not change when the pump speed is changed. Experience has shown that 9000 is a reasonable value of suction specific speed. Pumps with a minimum suction specific speed of 9000 are readily available, and are not normally subject to severe operating restrictions.

An example:

Flow 2,000 GPM; head 600 ft. What NPSH will be required?

Assume: at 600 ft., 3550 RPM operation will be required.

$$S = \frac{N (\text{GPM})^{1/2}}{(\text{NPSH}_R)^{3/4}}$$

$$9000 = \frac{3550 (2000)^{1/2}}{(\text{NPSH}_R)^{3/4}}$$

$$\text{NPSH}_R^{3/4} = 17.7$$

$$\text{NPSH}_R = 46 \text{ ft.}$$

A related problem is in selecting a new pump, especially at higher flow, for an existing system. Suction specific speed will highlight applications where NPSH_A may restrict pump selection. An example:

Existing system: Flow 2000 GPM; head 600 ft.: NPSH_A 30 ft. What is the maximum speed at which a pump can be run without exceeding NPSH available?

$$S = \frac{N (\text{GPM})^{1/2}}{(\text{NPSH})^{3/4}}$$

$$9000 = \frac{N (2000)^{1/2}}{30^{3/4}}$$

$$N = 2580 \text{ RPM}$$

Running a pump at this speed would require a gear and at this speed, the pump might not develop the required head. At a minimum, existing NPSH_A is constraining pump selection.

Same system as 1. Is a double suction pump practical?

For a double suction pump, flow is divided by two.

$$S = \frac{N (\text{GPM})^{1/2}}{(\text{NPSH})^{3/4}}$$

$$9000 = \frac{N (1000)^{1/2}}{(30)^{3/4}}$$

$$N = 3700 \text{ RPM}$$

Using a double suction pump is one way of meeting system NPSH.

The amount of energy in a pumped fluid, that flashes into vapor and then collapses back to a liquid in the higher pressure area of the impeller inlet, determines the extent of the noise and/or damage from cavitation. Suction Energy is defined as:

$$\text{Suction Energy} = D_e \times N \times S \times S_g$$

Where **D_e** = Impeller eye diameter (inches)

S_g = Specific gravity of liquid (S_g - 1.0 for cold water)

High Suction Energy starts at 160×10^6 for end suction pumps and 120×10^6 for horizontal split case pumps. Very high suction energy starts at 1.5 times the High Suction Energy values. For estimating purposes you can normally assume that the impeller eye diameter is approximately 90% of the suction nozzle size, for an end suction pump, and 75% of the suction size for a double suction split case pump.

An example:

Suction specific speed 9,000, pump speed 3550 RPM, suction nozzle size 6 inch, specific gravity 1.0, and the pump type is end suction.

$$D_e \approx .9 \times 6" = 5.4"$$

$$\begin{aligned} \text{Suction Energy} &= D_e \times N \times S \times S_g \\ &= 5.4 \times 3550 \times 9,000 \times 1.0 \\ &= 173 \times 10^6 \end{aligned}$$

Since $173 \times 10^6 > 160 \times 10^6$, this is a High Suction Energy pump.

TECH-A-7 Pump Characteristic Curves

The performance of a centrifugal pump can be shown graphically on a characteristic curve. A typical characteristic curve shows the total dynamic head, brake horsepower, efficiency, and net positive suction head all plotted over the capacity range of the pump.

Figures 5, 6, & 7 are non-dimensional curves which indicate the general shape of the characteristic curves for the various types of pumps. They show the head, brake horsepower, and efficiency plotted as a percent of their values at the design or best efficiency point of the pump.

Fig. 5 shows that the head curve for a radial flow pump is relatively flat and that the head decreases gradually as the flow increases. Note that the brake horsepower increases gradually over the flow range with the maximum normally at the point of maximum flow.

Mixed flow centrifugal pumps and axial flow or propeller pumps have considerably different characteristics as shown in Figs. 6 and 7. The head curve for a mixed flow pump is steeper than for a radial flow

pump. The shut-off head is usually 150% to 200% of the design head. The brake horsepower remains fairly constant over the flow range. For a typical axial flow pump, the head and brake horsepower both increase drastically near shutoff as shown in Fig. 7.

The distinction between the above three classes is not absolute, and there are many pumps with characteristics falling somewhere between the three. For instance, the Francis vane impeller would have a characteristic between the radial and mixed flow classes. Most turbine pumps are also in this same range depending upon their specific speeds.

Fig. 8 shows a typical pump curve as furnished by a manufacturer. It is a composite curve which tells at a glance what the pump will do at a given speed with various impeller diameters from maximum to minimum. Constant horsepower, efficiency, and NPSH_R lines are superimposed over the various head curves. It is made up from individual test curves at various diameters.

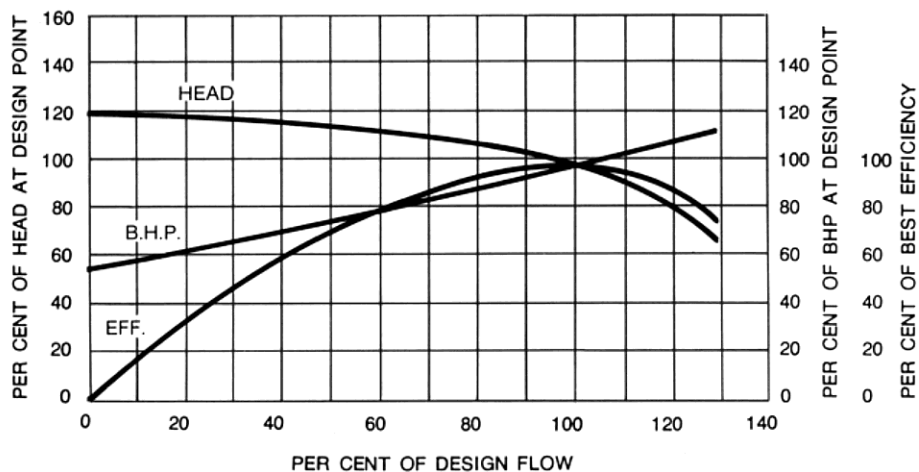


Fig. 5 Radial Flow Pump

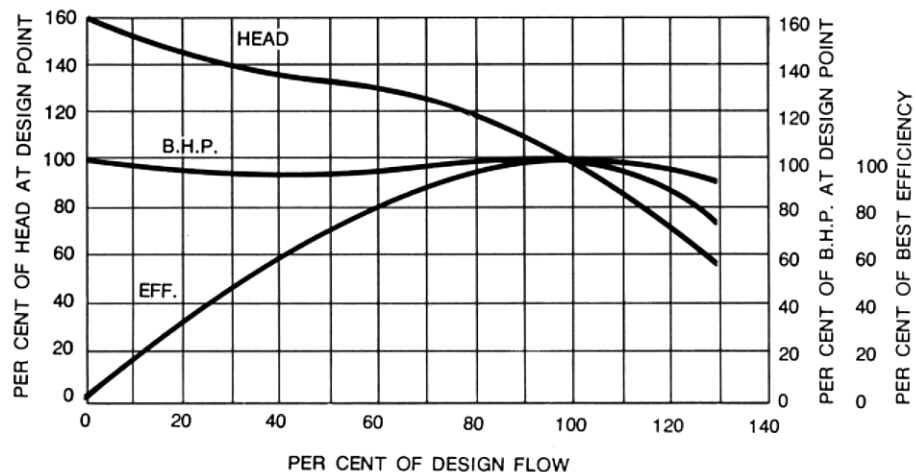


Fig. 6 Mixed Flow Pump

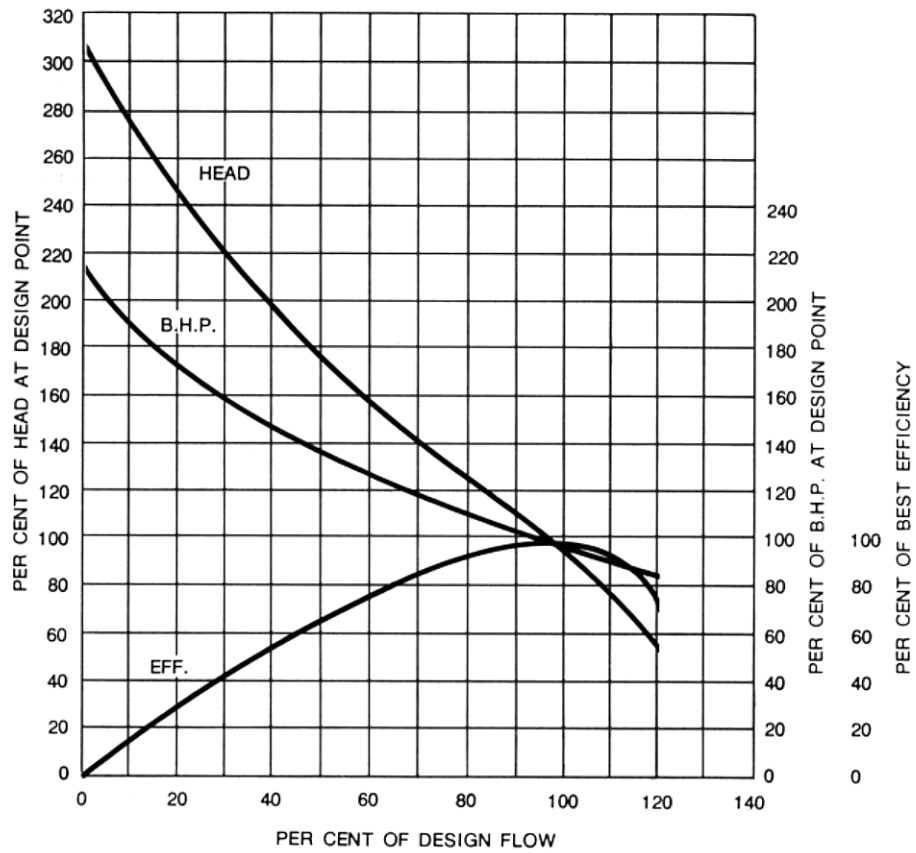


Fig. 7 Axial Flow Pump

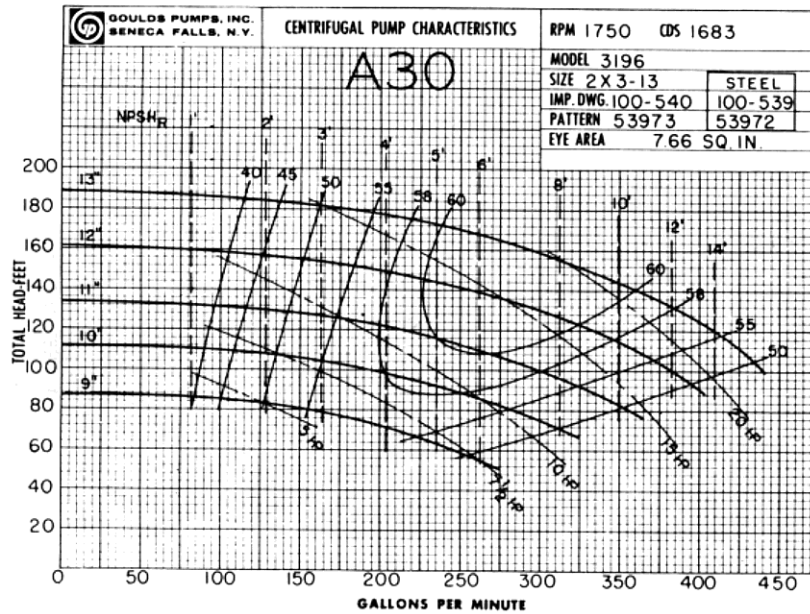


Fig. 8 Composite Performance Curve

TECH-A-8 Affinity Laws

The affinity laws express the mathematical relationship between the several variables involved in pump performance. They apply to all types of centrifugal and axial flow pumps. They are as follows:

1. With impeller diameter, D, held constant:

Where: Q = Capacity, GPM
H = Total Head, Feet
BHP = Brake Horsepower
N = Pump Speed, RPM

A. $\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$

B. $\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$

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

2. With speed, N, held constant:

A. $\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$

B. $\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$

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

When the performance (Q_1 , H_1 , & BHP_1) is known at some particular speed (N_1) or diameter (D_1), the formulas can be used to estimate the performance (Q_2 , H_2 , & BHP_2) at some other speed (N_2) or diameter (D_2). The efficiency remains nearly constant for speed changes and for small changes in impeller diameter.

EXAMPLE:

To illustrate the use of these laws, refer to Fig. 8. It shows the performance of a particular pump at 1750 RPM with various impeller diameters. This performance data has been determined by actual tests by the manufacturer. Now assume that you have a 13" maximum diameter impeller, but you want to belt drive the pump at 2000 RPM.

The affinity laws listed under 1 above will be used to determine the new performance, with $N_1 = 1750$ RPM and $N_2 = 2000$ RPM. The first step is to read the capacity, head, and horsepower at several points on the 13" dia. curve in Fig. 9. For example, one point may be near the best efficiency point where the capacity is 300 GPM, the head is 160 ft, and the BHP is approx. 20 hp.

$$\frac{300}{Q_2} = \frac{1750}{2000} \quad Q_2 = 343 \text{ gpm}$$

$$\frac{160}{H_2} = \left(\frac{1750}{2000}\right)^2 \quad H_2 = 209 \text{ ft.}$$

$$\frac{20}{BHP_2} = \left(\frac{1750}{2000}\right)^3 \quad BHP_2 = 30 \text{ hp}$$

This will then be the best efficiency point on the new 2000 RPM curve. By performing the same calculations for several other points on the 1750 RPM curve, a new curve can be drawn which will approximate the pump's performance at 2000 RPM, Fig. 9.

Trial and error would be required to solve this problem in reverse. In other words, assume you want to determine the speed required to make a rating of 343 GPM at a head of 209 ft. You would begin by selecting a trial speed and applying the affinity laws to convert the desired rating to the corresponding rating at 1750 RPM. When you arrive at the correct speed, 2000 RPM in this case, the corresponding 1750 RPM rating will fall on the 13" diameter curve.

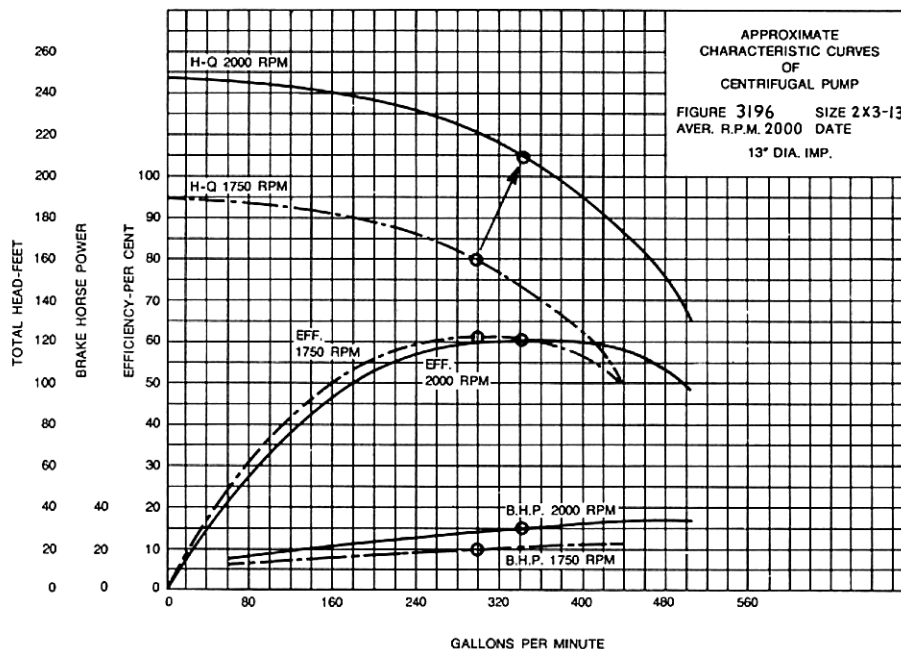
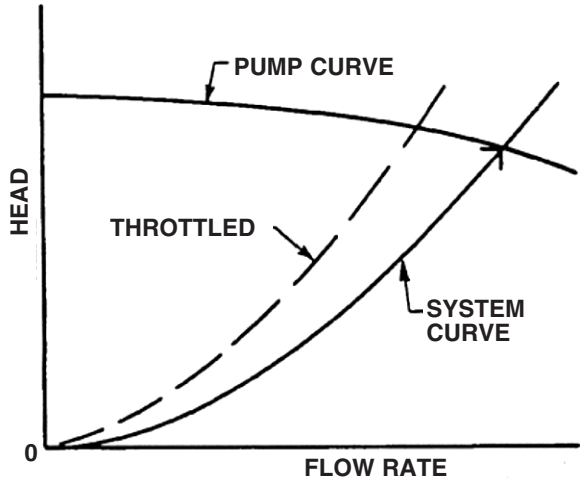


Fig. 9

TECH-A-9 System Curves

For a specified impeller diameter and speed, a centrifugal pump has a fixed and predictable performance curve. The point where the pump operates on its curve is dependent upon the characteristics of the system in which it is operating, commonly called the System Head Curve...or, the relationship between flow and hydraulic losses* in a system. This representation is in a graphic form and, since friction losses vary as a square of the flow rate, the system curve is parabolic in shape.



By plotting the system head curve and pump curve together, it can be determined:

1. Where the pump will operate on its curve.
2. What changes will occur if the system head curve or the pump performance curve changes.

NO STATIC HEAD – ALL FRICTION

As the levels in the suction and discharge are the same (Fig. 1), there is no static head and, therefore, the system curve starts at zero flow and zero head and its shape is determined solely from pipeline losses. The point of operation is at the intersection of the system head curve and the pump curve. The flow rate may be reduced by throttling valve.

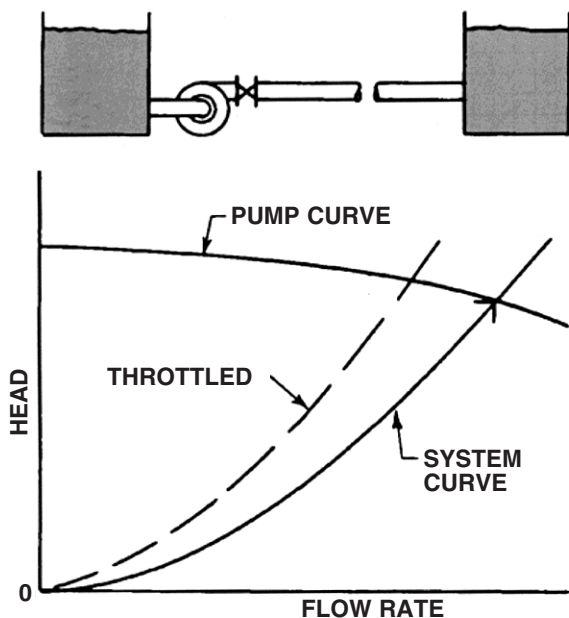


Fig. 1 No Static Head - All Friction

POSITIVE STATIC HEAD

The parabolic shape of the system curve is again determined by the friction losses through the system including all bends and valves. But in this case there is a positive static head involved. This static head does not affect the shape of the system curve or its "steepness", but it does dictate the head of the system curve at zero flow rate.

The operating point is at the intersection of the system curve and pump curve. Again, the flow rate can be reduced by throttling the discharge valve.

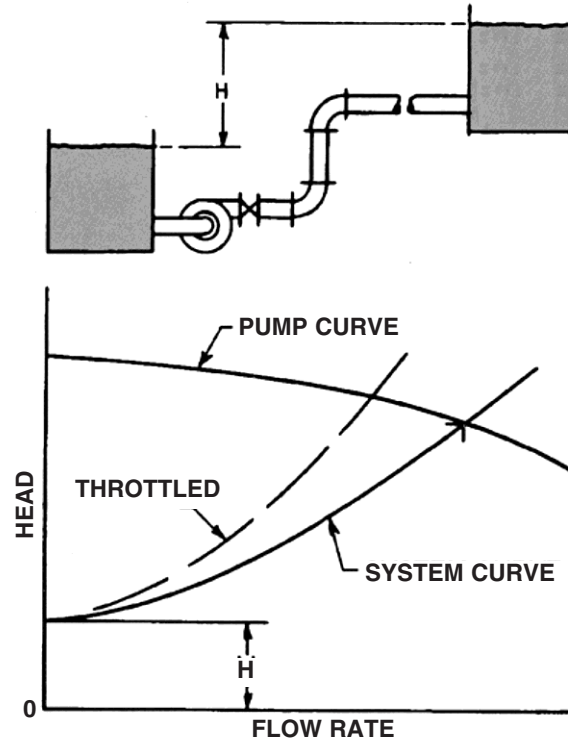


Fig. 2 Positive Suction Head

* Hydraulic losses in piping systems are composed of pipe friction losses, valves, elbows and other fittings, entrance and exit losses (these to the entrance and exit to and from the pipeline normally at the beginning and end – not the pump) and losses from changes in pipe size by enlargement or reduction in diameter.

NEGATIVE (GRAVITY) HEAD

In this illustration, a certain flow rate will occur by gravity head alone. But to obtain higher flows, a pump is required to overcome the pipe friction losses in excess of "H" – the head of the suction above the level of the discharge. In other words, the system curve is plotted exactly as for any other case involving a static head and friction head, except the static head is now negative. The system curve begins at a negative value and shows the limited flow rate obtained by gravity alone. More capacity requires extra work.

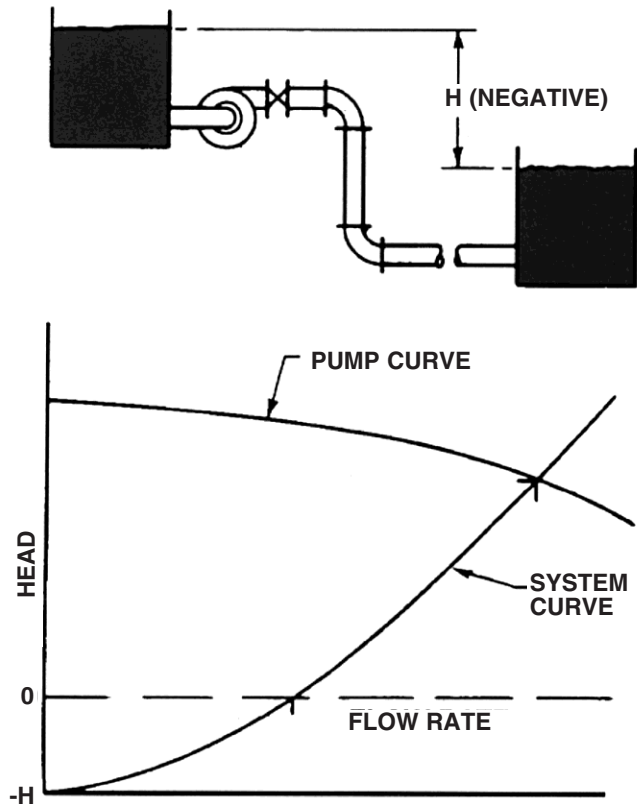


Fig. 3 Negative (Gravity) Head

MOSTLY LIFT- LITTLE FRICTION HEAD

The system head curve in this illustration starts at the static head "H" and zero flow. Since the friction losses are relatively small (possibly due to the large diameter pipe), the system curve is "flat". In this case, the pump is required to overcome the comparatively large static head before it will deliver any flow at all.

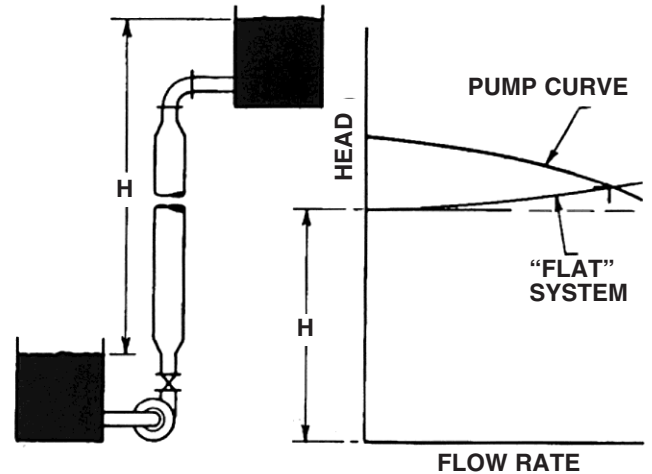


Fig. 4 Mostly Lift - Little Friction Head

TECH-A-10 Basic Formulas and Symbols

Formulas

$$\text{GPM} = \frac{0.002 \times \text{Lb./Hr.}}{\text{Sp. Gr.}}$$

$$\text{GPM} = \frac{\text{Lbs./Hr.}}{500 \times \text{Sp. Gr.}}$$

$$\text{GPM} = 449 \times \text{CFS}$$

$$\text{GPM} = 0.7 \times \text{BBL/Hr.}$$

$$H = \frac{2.31 \times \text{psi}}{\text{Sp. Gr.}}$$

$$H = \frac{1.134 \times \text{In. Hg.}}{\text{Sp. Gr.}}$$

$$h_v = \frac{V^2}{2g} = .0155 \ V^2$$

$$V = \frac{\text{GPM} \times 0.321}{A} = \frac{\text{GPM} \times 0.409}{(\text{I.D.})^2}$$

$$\text{BHP} = \frac{\text{GPM} \times H \times \text{Sp. Gr.}}{3960 \times \text{Eff.}} = \frac{\text{GPM} \times \text{psi}}{1715 \times \text{Eff.}}$$

$$\text{Eff.} = \frac{\text{GPM} \times H \times \text{Sp. Gr.}}{3960 \times \text{BHP}}$$

$$\text{Sp. Gr.} = \frac{141.5}{131.5 \times \text{degrees A.P.I.}}$$

$$N_c = \frac{187.7}{\sqrt{f}}$$

$$f = \frac{PL^3}{mEI}$$

$$N_s = \frac{N \sqrt{\text{GPM}}}{H^{3/4}}$$

$$H = \frac{v^2}{2g}$$

$$v = \frac{N \times D}{229}$$

$$\text{DEG. C} = (\text{DEG. F} - 32) \times 5 / 9$$

$$\text{DEG. F} = (\text{DEG. C} \times 5 / 9) + 32$$

Symbols

GPM = gallons per minute

CFS = cubic feet per second

Lb. = pounds

Hr. = hour

BBL = barrel (42 gallons)

Sp. Gr. = specific gravity

H = head in feet

psi = pounds per square inch

In. Hg. = inches of mercury

h_v = velocity head in feet

V = velocity in feet per second

g = 32.16 ft/sec² (acceleration of gravity)

A = area in square inches

I.D. = inside diameter in inches

BHP = brake horsepower

Eff. = pump efficiency expressed as a decimal

N_s = specific speed

N = speed in revolutions per minute

v = peripheral velocity of an impeller in feet per second

D = Impeller in inches

N_c = critical speed

f = shaft deflection in inches

P = total force in lbs.

L = bearing span in inches

m = constant usually between 48 and 75 for pump shafts

E = modules of elasticity, psi – 27 to 30 million for steel

*SEE SECTION TECH-D-8C FOR SLURRY FORMULAS



Section TECH-B

Pump Application Data

TECH-B-1 Corrosion & Materials of Construction

Selecting the right pump type and sizing it correctly are critical to the success of any pump application. Equally important is the selection of materials of construction. Choices must be made between metals and/or non-metals for pump components that come into contact with the pumpage. In addition, gaskets and O-ring material selections must be made to assure long leak-free operation of the pump's dynamic and static sealing joints. To assist in proper selection, included in this section is a brief discussion of specific types of corrosion and a general material selection guide.

Corrosion

Corrosion is the destructive attack of a metal by chemical or electrochemical reaction with its environment. It is important to understand the various types of corrosion and factors affecting corrosion rate to properly select materials.

TYPES OF CORROSION

(1) Galvanic corrosion is the electro-chemical action produced when one metal is in electrical contact with another more noble metal, with both being immersed in the same corroding medium called the electrolyte. A galvanic cell is formed and current flows between the two materials. The least noble material called the anode will corrode while the more noble cathode will be protected. It is important that the smaller wearing parts in a pump be of a more noble material than the larger more massive parts, as in an iron pump with bronze or stainless steel trim.

Following is a galvanic series listing the more common metals and alloys.

Corroded End

(Anodic, or least noble)
Magnesium
Magnesium Alloys
Zinc
Aluminum 2S
Cadmium
Aluminum 175T
Steel or Iron
Cast Iron
Stainless Steel, 400 Series (Active)
Stainless Steel, Type 304 (Active)
Stainless Steel, Type 316 (Active)
Lead-tin Solders
Lead
Tin
Nickel (Active)

Nickel base alloy (active)
Brasses
Copper
Bronzes
Copper-Nickel Alloy
Monel
Silver Solder
Nickel (Passive)
Nickel Base Alloy (Passive)
Stainless Steel, 400 Series (Passive)
Stainless Steel, Type 304 (Passive)
Stainless Steel, Type 316 (Passive)
Silver
Graphite
Gold
Platinum Protected End
(Cathodic, or most noble)

(2) Uniform Corrosion is the overall attack on a metal by a corroding liquid resulting in a relatively uniform metal loss over the exposed surface. This is the most common type of corrosion and it can be minimized by the selection of a material which offers resistance to the corroding liquid.

(3) Intergranular corrosion is the precipitation of chromium carbides at the grain boundaries of stainless steels. It results in the complete destruction of the mechanical properties of the steel for the depth of the attack. Solution annealing or the use of extra low carbon stainless steels will eliminate intergranular corrosion.

(4) Pitting Corrosion is a localized rather than uniform type of attack. It is caused by a breakdown of the protective film and results in rapid pit formation at random locations on the surface.

(5) Crevice or Concentration Cell Corrosion occurs in joints or small surface imperfections. Portions of the liquid become trapped and a difference in potential is established due to the oxygen concentration difference in these cells. The resulting corrosion may progress rapidly leaving the surrounding area unaffected.

(6) Stress Corrosion is the failure of a material due to a combination of stress and corrosive environment, whereas the material would not be affected by the environment alone.

(7) Erosion-Corrosion is the corrosion resulting when a metal's protective film is destroyed by high velocity fluids. It is distinguished from abrasion which is destruction by fluids containing abrasive solid particles.

pH VALUES

The pH of a liquid is an indication of its corrosive qualities, either acidic or alkaline. It is a measure of the hydrogen or hydroxide ion concentration in gram equivalents per liter. pH value is expressed as the logarithm to the base 10 of the reciprocal of the hydrogen ion concentration. The scale of pH values is from zero to 14, with 7 as a neutral point. From 6 to zero denotes increasing hydrogen ion concentration and thus increasing acidity, and from 8 to 14 denotes increasing hydroxide ion concentration and thus increasing alkalinity.

The table below outlines materials of construction usually recommended for pumps handling liquids of known pH value

pH Value	Material of Construction
10 to 14	Corrosion Resistant Alloys
8 to 10 6 to 8 4 to 6	Iron, Stainless Steel, Bronze, Carbon Steel
0 to 4	Corrosion Resistant Alloys

The pH value should only be used as a guide with weak aqueous solutions. For more corrosive solutions, temperature and chemical composition should be carefully evaluated in the selection of materials of construction.

TECH-B-2 Material Selection Chart

This chart is intended as a guide in the selection of economical materials. It must be kept in mind that corrosion rates may vary widely with temperature, concentration, and the presence of trace elements or abrasive solids. Blank spaces in the chart indicate a lack of accurate corrosion data for those specific conditions. In general, the chart is limited to metals and non-metals regularly furnished by ITT-Goulds.

Note: Maximum temperature limits are shown where data is available. Contact a Goulds representative for temperature limits of all materials before final material selection.

Code:

A Recommended
B Useful resistance
X Unsuitable
Steel Carbon steel, cast iron and ductile iron
Brz Bronze
316 Stainless steel
A-20 Carpenter stainless
CD4MCu CD4MCu stainless steel
Alloy 2205 Alloy 2205 stainless steel
C-276 Wrought Hastelloy® C-276 alloy
Ti Titanium unalloyed
Zi Zirconium
ETFE Ethylenetetrafluoroethylene (Tefzel®)
FP Fluoropolymers (e.g.,

FRP Teflon®) including perfluoroalkoxy (PFA), polytetrafluoroethylene (PTFE) and fluorinated ethylene propylene (FEP)
EPDM Fiber-reinforced plastic (vinylester resin)
FKM1 Ethylenepropylene rubber (Norden®)
FKM2 Standard grades; dipolymers of hexafluoropropylene (HFP) and vinylidene fluoride (VF₂) (Viton®)
Specialty grades; terpolymers comprising at least three of the following: HFP, VF₂, tetrafluoroethylene (TFE), perfluoromethylvinyl ether

(PMVE) or ethylene (E). Specialty grades may have significantly improved chemical compatibility compared to standard grades in many harsh chemical environments (Viton®).
FFKM Copolymer of TFE and PMVE (Kalrez®)
PVDF Polyvinylidene fluoride (Kynar®, Solef®)

¹Compatibility is dependent on specific freon. Contact elastomer manufacturer.

Corrosive	Steel	Brz	316	A-20	CD4MCu	ALLOY 2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF
Acetaldehyde, 70°F	B	A	A	A	A	A	A	A	A	A	A	X	A	X	X	A	X
Acetic acid, 70°F	X	A	A	A	A	A	A	A	A	A	A	X	A	X	B	A	A
Acetic acid, <50%, to boiling	X	B	A	A	B	A	A	A	A				A	X	B	A	B
Acetic acid, >50%, to boiling	X	X	B	A	X	A	A	A	A	104°C	A	X	B	X	B	A	X
Acetone, to boiling	A	A	A	A	A	A	A	A	A	104°C	A	X	A	X	X	A	X
Aluminum chloride, <10%, 70°F	X	B	X	B	X	B		B	A	A	A	A	A	A	A	A	
Aluminum chloride, >10%, 70°F	X	X	X	B	X	B		B	A	A	A	A	A	A	A	A	A (to 40°C)
Aluminum chloride, <10%, to boiling	X	X	X	X	X	X		X	A	104°C	A	X	A	A	A	A	A
Aluminum chloride, >10%, to boiling	X	X	X	X	X	X	X	X	A	104°C	A	X	A	A	A	A	A (to 40°C)
Aluminum sulphate, 70°F	X	B	A	A	A	A	B	A	A	A	A	A	A	A	A	A	A
Aluminum sulphate, <10%, to boiling	X	B	B	A	B	A	A	A	A	104°C	A		A	A	A	A	A
Aluminum sulphate, >10%, to boiling	X	X	X	B	X	B	B	X	B	104°C	A		A	A	A	A	A
Ammonium chloride, 70°F	X	X	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Ammonium chloride, <10%, to boiling	X	X	B	B	X	B	A	A	A	104°C	A		A	A	A	A	A
Ammonium chloride, >10%, to boiling	X	X	X	X	X	X	X	X	X	104°C	A		A	A	A	A	A
Ammonium fluosilicate, 70°F	X	X	X	B	X	B	X	X	X							A	
Ammonium sulphate, <40%, to boiling	X	X	B	B	X	B	B	A	A	104°C	A		A	X	B	A	A
Arsenic acid, to 225°F	X	X	X	B	X	B				A	A		A	A	A	A	A
Barium chloride, 70°F <30%	X	B	X	B	X	B	B	B	B	A	A	A	A	A	A	A	A
Barium chloride, <5%, to boiling	X	B	X	B	X	B	B	A	A	104°C	A		A	A	A	A	A
Barium chloride, >5%, to boiling	X	X	X	X	X	X	X	X	X	104°C	A		A	A	A	A	A
Barium hydroxide, 70°F	B	X	A	A	A	A	B	A	A	A	A	A	A	A	A	A	A
Barium nitrate, to boiling	X	X	B	B	B	B		B	B	104°C	A					A	A
Barium sulphide, 70°F	X	X	B	B	B	B		A	A	A	A	A	A	B	A	A	A
Benzoic acid	X	X	B	B	B	B	A	A	A	A	A		X	A	A	A	110°C
Boric acid, to boiling	X	X	B	B	B	B	A	B	B	104°C	A		A	A	A	A	A
Boron trichloride, 70°F dry	B	B	B	B	B	B	B									A	A
Boron trifluoride, 70°F 10%, dry	B	B	B	A	B	A	A						X	X	X	B	A
Brine (acid), 70°F	X	X	X	X	X	X	B	B		A	A	A	A	A	A	A	A
Bromine (dry), 70°F	X	X	X	X	X	X	B	X	X	A	A	X	X	A	A	A	A
Bromine (wet), 70°F	X	X	X	X	X	X	B	X	X	A	A	X	X	B	A	A	A
Calcium bisulphite, 70°F	X	X	B	B	B	B	B	A	A	A	A		X	A	A	A	A
Calcium bisulphite	X	X	X	B	X	B	X	A	A	A	A		X	A	A	A	95°C
Calcium chloride, 70°F	B	X	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Calcium chloride <5%, to boiling	X	X	B	B	B	B	A	A	A	104°C	A		A	A	A	A	A
Calcium chloride >5%, to boiling	X	X	X	B	X	B	A	B	B	104°C	A		A	B	A	A	A
Calcium hydroxide, 70°F	B	B	B	B	B	B	A	A		A	A	A	A	A	A	A	A
Calcium hydroxide, <30%, to boiling	X	B	B	B	B	B	A	A		104°C	A		A	B	A	A	A
Calcium hydroxide, >30%, to boiling	X	X	X	X	X	X	B	A		104°C	A		A	B	A	A	A

Corrosive	Steel	Brz	316	A-20	CD4MCu	ALLOY 2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF
Calcium hypochlorite, <2%, 70°F	X	X	X	X	X	X	A	A	A	A	A	X	B	A	A	A	A
Calcium hypochlorite, >2%, 70°F	X	X	X	X	X	X	B	A	B	A	A	X	B	A	A	A	A
Carbolic acid, 70°F (phenol)	X	B	A	A	A	A	A	A	A	A	A		B	B	A	A	50°C
Carbon bisulphide, 70°F	B	B	A	A	A	A		A		A	A		X	A	A	A	
Carbonic acid, 70°F	B	X	A	A	A	A	A	A	A	A	A		A	A	A	A	A
Carbon tetrachloride, dry to boiling	B	B	A	A	A	A	B	A	A	104°C	149°C		X	B	A	A	A
Chloric acid, 70°F	X	X	X	B	X	B	X			A	A					A	A
Chlorinated water, 70°F	X	X	B	B	B	B	A	A	A	A	A		A	A	A	A	
Chloroacetic acid, 70°F	X		X	X		X		A	B	A	A	A	A	X	B	A	X
Chlorosulphonic acid, 70°F	X	X	X	X	X	X	A	B	X	A	A	A	X	X	X	A	X
Chromic acid, <30%	X	X	X	B	X	B	B	A	A	65°C	A	X	X	A	A	A	80°C
Citric acid	X	X	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Copper nitrate, to 175°F	X	X	B	B	B	B	X	B		A	A	A				A	A
Copper sulphate, to boiling	X	X	X	B	X	B	A	A	A	104°C	A		A	A	A	A	A
Cresylic acid	X	X	B	B	B	B	B			A	A		X	A	A	A	65°C
Cupric chloride	X	X	X	X	X	X	X	B	X	A	A			A	A	A	A
Cyanohydrin, 70°F	X		B	B	B	B				A	A	A		X	X	A	
Dichloroethane	X	B	B	B	B	B	B	A	B	65°C	A			B	A	A	A
Diethylene glycol, 70°F	A	B	A	A	A	A	B	A	A	A	A		A	B	A	A	A
Dinitrochlorobenzene, 70°F (dry)	X	B	A	A	A	A	A	A	A	A	A			X	B	A	A
Ethanolamine, 70°F	B	X	B	B	B	B		A	A	A	A		B	X	X	A	X
Ethers, 70°F	B	B	B	A	A	A	B	A	A	A	A		X	X	X	A	B
Ethyl alcohol, to boiling	A	A	A	A	A	A	A	A	A	104°C	A		A	B	A	A	A
Ethyl cellulose, 70°F	A	B	B	B	B	B	B	A	A	A	A		B	X	X	A	
Ethyl chloride, 70°F	X	B	B	A	B	A	B	A	A	A	A	X	X	B	A	A	A
Ethyl mercaptan, 70°F	X	X	B	A	B	A	B			A	A	X	X	B	B	A	A
Ethyl sulphate, 70°F	X	B	B	A	B	A				A	A	X		X	X	A	
Ethylene chlorohydrin, 70°F	X	B	B	B	B	B	B	A	A	A	A	X	B	A	A	A	A
Ethylene dichloride, 70°F	X	B	B	B	B	B	X	A	A	A	A	X	X	A	A	A	A
Ethylene glycol, 70°F	B	B	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Ethylene oxide, 70°F	X	X	B	B	B	B	A	A	A	A	A		X	X	X	A	A
Ferric chloride, <5%, 70°F	X	X	X	X	X	X	A	A	B	A	A	A	A	A	A	A	A
Ferric chloride, >5%, 70°F	X	X	X	X	X	X	B	B	X	A	A	X	A	A	A	A	A
Ferric nitrate, 70°F	X	X	B	A	B	A	B			A	A	A	A	A	A	A	A
Ferric sulphate, 70°F	X	X	X	B	X	B	B	B	B	A	A	A	A	A	A	A	A
Ferrous sulphate, 70°F	X	X	X	B	X	B	B	A	A	A	A	A		A	A	A	A
Formaldehyde, to boiling	B	B	A	A	A	A	B	A	A	104°C	A		A	X	B	A	X
Formic acid, to 212°F	X	X	X	A	B	A	A	X	A	A	A		A	X	X	A	A
Freon, 70°F	A	A	A	A	A	A	A	A	A	A	A		A/X ¹	A/X ¹	A/X ¹	A/B ¹	A
Hydrochloride acid, <1%, 70°F	X	X	X	B	X	B	A	B	A	A	A	A	A	A	A	A	A
Hydrochloric acid, 1% to 20%, 70°F	X	X	X	X	X	X	A	X	A	A	A	A	A	A	A	A	A
Hydrochloric acid, >20%, 70°F	X	X	X	X		X	A	X	B	A	A	X	A	B	A	A	A
Hydrochloric acid, <1/2%, 175°F	X	X	X	X	X	X	A	X	A	A	A	X	X	B	A	A	A
Hydrochloric acid, 1/2% to 2%, 175°F	X	X	X		X		A	X	A	A	A	X	X	B	A	A	A
Hydrocyanic acid, 70°F	X	X	X	B	X	B	X			A	A	A	A	A	A	A	A
Hydrogen peroxide, <30%, <150°F	X	X	B	B	B	B	B	A	A	A	A		B	B	A	A	A
Hydrofluoric acid, <20%, 70°F	X	B	X	B	X	B	A	X	X	A	A		X	B	A	A	A
Hydrofluoric acid, >20%, 50°F	X	X	X	X	X	X	B	X	X	A	A		X	B	A	A	A
Hydrofluoric acid, to boiling	X	X	X	X	X	X	X	X	X				X	X	B	A	B
Hydrofluorsilicic acid, 70°F	X		X	B	X	B	B			A	A		B	A	A	A	A
Lactic acid, <50%, 70°F	X	B	A	A	A	A	B	A	A	A	A		A	A	A	A	A
Lactic acid, >50%, 70°F	X	B	B	B	B	B	B	A	A	A	A		A	A	A	A	A
Lactic acid, <5%, to boiling	X	X	X	B	X	B	B	A	A	104°C	A		X	B	A	A	50°C
Lime slurries, 70°F	B	B	B	B	A	B	B	B	B				A	A	A	A	
Magnesium chloride, 70°F	X	X	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A
Magnesium chloride, <5%, to boiling	X	X	X	B	X	B	A	A	A	104°C	A		A	A	A	A	140°C
Magnesium chloride, >5%, to boiling	X	X	X	X	X	X	B	B	B	104°C	A		A	A	A	A	140°C

Corrosive	Steel	Brz	316	A-20	CD4MCu	ALLOY 2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF
Magnesium hydroxide, 70°F	B	A	B	B	A	B	B	A		A	A	A	A	A	A	A	A
Magnesium sulphate	X	X	B	A	B	A	X	B	B	A	A		A	A	A	A	35°C
Maleic acid	X	X	B	B	B	B	B	A		A	A		B	A	A	A	20°C
Mercaptans	A	X	A	A	A	A				A	A			X	B	A	A
Mercuric chloride, <2%, 70°F	X	X	X	X	X	X	B	A	A	A	A		A	A	A	A	A
Mercurous nitrate, 70°F	X	X	B	B	B	B	C			A	A			A	A	A	A
Methyl alcohol, 70°F	A	A	A	A	A	A	A	A	A	A	A		A	X	A	A	A
Naphthalene sulphonic acid, 70°F	X	X	B	B	B	B	B			A	A					A	A
Napthalenic acid	X	X	B	B	B	B	B			A	A		X	A	A	A	A
Nickel chloride, 70°F	X	X	X	B	X	B		B	B	A	A	A	A	A	A	A	A
Nickel sulphate	X	X	B	B	B	B	B		A	A	A		A	A	A	A	A
Nitric acid	X	X	B	B	B	B		B	B				X	X	B	A	70%, 50°C
Nitrobenzene, 70°F	A	X	A	A	A	A	B	A		A	A	X	A	B	A	A	A
Nitroethane, 70°F	A	A	A	A	A	A	A	A	A	A	A	X	B	X	X	A	A
Nitropropane, 70°F	A	A	A	A	A	A	A	A	A	A	A	X	X	X	X	A	B
Nitrous acid, 70°F	X	X	X	X	X	X				A	A	A		X	X	A	
Nitrous oxide, 70°F	X	X	X	X	X	X	X			A	A			B	B	A	X
Oleic acid	X	X	B	B	B	B	X	X	X	A	A	X	B	B	B	A	20°C
Oleum acid, 70°F	B	X	B	B	B	B	B	B		A	A	X	X	B	A	A	X
Oxalic acid	X	X	X	B	X	B	B	X	A	A	A	X	A	A	A	A	50°C
Palmitic acid	B	B	B	A	B	A				A	A		B	A	A	A	20°C
Phenol (see carbolic acid)										A	A		B	B	A	A	50°C
Phosgene, 70°F	X	X	B	B	B	B	B			A	A			X	X	A	A
Phosphoric acid, <10%, 70°F	X	X	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Phosphoric acid, >10% to 70%, 70°F	X	X	A	A	A	A	X	B	B	A	A	X	A	A	A	A	A
Phosphoric acid, <20%, 175°F	X	X	B	B	B	B	A	X	B	A	A	X	A	A	A	A	A
Phosphoric acid, >20%, 175°F, <85%	X	X	X	B	X	B	X	X	X	A	A	X	A	A	A	A	A
Phosphoric acid, >10%, boil, <85%	X	X	X	X	X	X	X	X	X				A	A	A	A	A
Phthalic acid, 70°F	X	B	B	A	B	A	B	A	A	A	A			B	B	A	A
Phthalic anhydride, 70°F	B	X	A	A	A	A	A			A	A			B	B	A	
Picric acid, 70°F	X	X	X	B	X	B	B			A	A		B	A	A	A	A
Potassium carbonate	B	B	A	A	A	A	B	A	A	A	A	A		A	A	A	40°C
Potassium chlorate	B	X	A	A	A	A	B	A	A	A	A	A		A	A	A	95°C
Potassium chloride, 70°F	X	X	B	A	B	A	B	A	A	A	A	A	A	A	A	A	A
Potassium cyanide, 70°F	B	X	B	B	B	B	B			A	A	A	A	A	A	A	A
Potassium dichromate	B	B	A	A	A	A	B	A	A	A	A		A	A	A	A	40°C
Potassium ferricyanide	X	B	B	B	B	B	B	A	A	A	A			B	B	A	40°C
Potassium ferrocyanide, 70°F	X	B	B	B	B	B	B		B	A	A	A		B	B	A	A
Potassium hydroxide, 70°F	X	X	B	A	B	A	X	B	A	A	A	A	A	X	B	A	X
Potassium hypochlorite	X	X	X	B	X	B	B	A		A	A			X	X	A	95°C
Potassium iodide, 70°F	X	B	B	B	B	B	B	A	A	A	A			A	A	A	A
Potassium permanganate	B	B	B	B	B	B	B			A	A	A		B	B	A	20°C
Potassium phosphate	X	X	B	B	B	B		B	B	A	A	A		A	A	A	
Seawater, 70°F	X	B	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A
Sodium bisulphate, 70°F	X	X	X	B	X	B	B	B	A	A	A	A		A	A	A	A
Sodium bromide, 70°F	B	X	B	B	B	B	B			A	A	A		A	A	A	A
Sodium carbonate	B	B	B	A	B	A	B	A	A	A	A	A		A	A	A	40°C
Sodium chloride, 70°F	X	B	B	B	B	B	B	A	A	A	A	A	A	A	A	A	A
Sodium cyanide	B	X	B	B	B	B		B		A	A		A	A	A	A	35°C
Sodium dichromate	B	X	B	B	B	B		B		100°C	A			A	A	A	95°C
Sodium ethylate	B	A	A	A	A	A				A	A					A	A
Sodium fluoride	X	X	B	B	B	B	X	B	B	A	A			A	A	A	40°C
Sodium hydroxide, 70°F	B	B	B	A	B	A	A	A	A	A	A	A	A	B	A	A	X
Sodium hypochlorite	X	X	X	X	X	X	B	A	B	A	A	X	B	B	A	A	40%, 95°C
Sodium lactate, 70°F	B	X	X	X	X	X	X			A	A	A				A	A

Corrosive	Steel	Brz	316	A-20	CD4MCu	ALLOY 2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF
Stannic chloride, <5%, 70°F	X	X	X	X	X	X	B	A	A	A	A	A	A	A	A	A	A
Stannic chloride, >5%, 70°F	X	X	X	X	X	X	X	B	B	A	A		A	A	A	A	A
Sulphite liquors, to 175°F	X	X	B	B	B	B	B	A					B	A	A	A	A
Sulphur (molten)	B	X	A	A	A	A	A	A					A	A	A	A	120°C
Sulphur dioxide (spray), 70°F	X	X	B	B	B	B	B	X		A	A		A	A	A	A	A
Sulphuric acid, <2%, 70°F	X	X	B	A	B	A	A	B	A	A	A	A	A	A	A	A	A
Sulphuric acid, 2% to 40%, 70°F	X	X	X	B	X	B	A	X	A	A	A	A	B	A	A	A	A
Sulphuric acid, 40%, <90%, 70°F	X	X	X	B	X	B	A	X	X	A	A	X	B	B	A	A	A
Sulphuric acid, 93% to 98%, 70°F	B	X	B	B	B	B	A	X	X	A	A	X	X	B	A	A	A
Sulphuric acid, <10%, 175°F	X	X	X	B	X	B	A	X	B	A	A	A	X	A	A	A	A
Sulphuric acid, 10% to 60% & >80%, 175°F	X	X	X	B	X	B	X	X	X	A	A	X	B	A	A	A	A
Sulphuric acid, 60% to 80%, 175°F	X	X	X	X	X	X	B	X	X	A	A	X	X	B	A	A	A
Sulphuric acid, <3/4%, boiling	X	X	X	B	X	B	A	X	B				X	B	A	A	120°C
Sulphuric acid, 3/4% to 40%, boiling	X	X	X	X	X	X	X	X	B				X	B	A	A	120°C
Sulphuric acid, 40% to 65% & >85%, boiling	X	X	X	X	X	X	X	X	X	X			X	X	B	A	B
Sulphuric acid, 65% to 85%, boiling	X	X	X	X	X	X	X	X	X				X	X	B	A	95°C
Sulphurous acid, 70°F	X	X	X	B	X	B	B	A	B	A	A	A	X	X	B	A	A
Titanium tetrachloride, 70°F	X		X	B	X	B	X			A	A		X	B	B	B	A
Tirchlorethylene, to boiling	B	X	B	B	B	B	B	A	A				X	B	A	A	A
Urea, 70°F	X	X	B	B	B	B	X	B	B	A	A			B	A	A	A
Vinyl acetate	B	B	B	B	B	B	B			A	A			B	B	A	120°C
Vinyl chloride	B	X	B	B	B	B	B	A		A	A		X	A	A	A	95°C
Water, to boiling	B	A	A	A	A	A	A	A	A				A	A	A	A	A
Zinc chloride	X	X	B	A	B	A		A	A	A	A	A	A	A	A	A	140°C
Zinc cyanide, 70°F	X	B	B	B	B	B	B	B	B	A	A					A	A
Zinc sulphate	X	X	A	A	A	A	X	A		A	A	A	A	A	A	A	140°C

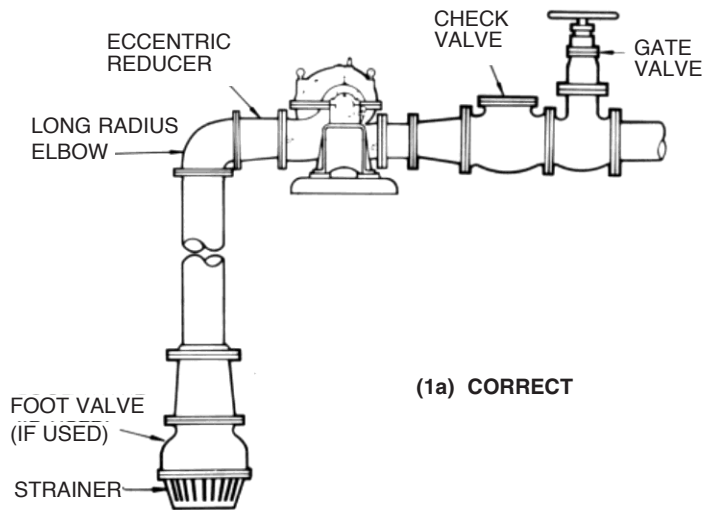
Elastomer Selection Guide

Please use the following chart as a general guide only. Refer to detailed selection tables or the factory for specific elastomer recommendations.

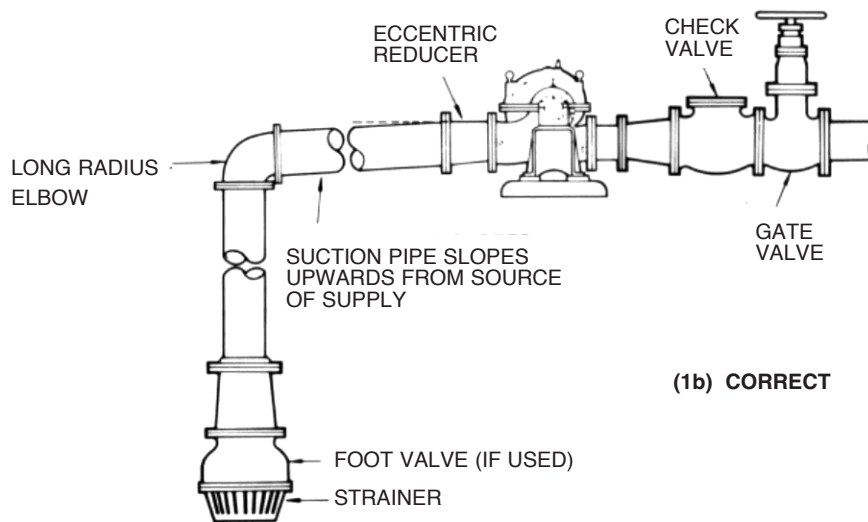
Elastomer	Shore (A) Hardness	Max Temp Limit	pH Range	Abrasion	Resistance to Moderate Chemicals	Oils Hydrocarbons
Natural Rubber	40	154 F	5 - 12	E	G (1)	P
Polyurethane	81	149 F	3 - 11	E (2)	G (1)	E
Neoprene	60	212 F	3 - 12	G	G (1)	G
Nitrile	60	220 F	4 - 12	G	G (1)	E
Hypalon	55	230 F	1 - 14	G	E	G
Chlorobutyl	50	300 F	3 - 12	G	E	P

- (1) Poor for oxidizing chemicals and strong acids.
(2) Fine particles only (200 mesh or less).

E = Excellent
G = Good
P = Poor

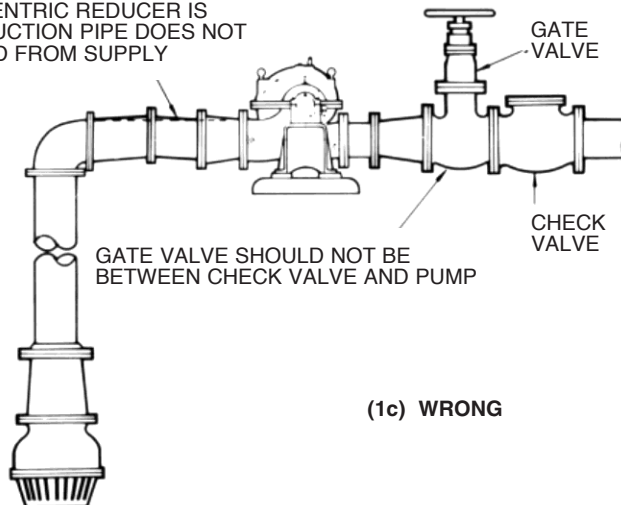


(1a) CORRECT



(1b) CORRECT

AIR POCKET BECAUSE ECCENTRIC REDUCER IS NOT USED AND BECAUSE SUCTION PIPE DOES NOT SLOPE GRADUALLY UPWARD FROM SUPPLY



(1c) WRONG

Fig. 1 Air Pockets in Suction Piping

TECH-B-3 Piping Design

The design of a piping system can have an important effect on the successful operation of a centrifugal pump. Such items as sump design, suction piping design, suction and discharge pipe size, and pipe supports must all be carefully considered.

Selection of the discharge pipe size is primarily a matter of economics. The cost of the various pipe sizes must be compared to the pump size and power cost required to overcome the resulting friction head.

The suction piping size and design is far more important. Many centrifugal pump troubles are caused by poor suction conditions.

The Suction pipe should never be smaller than the suction connection of the pump, and in most cases should be at least one size larger. Suction pipes should be as short and as straight as possible. Suction pipe velocities should be in the 5 to 8 feet per second range unless suction conditions are unusually good.

Higher velocities will increase the friction loss and can result in troublesome air or vapor separation. This is further complicated when elbows or tees are located adjacent to the pump suction nozzle, in that uneven flow patterns or vapor separation keeps the

liquid from evenly filling the impeller. This upsets hydraulic balance leading to noise vibration, possible cavitation, and excessive shaft deflection. Cavitation erosion damage, shaft breakage or premature bearing failure may result.

On pump installations involving suction lift, air pockets in the suction line can be a source of trouble. The Suction pipe should be exactly horizontal, or with a uniform slope upward from the sump to the pump as shown in Fig. 1. There should be no high spots where air can collect and cause the pump to lose its prime. Eccentric rather than concentric reducers should always be used.

If an elbow is required at the suction of a double suction pump, it should be in a vertical position if at all possible. Where it is necessary for some reason to use a horizontal elbow, it should be a long radius elbow and there should be a minimum of five diameters of straight pipe between the elbow and the pump as shown in Fig. 2. Fig. 3 shows the effect of an elbow directly on the suction. The liquid will flow toward the outside of the elbow and result in an uneven flow distribution into the two inlets of the double suction impeller. Noise and excessive axial thrust will result.

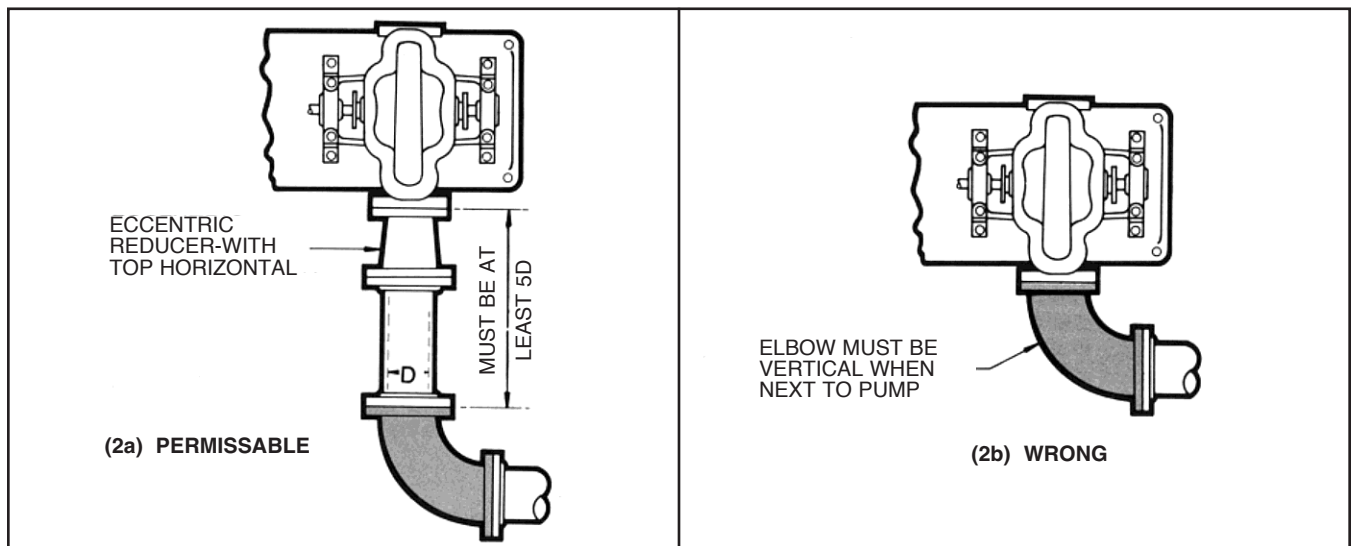


Fig. 2 Elbows At Pump Suction

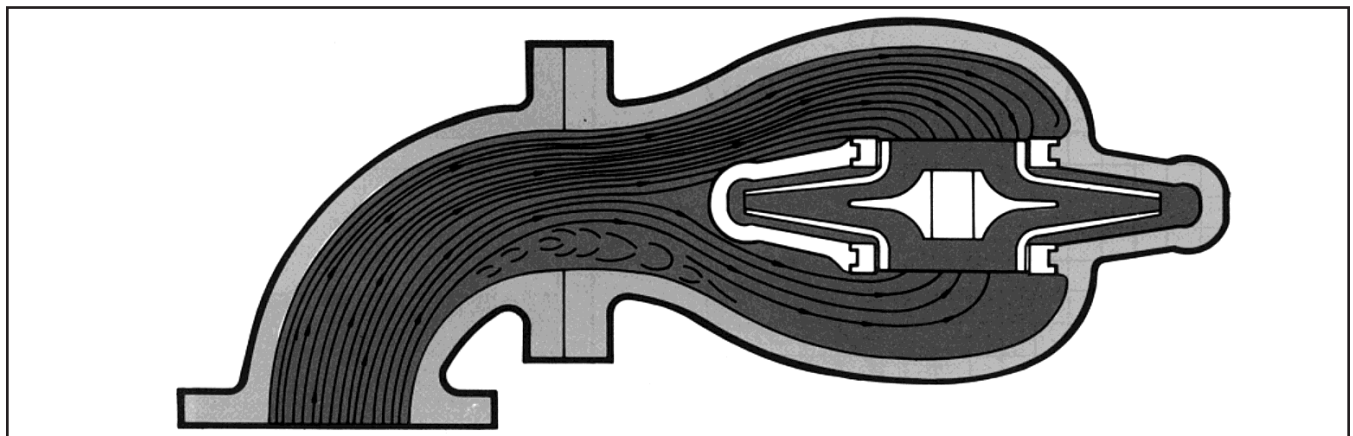


Fig. 3 Effect of Elbow Directly on Suction

There are several important considerations in the design of a suction supply tank or sump. It is imperative that the amount of turbulence and entrained air be kept to a minimum. Entrained air may cause reduced capacity and efficiency as well as vibration, noise, shaft breakage, loss of prime, and/or accelerated corrosion.

The free discharge of liquid above the surface of the supply tank at or near the pump suction can cause entrained air to enter the pump. All lines should be submerged in the tank, and baffles should be used in extreme cases as shown in Fig. 4.

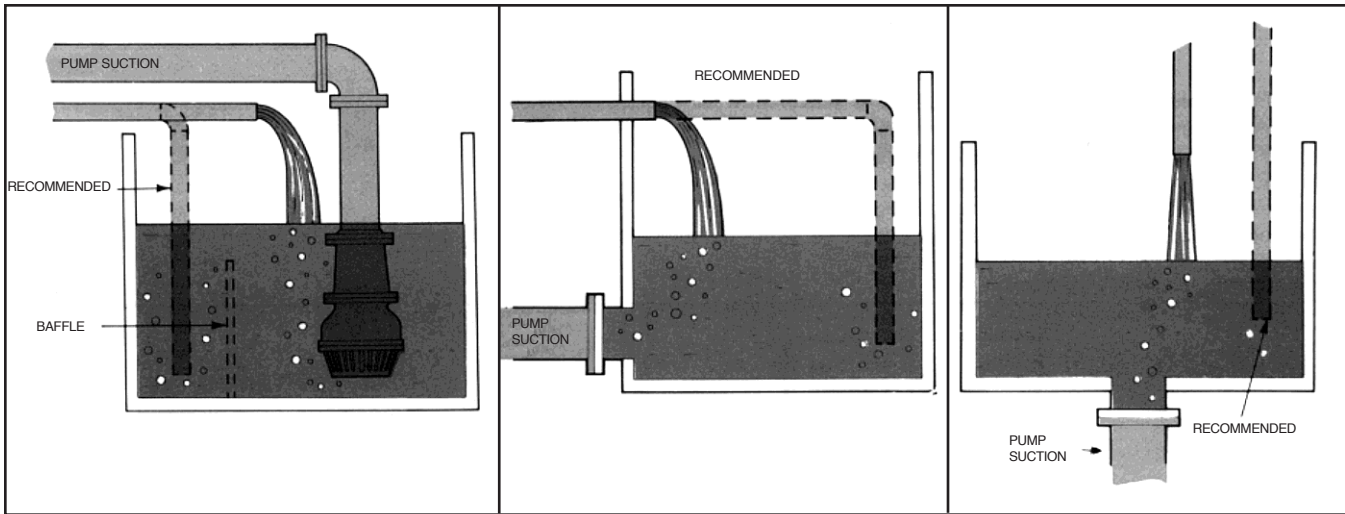


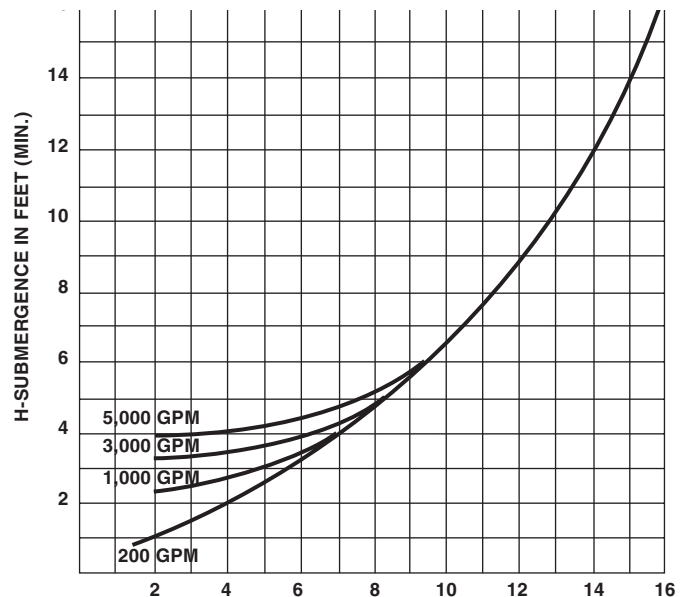
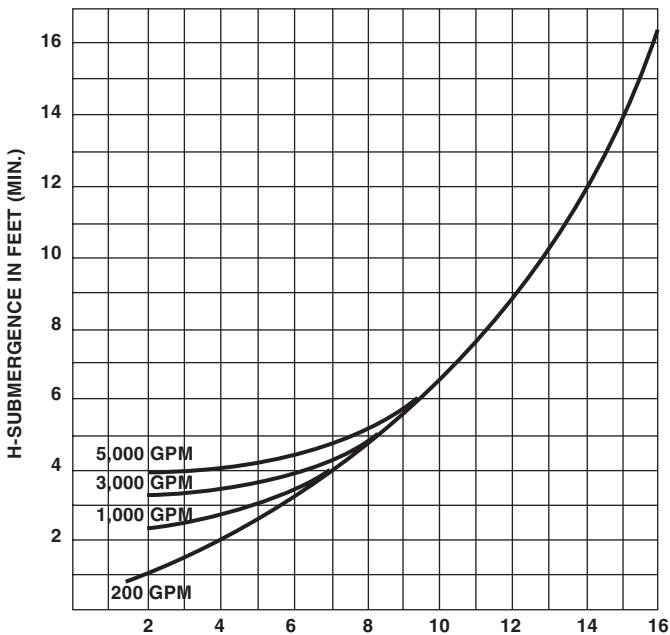
Fig. 4 Keeping Air Out of Pump

Improper submergence of the pump suction line can cause a vortex, which is a swirling funnel of air from the surface directly into the pump suction pipe. In addition to submergence, the location of the pipe in the sump and the actual dimensions of the sump are also important in preventing vortexing and/or excess turbulence.

For horizontal pumps, Fig. 5 can be used as a guide for minimum submergence and sump dimensions for flows up to approximately 5000 gpm. Baffles can be used to help prevent vortexing in cases where it is impractical or impossible to maintain the required submergence. Fig. 6 shows three such baffling arrangements.

On horizontal pumps, a bell should be used on the end of the suction pipe to limit the entrance velocity to 5.5 feet per second. Also, a reducer at the pump suction flange to smoothly accelerate and stabilize the flow into the pump is desirable.

The submergence of the suction pipe must also be carefully considered. The amount of submergence required depends upon the size and capacity of the individual pumps as well as on the sump design. Past experience is the best guide for determining the submergence. The pump manufacturer should be consulted for recommendations in the absence of other reliable data.



$$\text{VELOCITY IN FEET PER SEC.} = \frac{\text{QUAN. (G.P.M.)} \times .321}{\text{AREA (inches)}^2} \text{ OR } \frac{\text{G.P.M.} \times .4085}{D^2}$$

$$\text{VELOCITY IN FEET PER SEC.} = \frac{\text{QUAN. (G.P.M.)} \times .321}{\text{AREA (inches)}^2} \text{ OR } \frac{\text{G.P.M.} \times .4085}{D^2}$$

Fig. 5 Minimum Suction Pipe Submergence and Sump Dimensions

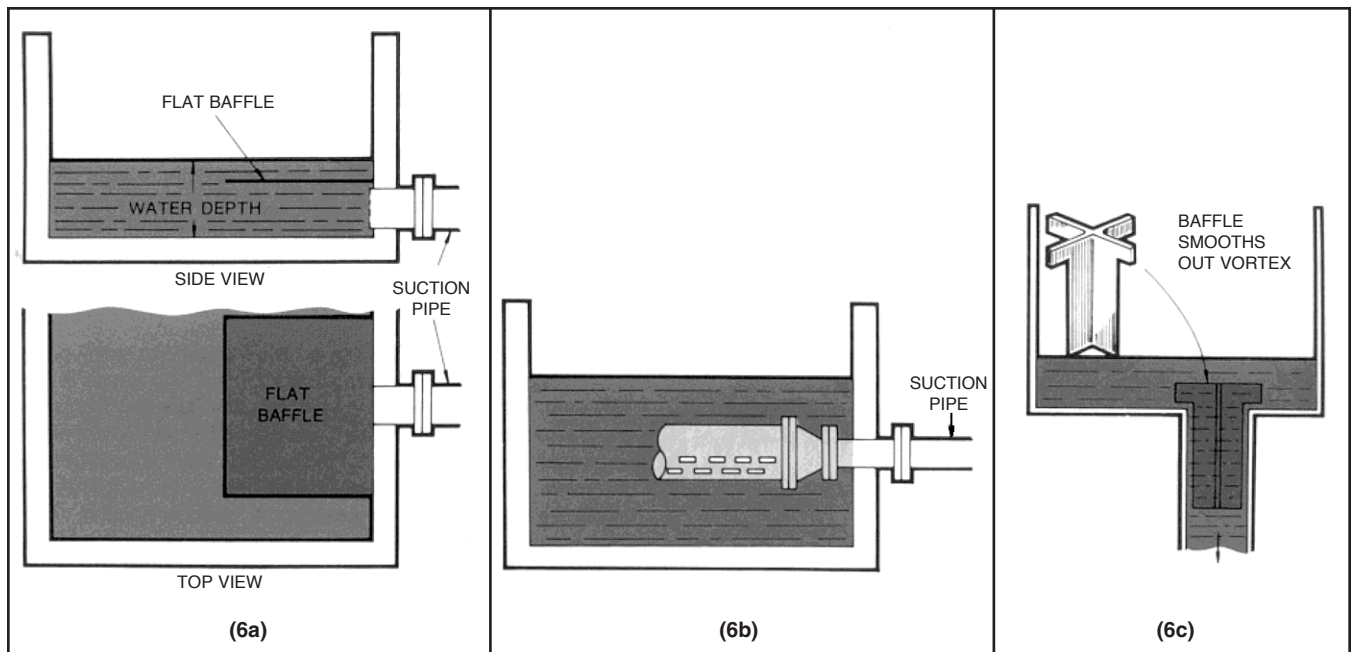


Fig. 6 Baffle Arrangements for Vortex Prevention

For larger units (over 5000 GPM) taking their suction supply for an intake sump (especially vertically submerged pumps), requires special attention.

The following section (intake System Design) addresses these larger pumps.

INTAKE SYSTEM DESIGN

The function of the intake structure, whether it be an open channel, a fully wetted tunnel, a sump, or a tank, is to supply an evenly distributed flow to the pump suction. An uneven distribution of flow, characterized by strong local currents, can result in formation of surface or submerged vortices and with certain low values of submergence, may introduce air into the pump, causing a reduction of capacity, an increase in vibration and additional noise. Uneven flow distribution can also increase or decrease the power consumption with a change in total developed head.

The ideal approach is a straight channel coming directly to the pump or suction pipe. Turns and obstructions are detrimental, since they may cause eddy currents and tend to initiate deep-cored vortices.

The amount of submergence available is only one factor affecting vortex-free operation. It is possible to have adequate submergence and still have submerged vortices that may have an adverse effect on pump operation. Successful, vortex-free operation will depend greatly on the approach upstream of the sump.

Complete analysis of intake structures can only be accurately accomplished by scale model tests. Model testing is especially recommended for larger pumping units.

GENERAL DATA INFORMATION

Subject to the qualifications of the foregoing statements, Figures 7 through 10 have been constructed for single and multiple intake arrangements to provide guidelines for basic sump dimensions.

Since these values are composite averages for many pump types and cover the entire range of specific speeds, they are not absolute values but typical values subject to variations.

All of the dimensions in Figures 7 through 10 are based on the rated capacity of the pump. If operation at an increased capacity is to be undertaken for extended periods of time, the maximum capacity should be used for obtaining sump dimensions.

If the position of the back wall is determined structurally, dimension B in Figures 7 to 10 may become excessive and a false back wall should be installed.

Dimension S in Figures 7 and 9 is a minimum value based on the normal low water level at the pump or suction pipe bell, taking into consideration friction losses through the inlet screen and approach channel. Note that this dimension represents submergence at the intake, or the physical height of the water level above the intake relating to the prevention of eddy formations and vortexing.

The channel floor should be level for at least a distance Y (see Figures 7 through 10) upstream before any slope begins. The screen or gate widths should not be substantially less than W, and heights should not be less than the maximum anticipated water level to avoid overflow. Depending on the approach conditions before the sump, it may be necessary to construct straightening vanes in the approach channel, increase dimension A and/or conduct an intake model test to work out some other combination of these factors.

Dimension W is the width of an individual pump cell or the center-to-center distance of two pumps if no dividing wall is used.

On multiple intake installations, the recommended dimensions in Figures 7 and 8 apply as noted above, and the following additional factors should be considered.

As shown in Fig. 10 (A), low velocity and straight in-line flow to all units simultaneously is a primary recommendation. Velocities in the sump should be approximately one foot per second, but velocities of two feet per second may prove satisfactory. This is particularly true when the design is based on a model study. Not recommended would be an abrupt change in the size of the inlet pipe to the sump or the inlet from one side introducing eddying.

In many cases, as shown in Fig. 10 (B), pumps operate satisfactorily without separating walls below 5,000 GPM. If walls must be used for structural purposes or some pumps operate intermittently, then the walls should extend from the rear wall approximately five times the D dimension given in Fig. 7.

If walls are used, increase dimension W by the thickness of the wall for correct centerline spacing and use round or ogive ends of walls. Not recommended is the placement of a number of pumps or suction pipes around the sides of a sump with or without dividing walls.

Abrupt changes in size, as shown in Fig. 10 (C), from inlet pipe or channel to the sump are not desirable. Connection of a pipe to a sump is best accomplished using a gradually increasing taper section. The angle should be as small as possible, preferably not more than 10 degrees. With this arrangement, sump velocities less than one foot per second are desirable.

Specifically not recommended is a pipe directly connected to a sump with suction intakes close to the sump inlet, since this results in an abrupt change in the flow direction. Centering pumps or suction

pipes in the sump leaves large vortex areas behind the intake which will cause operational trouble.

If the sump velocity, as shown in Fig. 10 (D), can be kept low (approximately one foot per second), an abrupt change from inlet pipe to sump can be accommodated if the sump length equals or exceeds the values shown. As ratio Z/P increases, the inlet velocity at P may be increased up to an allowed maximum of eight feet per second at Z/P 10. Intakes "in line" are not recommended unless a trench-type of intake is provided (per ANSI/HI 9.8), or the ratio of sump to intake size is quite large and intakes are separated by a substantial margin longitudinally. A sump can generally be constructed at less cost by using a recommended design.

As shown in Fig. 10 (E), it is sometimes desirable to install pumps in tunnels or pipe lines. A drop pipe or false well to house the unit with a vaned inlet elbow facing upstream is satisfactory in flows up to eight feet per second. Without inlet elbow, the suction bell should be positioned at least two pipe (vertical) diameters above the top of the tunnel. The unit should not be suspended in the tunnel flow, unless the tunnel velocity is less than two feet per second. There must be no air along the top of the tunnel, and the minimum submergence must be provided.

In general: Keep inlet velocity to the sump below two feet per second. Keep velocity in sump below 1.5 foot per second. Avoid changing direction of flow from inlet to pump or suction pipe, or change direction gradually and smoothly, guiding flow.

$$D = (.0744Q)^{0.5} \text{ Recommended}$$

$$W = 2D$$

$$Y \geq 5D$$

$$A \geq 5D$$

$$C = .3D \text{ to } .5D$$

$$B = .75D$$

$$S = D + 0.574Q/D^{1.5}$$

Where:

S - inches

Q - Flow (GPM)

D - inches

Fig. 7 Sump Dimensions

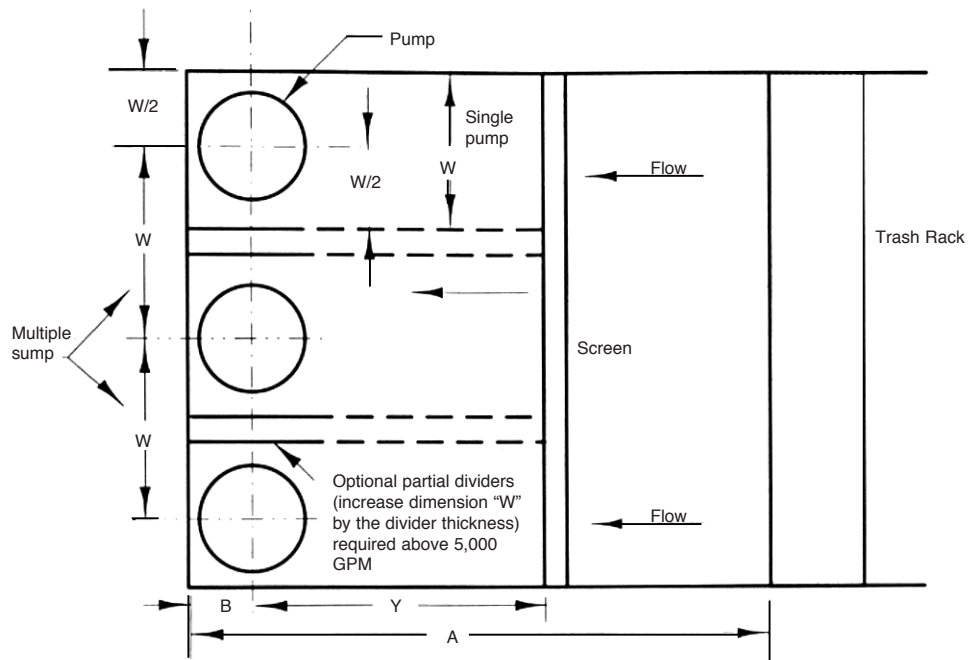


Fig. 8 Sump dimensions, plan view, wet pit type pumps

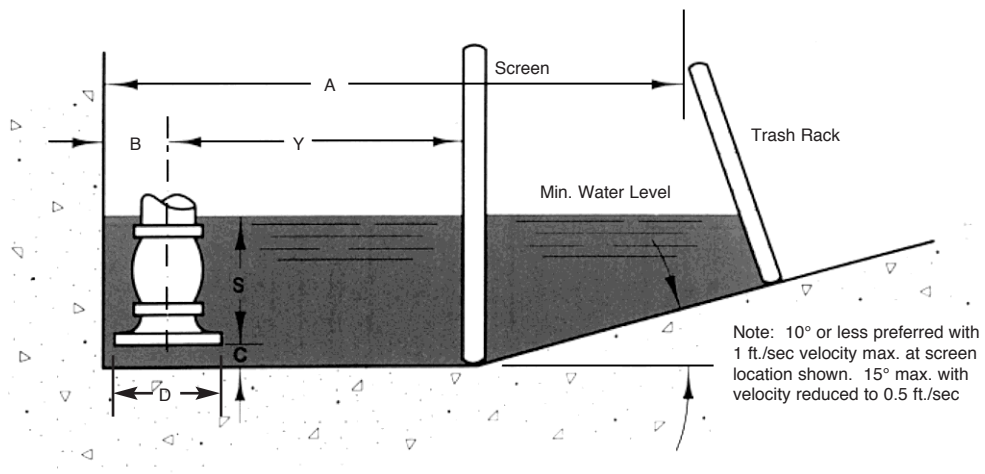


Fig. 9 Sump dimensions, elevation view, wet pit type pumps

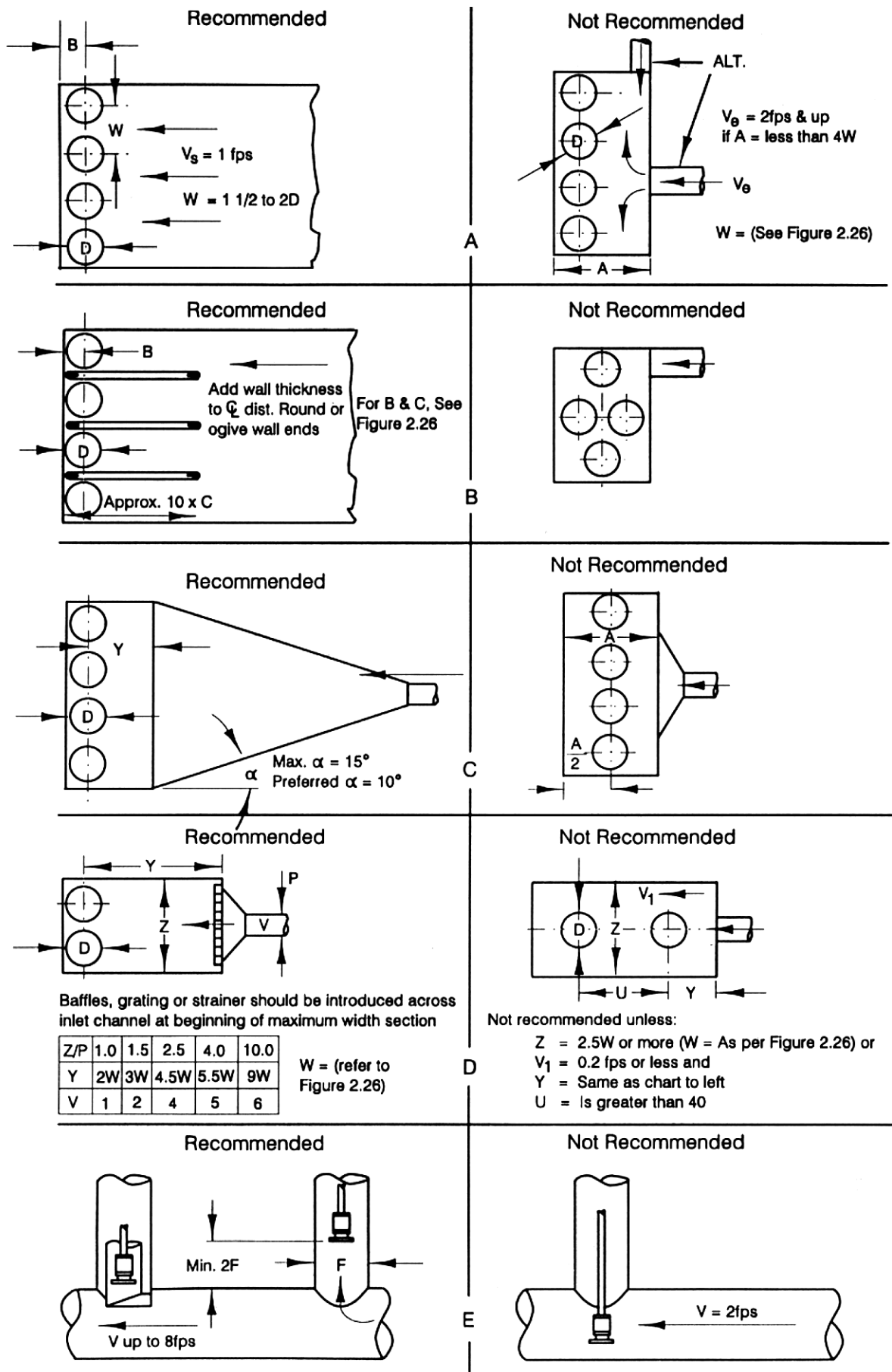


Fig. 10 Multiple pump installations

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TECH-B-4A Sealing

The proper selection of a seal is critical to the success of every pump application. For maximum pump reliability, choices must be made between the type of seal and the seal environment. In addition, a sealless pump is an alternative which would eliminate the need for a dynamic type seal entirely.

Sealing Basics

There are two basic kinds of seals: static and dynamic. Static seals are employed where no movement occurs at the juncture to be sealed. Gaskets and O-rings are typical static seals.

Dynamic seals are used where surfaces move relative to one another. Dynamic seals are used, for example, where a rotating shaft transmits power through the wall of a tank (Fig. 1), through the casing of a pump (Fig. 2), or through the housing of other rotating equipment such as a filter or screen.

A common application of sealing devices is to seal the rotating shaft of a centrifugal pump. To best understand how such a seal functions, a quick review of pump fundamentals is in order.

In a centrifugal pump, the liquid enters the suction of the pump at the center (eye) of the rotating impeller (Figures 3 and 4).

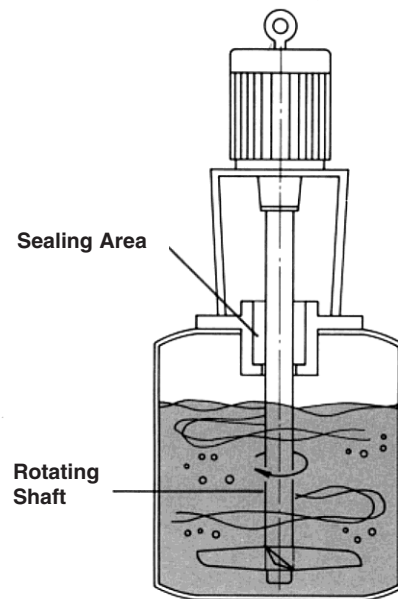
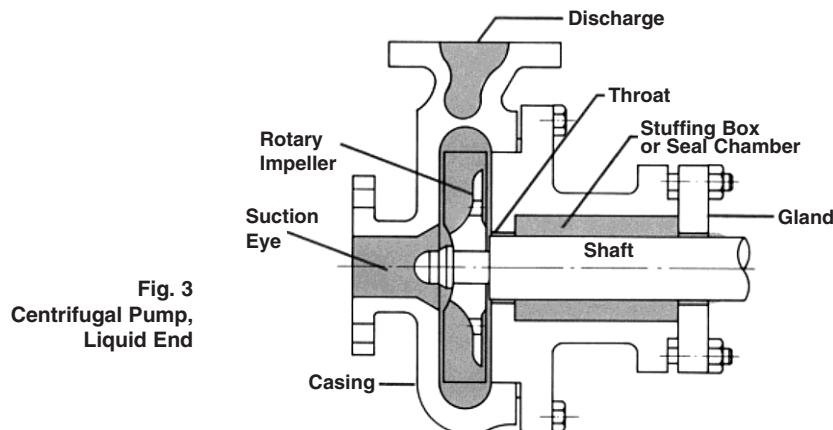
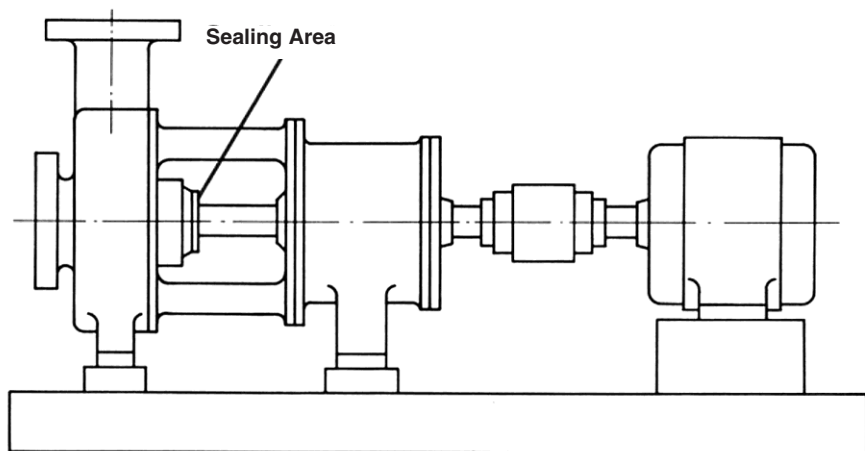


Fig. 1 Cross Section of Tank and Mixer

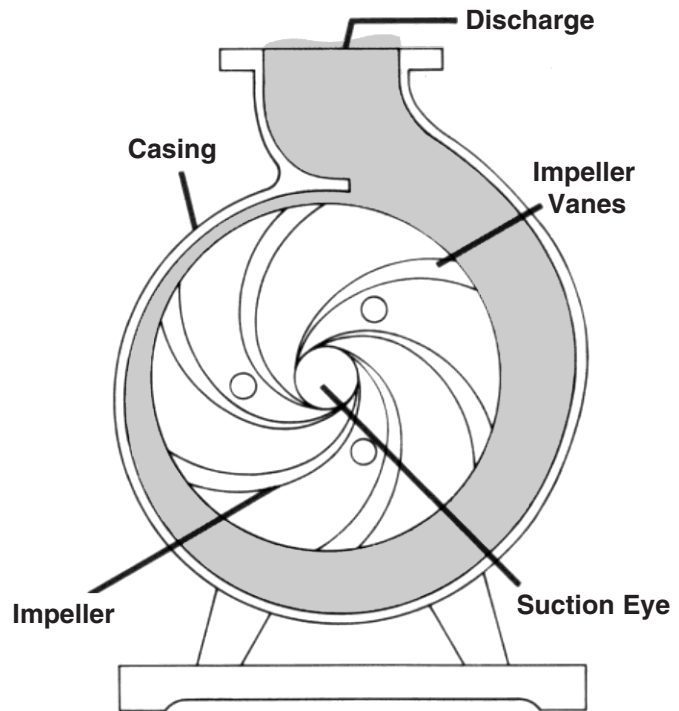


As the impeller vanes rotate, they transmit motion to the incoming product, which then leaves the impeller, collects in the pump casing, and leaves the pump under pressure through the pump discharge.

Discharge pressure will force some product down behind the impeller to the drive shaft, where it attempts to escape along the rotating drive shaft. Pump manufacturers use various design techniques to reduce the pressure of the product trying to escape. Such techniques include: 1) the addition of balance holes through the impeller to permit most of the pressure to escape into the suction side of the impeller, or 2) the addition of back pump-out vanes on the back side of the impeller.

However, as there is no way to eliminate this pressure completely, sealing devices are necessary to limit the escape of the product to the atmosphere. Such sealing devices are typically either compression packing or end-face mechanical seals.

Fig. 4
Fluid Flow in a
Centrifugal Pump



Stuffing Box Packing

A typical packed stuffing box arrangement is shown in Fig. 5. It consists of: A) Five rings of packing, B) A lantern ring used for the injection of a lubricating and/or flushing liquid, and C) A gland to hold the packing and maintain the desired compression for a proper seal.

The function of packing is to control leakage and not to eliminate it completely. The packing must be lubricated, and a flow from 40 to 60 drops per minute out of the stuffing box must be maintained for proper lubrication.

The method of lubricating the packing depends on the nature of the liquid being pumped as well as on the pressure in the stuffing box. When the pump stuffing box pressure is above atmospheric pressure and the liquid is clean and nonabrasive, the pumped liquid itself will lubricate the packing (Fig. 6). When the stuffing box pressure is below atmospheric pressure, a lantern ring is employed and lubrication is injected into the stuffing box (Fig. 7). A bypass line from the pump discharge to the lantern ring connection is normally used providing the pumped liquid is clean.

When pumping slurries or abrasive liquids, it is necessary to inject a clean lubricating liquid from an external source into the lantern ring

(Fig. 8). A flow of from .2 to .5 gpm is desirable and a valve and flowmeter should be used for accurate control. The seal water pressure should be from 10 to 15 psi above the stuffing box pressure, and anything above this will only add to packing wear. The lantern ring is normally located in the center of the stuffing box. However, for extremely thick slurries like paper stock, it is recommended that the lantern ring be located at the stuffing box throat to prevent stock from contaminating the packing.

The gland shown in Figures 5 through 8 is a quench type gland. Water, oil, or other fluids can be injected into the gland to remove heat from the shaft, thus limiting heat transfer to the bearing frame. This permits the operating temperature of the pump to be higher than the limits of the bearing and lubricant design. The same quench gland can be used to prevent the escape of a toxic or volatile liquid into the air around the pump. This is called a smothering gland, with an external liquid simply flushing away the undesirable leakage to a sewer or waste receiver.

Today, however, stringent emission standards limit use of packing to non-hazardous water based liquids. This, plus a desire to reduce maintenance costs, has increased preference for mechanical seals.

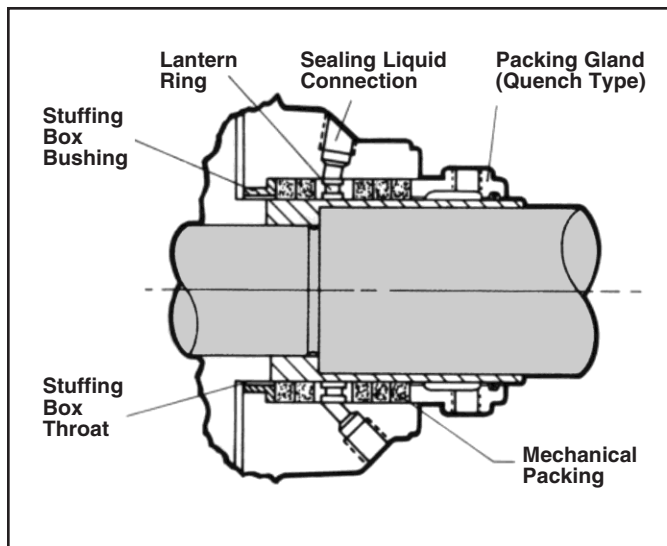


Fig. 5 Typical Stuffing Box Arrangement
(Description of Parts)

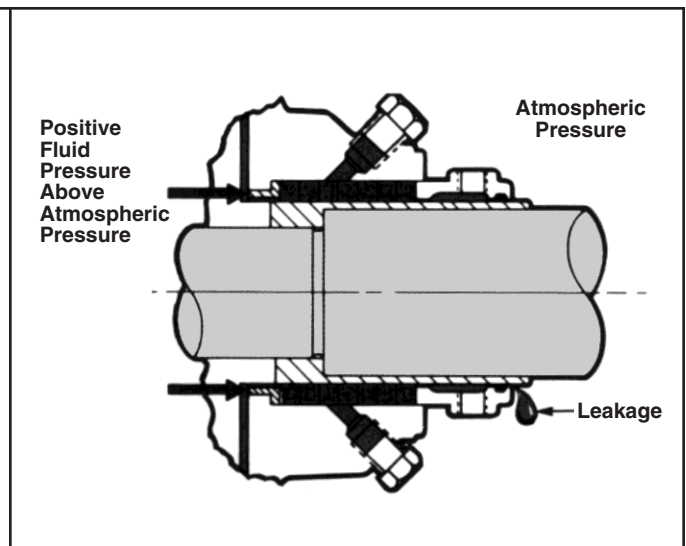


Fig. 6 Typical Stuffing Box Arrangement When Stuffing Box
Pressure is Above Atmospheric Pressure

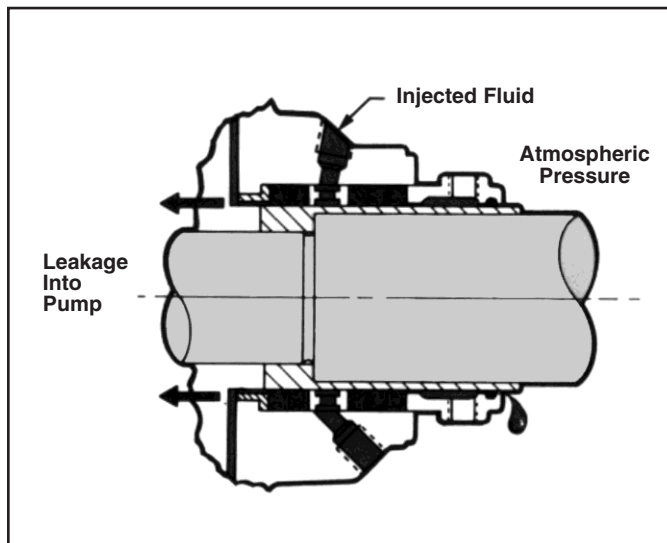


Fig. 7 Typical Stuffing Box Arrangement When Stuffing Box
Pressure is Below Atmospheric Pressure

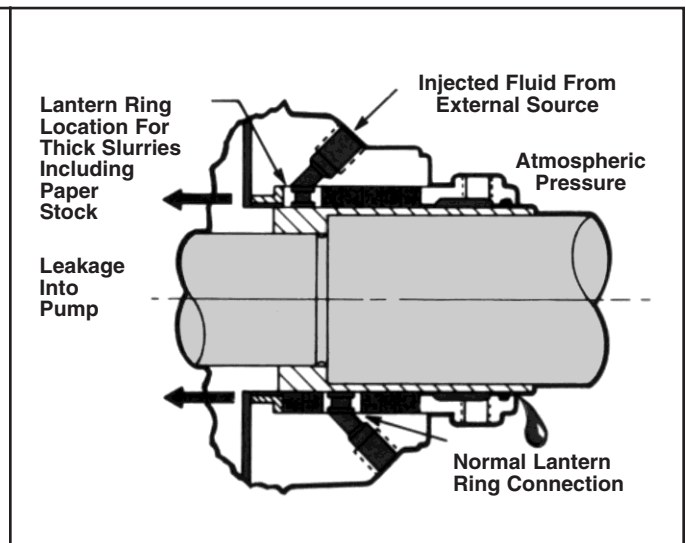


Fig. 8 Typical Stuffing Box Arrangement When
Pumping Slurries

Mechanical Seals

A mechanical seal is a sealing device which forms a running seal between rotating and stationary parts. They were developed to overcome the disadvantages of compression packing. Leakage can be reduced to a level meeting environmental standards of government regulating agencies and maintenance costs can be lower. Advantages of mechanical seals over conventional packing are as follows:

1. Zero or limited leakage of product (meet emission regulations.)
2. Reduced friction and power loss.
3. Elimination of shaft or sleeve wear.
4. Reduced maintenance costs.
5. Ability to seal higher pressures and more corrosive environments.
6. The wide variety of designs allows use of mechanical seals in almost all pump applications.

The Basic Mechanical Seal

All mechanical seals are constructed of three basic sets of parts as shown in Fig. 9:

1. A set of primary seal faces: one rotary and one stationary...shown in Fig. 9 as seal ring and insert.
2. A set of secondary seals known as shaft packings and insert mountings such as O-rings, wedges and V-rings.
3. Mechanical seal hardware including gland rings, collars, compression rings, pins, springs and bellows.

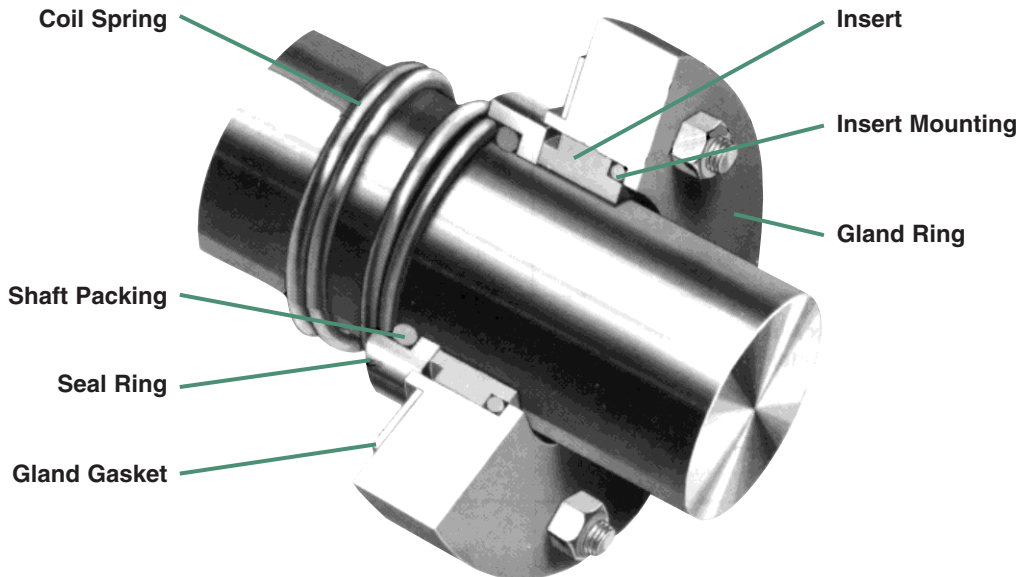


Fig. 9 A Simple Mechanical Seal

How A Mechanical Seal Works

The primary seal is achieved by two very flat, lapped faces which create a difficult leakage path perpendicular to the shaft. Rubbing contact between these two flat mating surfaces minimizes leakage. As in all seals, one face is held stationary in a housing and the other face is fixed to, and rotates with, the shaft. One of the faces is usually a non-galling material such as carbon-graphite. The other is usually a relatively hard material like silicon-carbide. Dissimilar materials are usually used for the stationary Insert and the rotating seal ring face in order to prevent adhesion of the two faces. The softer face usually has the smaller mating surface and is commonly called the wear nose.

There are four main sealing points within an end face mechanical seal (Fig. 10). The primary seal is at the seal face, Point A. The leakage path at Point B is blocked by either an O-ring, a V-ring or a wedge. Leakage paths at Points C and D are blocked by gaskets or O-rings.

The faces in a typical mechanical seal are lubricated with a boundary layer of gas or liquid between the faces. In designing seals for the desired leakage, seal life, and energy consumption, the designer must consider how the faces are to be lubricated and select from a number of modes of seal face lubrication.

To select the best seal design, it's necessary to know as much as possible about the operating conditions and the product to be sealed. Complete information about the product and environment will allow selection of the best seal for the application.

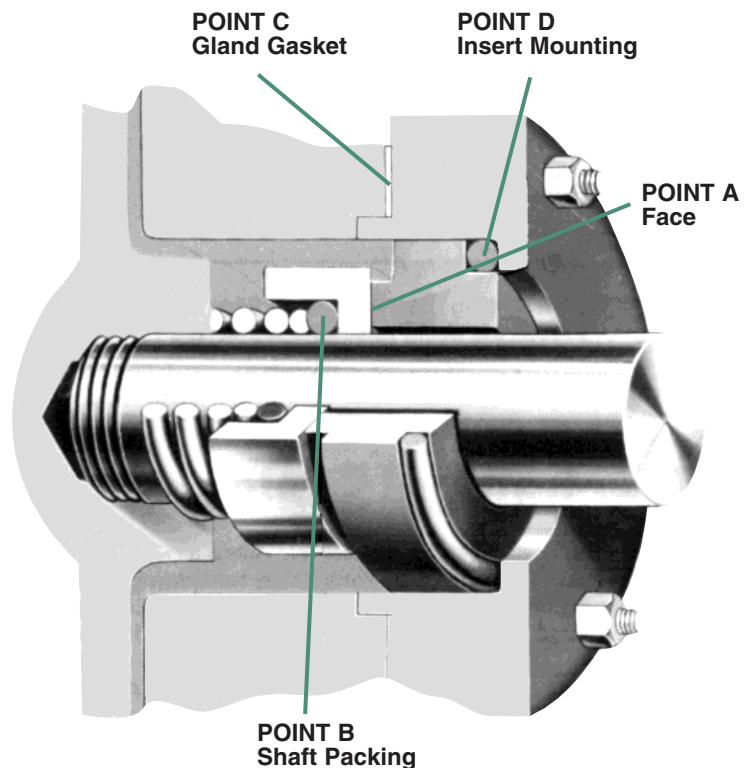


Fig. 10 Sealing Points for Mechanical Seal

Mechanical Seal Types

Mechanical seals can be classified into several types and arrangements:



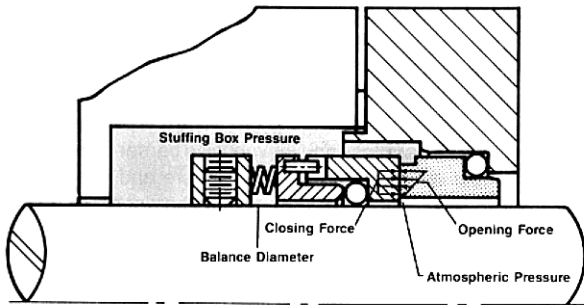
PUSHER:

Incorporate secondary seals that move axially along a shaft or sleeve to maintain contact at the seal faces. This feature compensates for seal face wear and wobble due to misalignment. The pusher seals advantage is that it's inexpensive and commercially available in a wide range of sizes and configurations. Its disadvantage is that it's prone to secondary seal hang-up and fretting of the shaft or sleeve. Examples are Dura RO and Crane Type 9T.



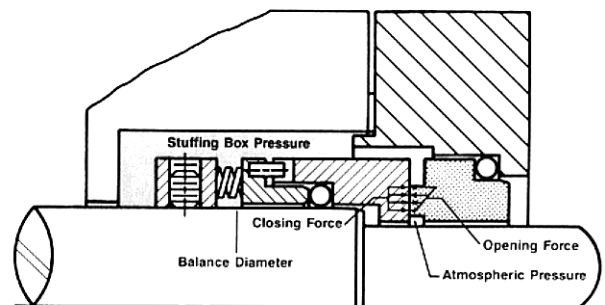
NON-PUSHER:

The non-pusher or bellows seal does not have to move along the shaft or sleeve to maintain seal face contact. The main advantages are its ability to handle high and low temperature applications, and does not require a secondary seal (not prone to secondary seal hang-up). A disadvantage of this style seal is that its thin bellows cross sections must be upgraded for use in corrosive environments. Examples are Dura CBR and Crane 215, and Sealol 680.



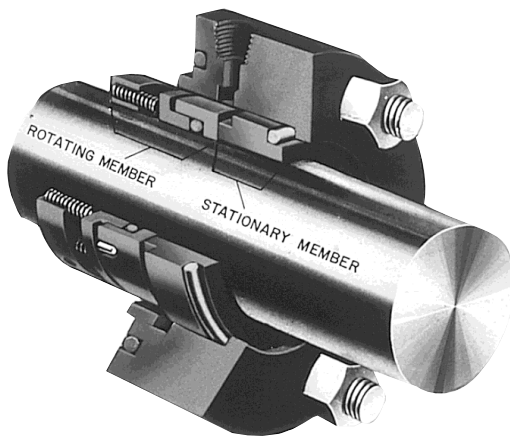
UNBALANCED:

They are inexpensive, leak less, and are more stable when subjected to vibration, misalignment, and cavitation. The disadvantage is their relative low pressure limit. If the closing force exerted on the seal faces exceeds the pressure limit, the lubricating film between the faces is squeezed out and the highly loaded dry running seal fails. Examples are the Dura RO and Crane 9T.



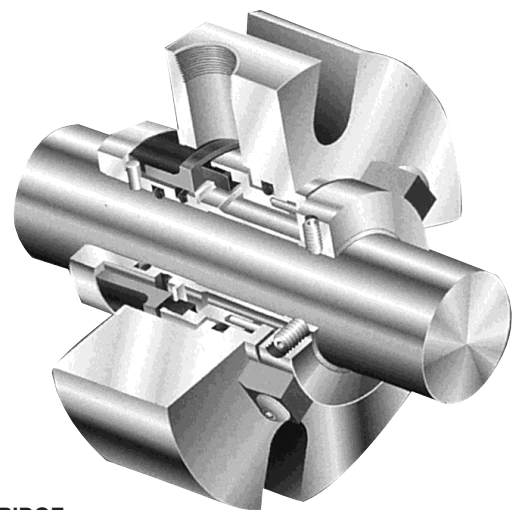
BALANCED:

Balancing a mechanical seal involves a simple design change which reduces the hydraulic forces acting to close the seal faces. Balanced seals have higher pressure limits, lower seal face loading, and generate less heat. This makes them well suited to handle liquids with poor lubricity and high vapor pressures such as light hydrocarbons. Examples are Dura CBR and PBR and Crane 98T and 215.



CONVENTIONAL:

Examples are the Dura RO and Crane Type 1 which require setting and alignment of the seal (single, double, tandem) on the shaft or sleeve of the pump. Although setting a mechanical seal is relatively simple, today's emphasis on reducing maintenance costs has increased preference for cartridge seals.



CARTRIDGE:

Examples are Dura P-50 and Crane 1100 which have the mechanical seal premounted on a sleeve including the gland and fit directly over the Model 3196 shaft or shaft sleeve (available single, double, tandem). The major benefit, of course is no requirement for the usual seal setting measurements for their installation. Cartridge seals lower maintenance costs and reduce seal setting errors.

Mechanical Seal Arrangements

SINGLE INSIDE:

This is the most common type of mechanical seal. These seals are easily modified to accommodate seal flush plans and can be balanced to withstand high seal environment pressures. Recommended for relatively clear non-corrosive and corrosive liquids with satisfactory lubricating properties where cost of operation does not exceed that of a double seal. Examples are Dura RO and CBR and Crane 9T and 215. Reference Conventional Seal.

SINGLE OUTSIDE:

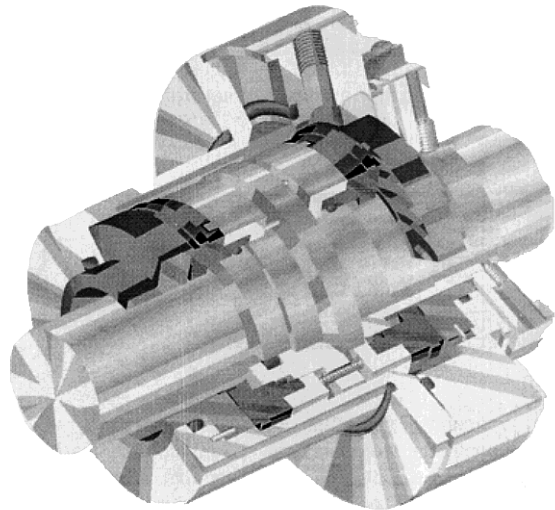
If an extremely corrosive liquid has good lubricating properties, an outside seal offers an economical alternative to the expensive metal required for an inside seal to resist corrosion. The disadvantage is that it is exposed outside of the pump which makes it vulnerable to damage from impact and hydraulic pressure works to open the seal faces so they have low pressure limits (balanced or unbalanced).



DOUBLE (DUAL PRESSURIZED):

This arrangement is recommended for liquids that are not compatible with a single mechanical seal (i.e. liquids that are toxic, hazardous [regulated by the EPA], have suspended abrasives, or corrosives which require costly materials). The advantages of the double seal are that it can have five times the life of a single seal in severe environments. Also, the metal inner seal parts are never exposed to the liquid product being pumped, so viscous, abrasive, or thermosetting liquids are easily sealed without a need for expensive metallurgy. In addition, recent testing has shown that double seal life is virtually unaffected by process upset conditions during pump operation. A significant advantage of using a double seal over a single seal.

The final decision between choosing a double or single seal comes down to the initial cost to purchase the seal, cost of operation of the seal, and environmental and user plant emission standards for leakage from seals. Examples are Dura double RO and X-200 and Crane double 811T.



DOUBLE GAS BARRIER (PRESSURIZED DUAL GAS):

Very similar to cartridge double seals...sealing involves an inert gas, like nitrogen, to act as a surface lubricant and coolant in place of a liquid barrier system or external flush required with conventional or cartridge double seals. This concept was developed because many barrier fluids commonly used with double seals can no longer be used due to new emission regulations. The gas barrier seal uses nitrogen or air as a harmless and inexpensive barrier fluid that helps prevent product emissions to the atmosphere and fully complies with emission regulations. The double gas barrier seal should be considered for use on toxic or hazardous liquids that are regulated or in situations where increased reliability is the required on an application. Examples are Dura GB200, GF200, and Crane 2800.



TANDEM (DUAL UNPRESSURIZED):

Due to health, safety, and environmental considerations, tandem seals have been used for products such as vinyl chloride, carbon monoxide, light hydrocarbons, and a wide range of other volatile, toxic, carcinogenic, or hazardous liquids.

Tandem seals eliminate icing and freezing of light hydrocarbons and other liquids which could fall below the atmospheric freezing point of water in air (32°F or 0°C). (Typical buffer liquids in these applications are ethylene glycol, methanol, and propanol.) A tandem also increases online reliability. If the primary seal fails, the outboard seal can take over and function until maintenance of the equipment can be scheduled. Examples are Dura TMB-73 and tandem PTO.

Mechanical Seal Selection

The proper selection of a mechanical seal can be made only if the full operating conditions are known:

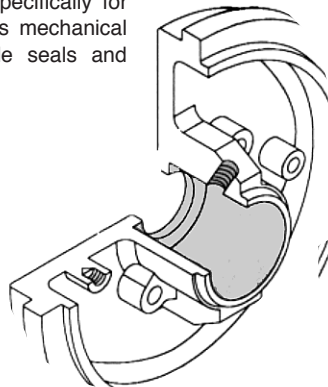
1. **Liquid.** Identification of the exact liquid to be handled is the first step in seal selection. The metal parts must be corrosion resistant, usually steel, bronze, stainless steel, or Hastelloy. The mating faces must also resist corrosion and wear. Carbon, ceramic, silicon carbide or tungsten carbide may be considered. Stationary sealing members of Buna, EPR, Viton and Teflon are common.
2. **Pressure.** The proper type of seal, balanced or unbalanced, is based on the pressure on the seal and on the seal size.
3. **Temperature.** In part, determines the use of the sealing members. Materials must be selected to handle liquid temperature.
4. **Characteristics of Liquid.** Abrasive liquids create excessive wear and short seal life. Double seals or clear liquid flushing from an external source allow the use of mechanical seals on these difficult liquids. On light hydrocarbons balanced seals are often used for longer seal life even though pressures are low.
5. **Reliability and Emission Concerns.** The seal type and arrangement selected must meet the desired reliability and emission standards for the pump application. Double seals and double gas barrier seals are becoming the seals of choice.

Seal Environment

The number one cause of pump downtime is failure of the shaft seal. These failures are normally the result of an unfavorable seal environment such as improper heat dissipation (cooling), poor lubrication of seal faces, or seals operating in liquids containing solids, air or vapors. To achieve maximum reliability of a seal application, proper choices of seal housings (standard bore stuffing box, large bore, or large tapered bore seal chamber) and seal environmental controls (CPI and API seal flush plans) must be made.

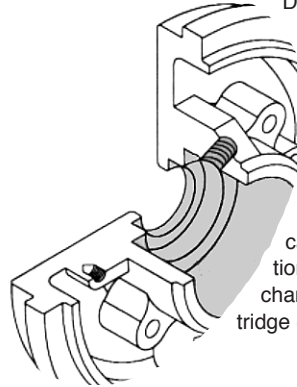
STANDARD BORE STUFFING BOX COVER

Designed thirty years ago specifically for packing. Also accommodates mechanical seals (clamped seat outside seals and conventional double seals.)



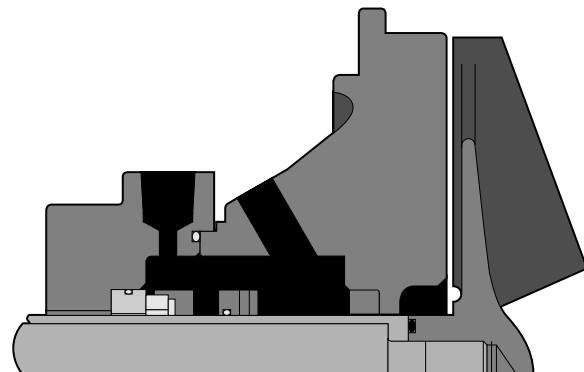
CONVENTIONAL LARGE BORE SEAL CHAMBER

Designed specifically for mechanical seals. Large bore provides increased life of seals through improved lubrication and cooling of faces. Seal environment should be controlled through use of CPI or API flush plans. Often available with internal bypass to provide circulation of liquid to faces without using external flush. Ideal for conventional or cartridge single mechanical seals in conjunction with a flush and throat bushing in bottom of chamber. Also excellent for conventional or cartridge double or tandem seals.

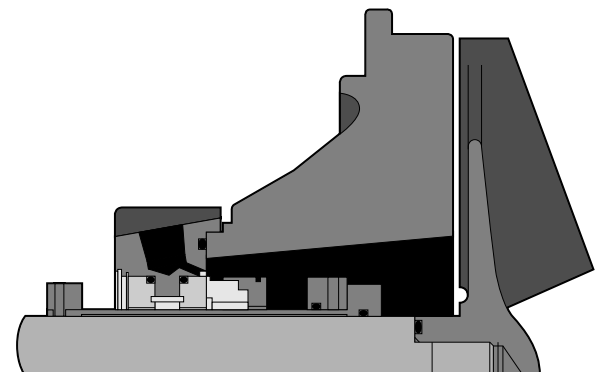


LARGE BORE SEAL CHAMBERS

Introduced in the mid-80's, enlarged bore seal chambers with increased radial clearance between the mechanical seal and seal chamber wall, provide better circulation of liquid to and from seal faces. Improved lubrication and heat removal (cooling) of seal faces extend seal life and lower maintenance costs.



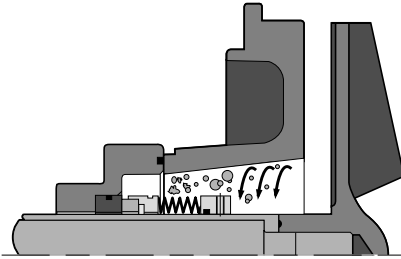
BigBore™ Seal Chamber



TaperBore™ Seal Chamber

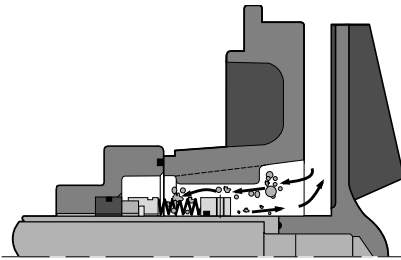
Large Tapered Bore Seal Chambers

Provide increased circulation of liquid at seal faces without use of external flush. Offers advantages of lower maintenance costs, elimination of tubing/piping, lower utility costs (associated with seal flushing) and extended seal reliability. The tapered bore seal chamber is commonly available with ANSI chemical pumps. API process pumps use conventional large bore seal chambers. Paper stock pumps use both conventional large bore and large tapered bore seal chambers. Only tapered bore seal chambers with flow modifiers provide expected reliability on services with or without solids, air or vapors.



Conventional Tapered Bore Seal Chamber: Mechanical Seals Fail When Solids or Vapors Are Present in Liquid

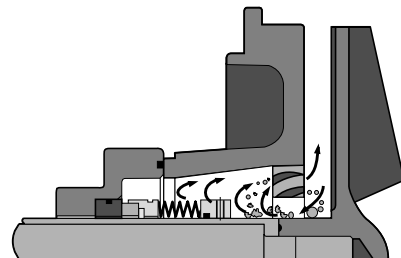
Many users have applied the conventional tapered bore seal chamber to improve seal life on services containing solids or vapors. Seals in this environment failed prematurely due to entrapped solids and vapors. Severe erosion of seal and pump parts, damaged seal faces and dry running were the result.



Modified Tapered Bore Seal Chamber with Axial Ribs: Good for Services Containing Air, Minimum Solids

This type of seal chamber will provide better seal life when air or vapors are present in the liquid. The axial ribs prevent entrapment of vapors through improved flow in the chamber. Dry running failures are eliminated. In addition, solids less than 1% are not a problem.

The new flow pattern, however, still places the seal in the path of solids/liquid flow. The consequence on services with significant solids (greater than 1%) is solids packing the seal spring or bellows, solids impingement on seal faces and ultimate seal failure.



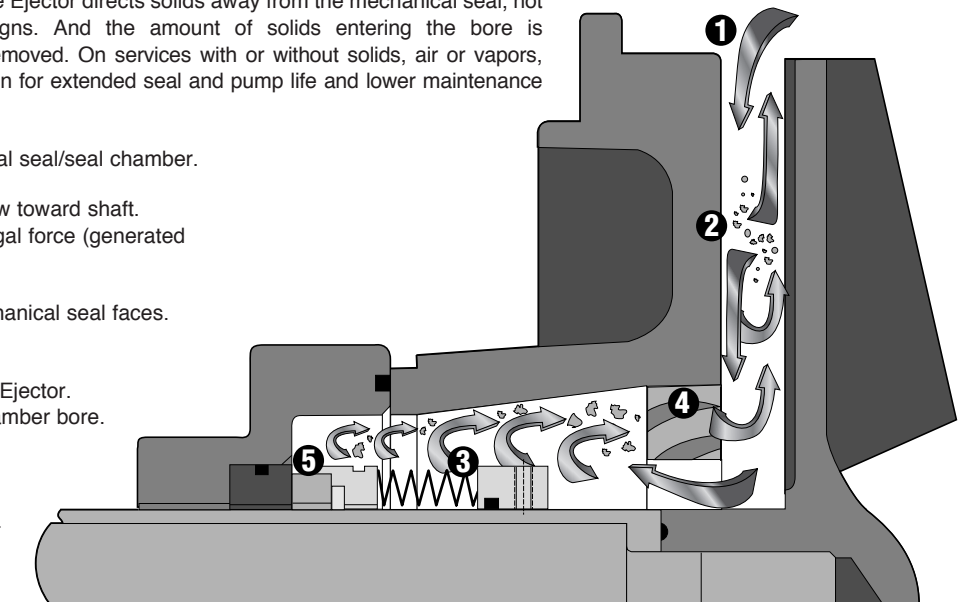
Goulds Standard TaperBore™ PLUS Seal Chamber: The Best Solution for Services Containing Solids and Air or Vapors

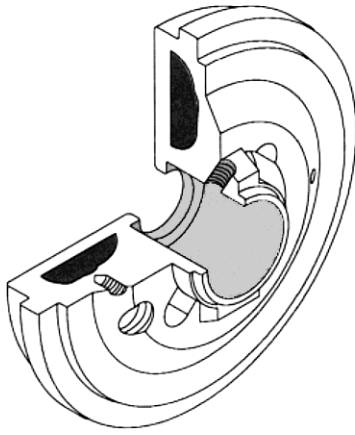
To eliminate seal failures on services containing vapors as well as solids, the flow pattern must direct solids away from the mechanical seal, and purge air and vapors. Goulds Standard TaperBore™ PLUS completely reconfigures the flow in the seal chamber with the result that seal failures due to solids are eliminated. Air and vapors are efficiently removed eliminating dry run failures. Extended seal and pump life with lower maintenance costs are the results.

Goulds TaperBore™ Plus: How It Works

The unique flow path created by the Vane Particle Ejector directs solids away from the mechanical seal, not at the seal as with other tapered bore designs. And the amount of solids entering the bore is minimized. Air and vapors are also efficiently removed. On services with or without solids, air or vapors, Goulds TaperBore™ PLUS is the effective solution for extended seal and pump life and lower maintenance costs.

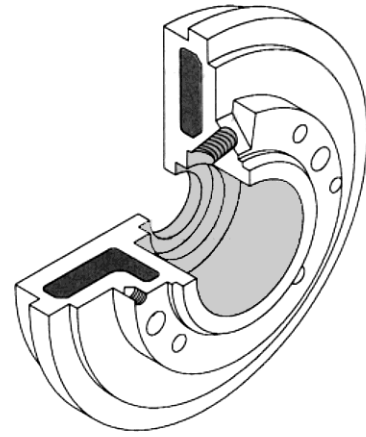
- 1** Solids/liquid mixture flows toward mechanical seal/seal chamber.
- 2** Turbulent zone. Some solids continue to flow toward shaft. Other solids are forced back out by centrifugal force (generated by back pump-out vanes).
- 3** Clean liquid continues to move toward mechanical seal faces. Solids, air, vapors flow away from seal.
- 4** Low pressure zone create by Vane Particle Ejector. Solids, air, vapor liquid mixture exit seal chamber bore.
- 5** Flow in TaperBore™ PLUS seal chamber assures efficient heat removal (cooling) and lubrication. Seal face heat is dissipated. Seal faces are continuously flushed with clean liquid.





JACKETED STUFFING BOX COVER

Designed to maintain proper temperature control (heating or cooling) of seal environment. (Jacketed covers do not help lower seal face temperatures to any significant degree). Good for high temperature services that require use of a conventional double seal or single seal with a flush and API or CPI plan 21.



JACKETED LARGE BORE SEAL CHAMBER

Maintains proper temperature control (heating or cooling) of seal environment with improved lubrication of seal faces. Ideal for controlling temperature for services such as molten sulfur and polymerizing liquids. Excellent for high temperature services that require use of conventional or cartridge single mechanical seals with flush and throat bushing in bottom of seal chamber. Also, great for conventional or cartridge double or tandem seals.

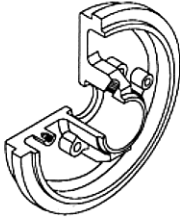
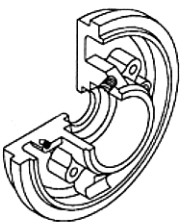
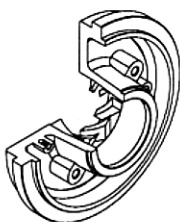
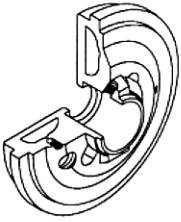
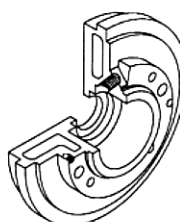
Stuffing Box Cover and Seal Chamber Guides

The following two selection guides are designed to assist selection of the proper seal housing for a pump application.

Stuffing Box and Seal Chamber Application Guide	
Stuffing Box Cover Seal Chamber	Application
Standard Bore Stuffing Box Cover	Use for soft packing. Outside mechanical seals. Double seals. Also, accommodates other Mechanical seals.
Jacketed Stuffing Box Cover	Same as but also need to control temperatures of liquid in seal area.
Conventional Large Bore	Use for all mechanical seal applications where the seal environment requires use of CPI or API seal flush pans. Cannot be used with outside type mechanical seals
Jacked Large Bore	Same as Large Bore but also need to control temperature of liquid in seal area.
Tapered Large Bore with Axial Ribs	Clean services that require use of single mechanical seals. Can also be used with cartridge double seals. Also, effective on services with light solids up to 1% by weight. Paper stock to 1% by weight.
Tapered Large Bore with Patented Vane Particle Ejector (Alloy Construction)	Services with light to moderate solids up to 10% by weight. Paper stock to 5% by Weight. Ideal for single mechanical seals. No flush required. Also, accommodates cartridge double seals. Cannot be used with outside mechanical seals.

Selection Guide

Goulds Engineered Seal Chambers Provide Best Seal Environment For Selected Sealing Arrangements/Services

	TYPE 1 Standard Bore Stuffing Box Cover <i>Designed for packing. Also accommodates mechanical seals.</i>	TYPE 2 Conventional Large Bore <i>Enlarged chamber for increased seal life through improved lubrication and cooling. Seal environment should be controlled through use of CPI flush plans.</i>	TYPE 3 Tapered Bore <i>Lower seal face temperatures, self-venting and draining. Solids and vapors circulated away from seal faces. Often no flush required. Superior patented design maximizes seal life with or without solids and vapor in liquid.</i>	TYPE 4 Jacketed Stuffing Box <i>Maintains proper temperature control (heating or cooling) of seal environment.</i>	TYPE 5 Jacketed Large Bore <i>Maintains proper temperature control (heating or cooling) of seal environment with improved lubrication of seal faces. Ideal for controlling temperatures on services such as molten sulfur and polymerizing liquids.</i>
A Ideally Suited B Acceptable C Not Recommended Service Acceptable Ideally Suited					
Ambient Water With Flush	A	A	A	-	-
Entrained Air or Vapor	C	B	A	C	B
Solids 0-10%, No Flush	C	C	A	C	C
Solids up to and greater than 10% With Flush	B	A	A	B	A
Paper Stock 0-5%, With No Flush	C	C	A	-	-
Paper Stock 0-5%, With Flush	B	A	A	-	-
Slurries 0-5%, No Flush	C	C	A	C	C
High Boiling Point Liquids, no flush	C	C	A	C	C
Temperature Control	C	C	C	B	A
Self-Venting and Draining	C	C	A	C	C
Seal Face Heat Removal	C	A	A	C	A
Molten or Polymerizing Liquid, No Flush	C	C	B	C	C
Molten or Polymerizing Liquid With Flush	C	B	B	C	A

Environmental Controls

Environmental controls are necessary for reliable performance of a mechanical seal on many applications. Goulds Pumps and the seal vendors offer a variety of arrangements to combat these problems.

1. Corrosion
2. Temperature Control
3. Dirty or incompatible environments

CORROSION

Corrosion can be controlled by selecting seal materials that are not attacked by the pumpage. When this is difficult, external fluid injection of a non-corrosive chemical to lubricate the seal is possible. Single or double seals could be used, depending on if the customer can stand delusion of his product.

TEMPERATURE CONTROL

As the seal rotates, the faces are in contact. This generates heat and if this heat is not removed, the temperature in the stuffing box or seal chamber can increase and cause sealing problems. A simple by-pass of product over the seal faces will remove the heat generated by the seal (Fig. 25). For higher temperature services, by-pass of product through a cooler may be required to cool the seal sufficiently (Fig. 26). External cooling fluid injection can also be used.

DIRTY or INCOMPATIBLE ENVIRONMENTS

Mechanical seals do not normally function well on liquids which contain solids or can solidify on contact with the atmosphere. Here, by-pass flush through a filter, a cyclone separator or a strainer are methods of providing a clean fluid to lubricate seal faces.

Strainers are effective for particles larger than the openings on a 40 mesh screen.

Cyclone separators are effective on solids 10 micron or more in diameter, if they have a specific gravity of 2.7 and the pump develops a differential pressure of 30-40 psi. Filters are available to remove solids 2 microns and larger.

If external flush with clean liquid is available, this is the most fail proof system. Lip seal or restricting bushings are available to control flow of injected fluid to flows as low as $\frac{1}{8}$ GPM.

Quench type glands are used on fluids which tend to crystallize on exposure to air. Water or steam is put through this gland to wash away any build up. Other systems are available as required by the service.

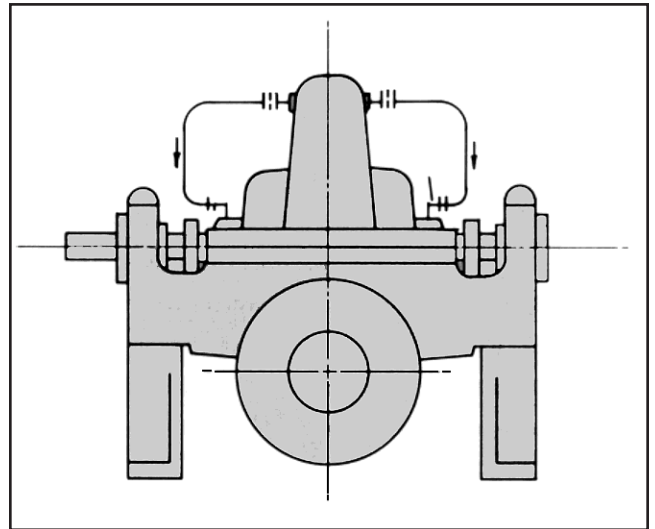


Fig. 25

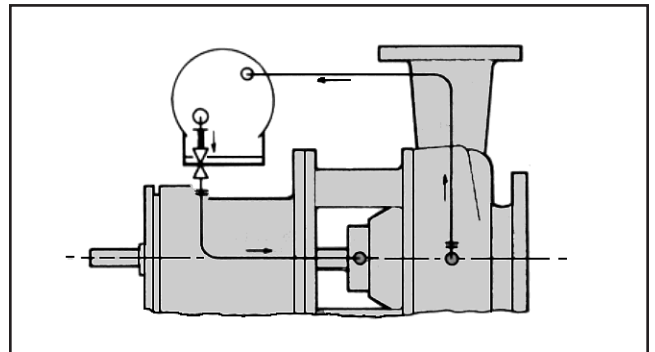


Fig. 26

API and CPI Plans

API and CPI mechanical seal flush plans are commonly used with API and CPI process pumps. The general arrangement of the plans are similar regardless of the designation whether API or CPI. The difference between the flush plans is the construction which provides applicable pressure-temperature capability for each type of pump. API plans have higher pressure and temperature capability than CPI plans. Each plan helps provide critical lubrication and cooling of seal faces to maximize seal reliability.

Plan No.	Recommended Applications
01	Single mechanical seals and TDH less than 125 feet.
02	Used with some outside seals. In most cases not recommended.
11	Single and tandem seals. Always consider a plan 11 with balanced seals. Apply when TDH is greater than 125 ft.
12	Same application as 11. Additionally, a 12 will strain particles from the flush liquid. This helps prevent solid impingement on seal faces.
13	Single and tandem seals. Use when difference in pressure between the seal chamber or stuffing box and pump suction exceed 35 psi.
21	Single and tandem seals. Required when the flush needs to be cooled before flushing at the seal faces. (ex. water above 200°F, light hydrocarbons or any other liquids with poor lubricating qualities and high vapor pressures.)
22	Same application as 21. Additionally, a plan 22 will strain particles from the flush liquid. This helps prevent solid impingement on seal faces.
23	Single and tandem seals. Use when difference in pressure between the seal chamber or stuffing box and pump suction exceed 35 psi. 3600 RPM only.
31	Single and tandem seals. Apply when strainers are inadequate to clean flushing liquid.
32	Single and tandem seals. Required when pumpage is not suitable to lubricate seal faces. Use of bushing or lip seal is also recommended.
33	Used with double seals when external system is available from user.
41	Apply with liquids that require simultaneous cyclone separation and cooling. (Single and tandem seals).
51	Single seals. Required when sealed liquid will crystallize, coke, solidify, etc. at seal faces if contact with air. Common blankets are isopropyl alcohol, glycol, and water. Normally used with FVD gland and bushing or packed auxiliary box.
52	Tandem seals. Plan provides buffer liquid for outside seal. A plan 01 or plan 11 is also recommended with tandem seals to properly flush inboard seal. Pumping rings recommended.
53	Double seals. Plan provides flushing and cooling to both sets of seal faces. Pumping ring recommended.
54	Double seals or packed auxiliary stuffing box.

Dynamic Seal - an Alternative to the Mechanical Seal

On some tough pumping services like paper stock and slurries, mechanical seals require outside flush and constant, costly attention. Even then, seal failures are common, resulting in downtime.

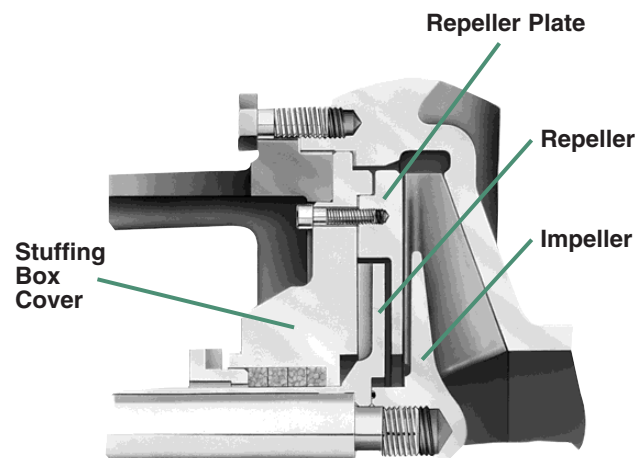
Goulds offers a Dynamic Seal which, simply by fitting a repeller between the stuffing box and impeller, eliminates the need for a mechanical seal.

BENEFITS OF GOULDS DYNAMIC SEAL:

- External seal water not required.
- Elimination of pumpage contamination and product dilution
- Reduces utility cost
- No need to treat seal water
- Eliminates problems associated with piping from a remote source

HOW IT WORKS

At start-up, the repeller functions like an impeller, and pumps liquid and solids from the stuffing box. When pump is shut down, packing (illustrated) or other type of secondary seal prevents pumpage from leaking.



TECH-B-4B Magnetic Drive Pumps

INTRODUCTION

Environmental concerns and recurring mechanical seal problems have created a need for sealless pumps in the chemical and petrochemical industries. In some cases, more stringent regulations by the EPA, OSHA and local agencies are mandating the use of sealless pumps. One type of sealless pump is the magnetic drive pump which uses a permanent magnetic coupling to transmit torque to the impeller without the need for a mechanical seal for packing.

PRINCIPLES OF OPERATION

Magnetic drive pumps use a standard electric motor to drive a set of permanent magnets that are mounted on a carrier or drive assembly located outside of the containment shell. The drive magnet assembly is mounted on a second shaft which is driven by a standard motor. The external rotating magnetic field drives the inner rotor.

The coaxial synchronous torque coupling consists of two rings of permanent magnets as shown in Fig. 1. A magnetic force field is established between the north and south pole magnets in the drive and driven assemblies. This provides the no slip or synchronous capability of the torque coupling. The magnetic field is shown as dashed lines and shaded areas in Fig. 3.

Fig. 1 Typical Magnetic Drive Pump

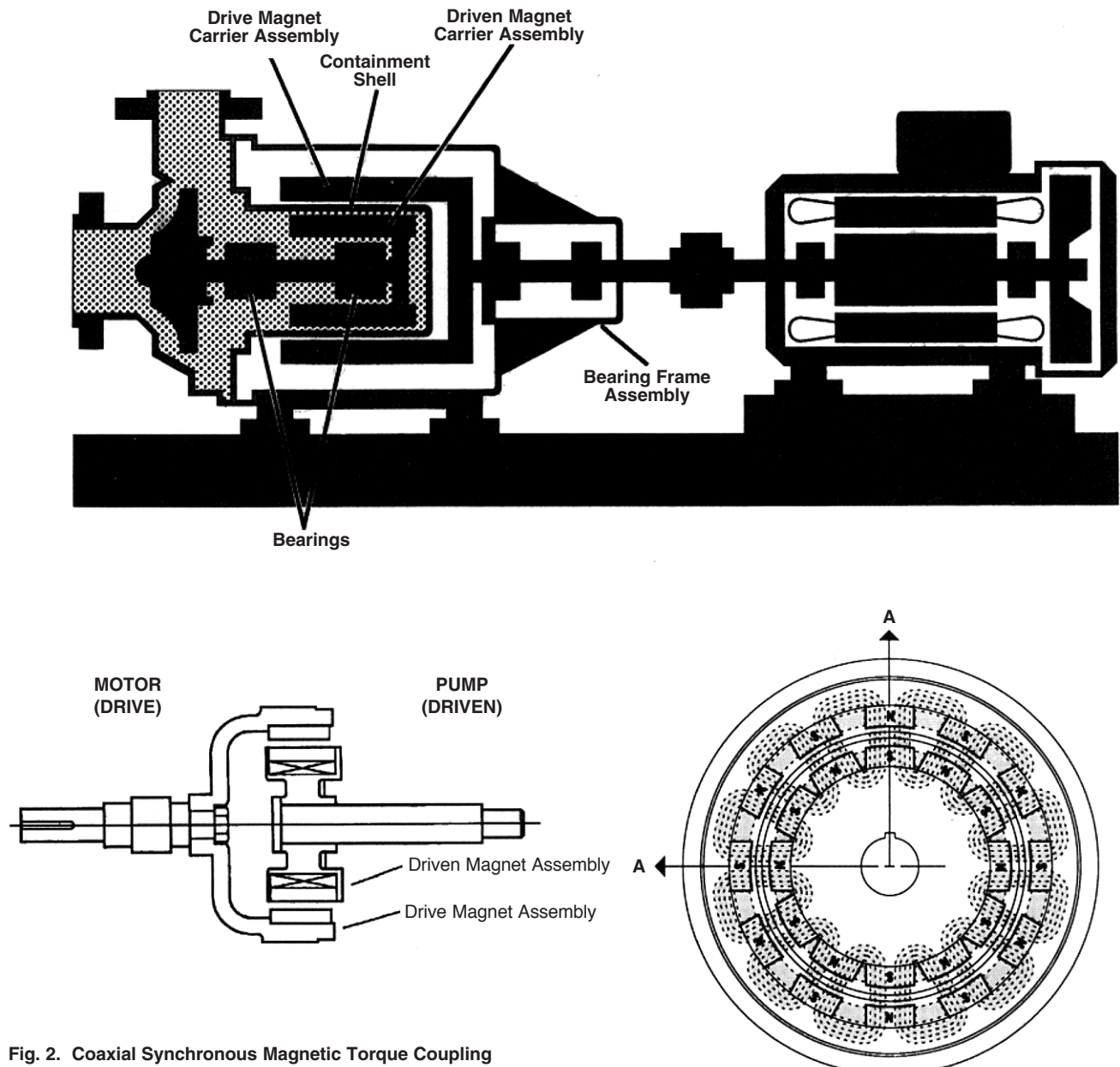


Fig. 2. Coaxial Synchronous Magnetic Torque Coupling

Fig. 3

Two Types of Magnetic Drive Pump Designs

A. Rotating Driven Shaft

This type of design typically uses metal components and is best suited for heavy duty applications. The metallic construction offers the best strength, temperature and pressure capability required for heavy duty applications. Corrosion resistant high alloy materials such as 316SS, Hastelloy, and Alloy 20 are offered. The rotating shaft does, however, increase the number of parts required and thus increases the complexity and cost of the pump.

This type of design typically uses a pressurized recirculation circuit, which helps prevent vaporization of liquid required for process lubricated bearings. (Refer to Model 3296, Section CHEM-3A).

B. Stationary Shaft

This type of design typically uses non-metallic components such as ceramics and plastics. It is best suited for light to medium duty applications. The stationary shaft design significantly reduces the number of parts required, simplifying maintenance and reducing cost. Corrosion resistant materials such as silicon carbide ceramics and fluoropolymer plastics (Teflon, Tefzel, etc.) provide excellent range of application. The use of plastics materials does, however, limit the temperature range of these designs to 200° F to 250° F. (Refer to Model 3298, Section CHEM-3C).

Containment Shell Designs

The containment shell is the pressure containing barrier which is fitted between the drive and the driven magnet assembly. It must contain full working pressure of the pump, since it isolates the pumped liquid from the atmosphere. One-piece formed shells offer the best reliability, eliminating welds used for two-piece shells.

Since the torque coupling magnetic force field must pass through the shell, it must be made of a non-magnetic material. Non-magnetic metals such as Hastelloy and 316SS are typical choices for the containment shell. The motion of the magnets past an electrically conductive containment shell produces eddy currents, which generate heat and must be removed by a process fluid recirculation circuit.

The eddy currents also create a horsepower loss, which reduces the efficiency of the pump. Metals with low electrical conductivity have lower eddy current losses, providing superior pump efficiency. Hastelloy has a relatively low electrical conductivity and good corrosion resistance, thus is an excellent choice for metal containment shells. Electrically non-conductive materials such as plastic and ceramics are also good choices for containment shells, since the eddy current losses are totally eliminated. This results in pump efficiencies equal to conventionally sealed pumps. Plastic containment shells are generally limited to lower pressures and temperatures due to the limited strength of plastics.

Sleeve and Thrust Bearings

Magnetic drive pumps utilize process lubricated bearings to support the inner drive rotor. These bearings are subject to the corrosive nature of the liquids being pumped, thus need to be made from corrosion resistant materials. Two commonly used materials are hard carbon and silicon carbide (SiC). Pure sintered SiC is superior to reaction bonded SiC, since reaction bonded SiC has free silicon left in the matrix, resulting in lower chemical resistance and lower strength.

Hard carbon against silicon carbide offers excellent service life for many chemical applications and also offers the advantage of short term operation in marginal lubrication conditions.

Silicon carbide against silicon carbide offers excellent service life for nearly all chemical applications. Its hardness, high thermal conductivity, and strength make it an excellent bearing material. Silicon carbide must be handled carefully to prevent chipping. Silicon carbide against silicon carbide has very limited capability in marginal lubrication conditions.

Recirculation Circuit

All magnetic drive pumps circulate some of the process fluid to lubricate and cool the bearings supporting the inner rotor.

Magnetic drive pumps with metal containment shells, also require a circulation of some process fluid through the containment shell to remove heat generated by eddy currents. For pumps with metal containment shells, the fluid recirculation path must be carefully engineered to prevent vaporization of the process liquid necessary to lubricate the bearings. A pressurized circuit as shown in Fig. 4 offers excellent reliability for pumps with metal containment shells.

Magnetic drive pumps with electrically non-conductive containment shells, such as plastic or ceramic have no heat generated by eddy currents. Since no heat is required to be removed from the containment shell, a much simpler recirculation circuit can be used.

For liquids near vaporization, a calculation must be made to ensure the process fluid does not vaporize at the bearings. This calculation includes the effects of process fluid specific heat, vapor pressure, drive losses, recirculation flow, etc. This calculation procedure can be found in the GOULDS PUMPS HANDBOOK FOR MAGNETIC DRIVE PUMPS. An external cooling system can be added to the recirculation circuit to prevent vaporization.

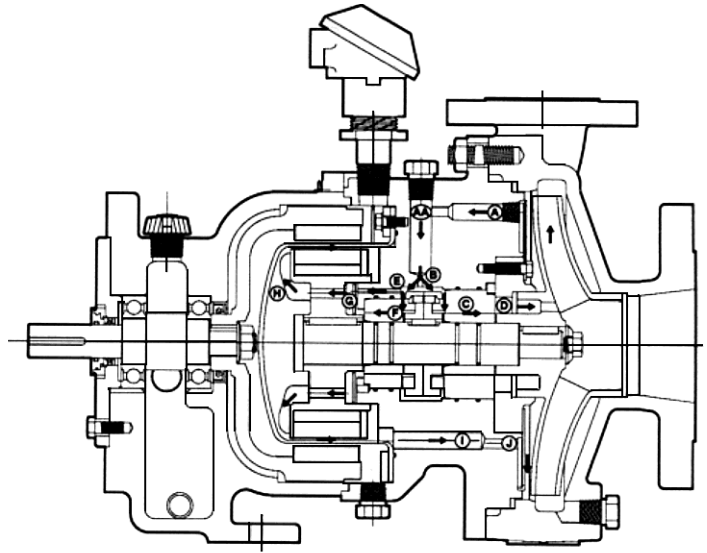


Fig. 4 Recirculation Circuit

Fail Safe Devices

DESCRIPTION

Condition monitoring of the pump is a "key objective" and provides the user with an assurance of safety and reliability.

System and pump malfunctions can result from the following:

- No-flow condition through the pump
- Dry running as a result of plugged liquid circulation paths in the pump bearing and magnets assembly section
- Cavitation due to insufficient $NPSH_A$
- Uncoupling of the magnetic drive due to overload
- Temperature and pressure transients in the system
- "Flashing" in the pump liquid circulation paths due to pressure and temperature transients.

These malfunctions can contribute to:

- Overheating of the drive and driven magnet assemblies
- Overload of drive motor and drive magnetic assembly
- Extreme pump bearing load conditions
- Damage to pump due to extremes in temperatures and pressures due to transients that exceed normal design.

Various fail safe devices are available with the pump to control malfunctions and provide safety and reliability including:

- thermocouple / controller
- low amp relay
- liquid leak detector
- power monitor

TECH-B-5 Field Testing Methods

A. Determination of total head

The total head of a pump can be determined by gauge readings as illustrated in Fig. 1.

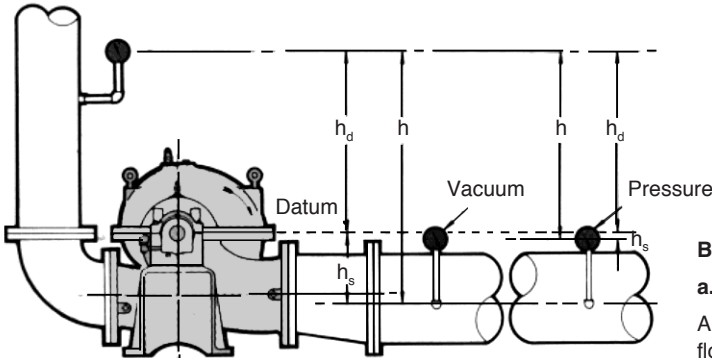


Fig. 1 Determination of Total Head From Gauge Readings

Negative Suction Pressure:

TDH = Discharge gauge reading converted to feet of liquid + vacuum gauge reading converted to feet of liquid + distance between point of attachment of vacuum gauge and the centerline of the discharge gauge, h, in feet + $\left(\frac{Vd^2}{2g} - \frac{Vs^2}{2g} \right)$

Positive Suction Pressure:

or TDH = Discharge gauge reading converted to feet of liquid-pressure gauge reading in suction line converted to ft. of liquid + distance between center of discharge and suction gauges, h, in feet + $\left(\frac{Vd^2}{2g} - \frac{Vs^2}{2g} \right)$

In using gauges when the pressure is positive or above atmospheric pressure, any air in the gauge line should be vented off by loosening the gauge until liquid appears. This assures that the entire gauge line is filled with liquid and thus the gauge will read the pressure at the elevation of the centerline of the gauge. However, the gauge line will be empty of liquid when measuring vacuum and the gauge will read the vacuum at the elevation of the point of attachment of the gauge line to the pipe line. These assumptions are reflected in the above definitions.

The final term in the above definitions accounts for a difference in size between the suction and discharge lines. The discharge line is normally smaller than the suction line and thus the discharge velocity is higher. A higher velocity results in a lower pressure since the sum of the pressure head and velocity head in any flowing liquid remains constant. Thus, when the suction and discharge line sizes at the gauge attachment points are different, the resulting difference in velocity head must be included in the total head calculation.

Manometers can also be used to measure pressure. The liquid used in a manometer is normally water or mercury, but any liquid of known specific gravity can be used. Manometers are extremely accurate for determining low pressures or vacuums and no calibration is needed. They are also easily fabricated in the field to suit any particular application. Figs. 2 & 3 illustrate typical manometer set ups.

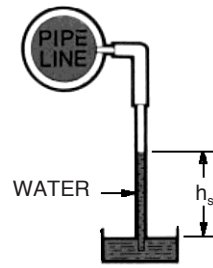


Fig. 2 Manometer Indicating Vacuum

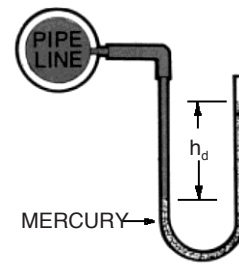


Fig. 3 Manometer Indicating Pressure

B. Measurement of capacity

a.) Magnetic Flow Meter

A calibrated magnetic flow meter is an accurate means of measuring flow in a pumping system. However, due to the expense involved, magnetic flow meters are only practical in small factory test loops and in certain process pumping systems where flow is critical.

b.) Volumetric measurement

Pump capacity can be determined by weighing the liquid pumped or measuring its volume in a calibrated vessel. This is often practical when pumping into an accurately measured reservoir or tank, or when it is possible to use small containers which can be accurately weighed. These methods, however, are normally suited only to relatively small capacity systems.

c.) Venturi meter

A venturi meter consists of a converging section, a short constricting throat section and then a diverging section. The object is to accelerate the fluid and temporarily lower its static pressure. The flow is then a function of the pressure differential between the full diameter line and the throat. Fig. 4 shows the general shape and flow equation. The meter coefficient is determined by actual calibration by the manufacturer and when properly installed the Venturi meter is accurate to within plus or minus 1%.

$$Q(\text{GPM}) = 5.67 C D_2^2 \sqrt{\frac{H}{1 - R^4}}$$

C = Instrument Coefficient

D₁ = Entrance Diameter in Inches

D₂ = Throat Diameter in Inches

R = D₂/D₁

H = Differential Head in Inches = h₁ - h₂

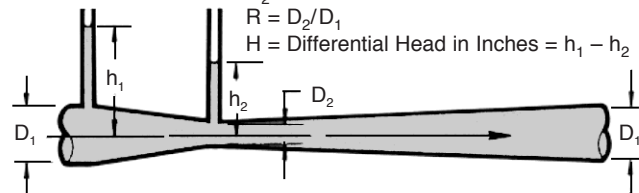


Fig. 5 Venturi Meter

d.) Nozzle

A nozzle is simply the converging portion of a venturi tube with the liquid exiting to the atmosphere. Therefore, the same formula can be used with the differential head equal to the gauge reading ahead of the nozzle. Fig. 5 lists theoretical nozzle discharge flows.

Theoretical Discharge of Nozzles in U.S. GPM

Head		Veloc'y of Disch. Feet per Sec.	Diameter of Nozzle in Inches												
Lbs.	Feet		1/16	1/8	3/16	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8
10	23.1	38.6	0.37	1.48	3.32	5.91	13.3	23.6	36.9	53.1	72.4	94.5	120	148	179
15	34.6	47.25	0.45	1.81	4.06	7.24	16.3	28.9	45.2	65.0	88.5	116.	147	181	219
20	46.2	54.55	0.52	2.09	4.69	8.35	18.8	33.4	52.2	75.1	102.	134.	169	209	253
25	57.7	61.0	0.58	2.34	5.25	9.34	21.0	37.3	58.3	84.0	114.	149	189	234	283
30	69.3	66.85	0.64	2.56	5.75	10.2	23.0	40.9	63.9	92.0	125.	164.	207	256	309
35	80.8	72.2	0.69	2.77	6.21	11.1	24.8	44.2	69.0	99.5	135.	177.	224	277	334
40	92.4	77.2	0.74	2.96	6.64	11.8	26.6	47.3	73.8	106.	145.	189.	239	296	357
45	103.9	81.8	0.78	3.13	7.03	12.5	28.2	50.1	78.2	113.	153.	200.	253	313	379
50	115.5	86.25	0.83	3.30	7.41	13.2	29.7	52.8	82.5	119.	162.	211.	267	330	399
55	127.0	90.4	0.87	3.46	7.77	13.8	31.1	55.3	86.4	125.	169.	221.	280	346	418
60	138.6	94.5	0.90	3.62	8.12	14.5	32.5	57.8	90.4	130.	177.	231.	293	362	438
65	150.1	98.3	0.94	3.77	8.45	15.1	33.8	60.2	94.0	136.	184.	241.	305	376	455
70	161.7	102.1	0.98	3.91	8.78	15.7	35.2	62.5	97.7	141.	191.	250.	317	391	473
75	173.2	105.7	1.01	4.05	9.08	16.2	36.4	64.7	101.	146.	198.	259.	327	404	489
80	184.8	109.1	1.05	4.18	9.39	16.7	37.6	66.8	104.	150.	205.	267.	338	418	505
85	196.3	112.5	1.08	4.31	9.67	17.3	38.8	68.9	108.	155.	211.	276.	349	431	521
90	207.9	115.8	1.11	4.43	9.95	17.7	39.9	70.8	111.	160.	217.	284.	359	443	536
95	219.4	119.0	1.14	4.56	10.2	18.2	41.0	72.8	114.	164.	223.	292.	369	456	551
100	230.9	122.0	1.17	4.67	10.5	18.7	42.1	74.7	117.	168.	229.	299.	378	467	565
105	242.4	125.0	1.20	4.79	10.8	19.2	43.1	76.5	120.	172.	234	306.	388	479	579
110	254.0	128.0	1.23	4.90	11.0	19.6	44.1	78.4	122.	176.	240	314.	397	490	593
115	265.5	130.9	1.25	5.01	11.2	20.0	45.1	80.1	125.	180.	245.	320.	406	501	606
120	277.1	133.7	1.28	5.12	11.5	20.5	46.0	81.8	128.	184.	251.	327.	414	512	619
125	288.6	136.4	1.31	5.22	11.7	20.9	47.0	83.5	130.	188.	256.	334.	423	522	632
130	300.2	139.1	1.33	5.33	12.0	21.3	48.0	85.2	133.	192.	261.	341.	432	533	645
135	311.7	141.8	1.36	5.43	12.2	21.7	48.9	86.7	136.	195.	266.	347.	439	543	656
140	323.3	144.3	1.38	5.53	12.4	22.1	49.8	88.4	138.	199.	271.	354	448	553	668
145	334.8	146.9	1.41	5.62	12.6	22.5	50.6	89.9	140.	202.	275.	360	455	562	680
150	346.4	149.5	1.43	5.72	12.9	22.9	51.5	91.5	143.	206.	280.	366.	463	572	692
175	404.1	161.4	1.55	6.18	13.9	24.7	55.6	98.8	154.	222.	302.	395.	500	618	747
200	461.9	172.6	1.65	6.61	14.8	26.4	59.5	106.	165.	238.	323.	423	535	660	799
250	577.4	193.0	1.85	7.39	16.6	29.6	66.5	118.	185.	266.	362.	473.	598	739	894
300	692.8	211.2	2.02	8.08	18.2	32.4	72.8	129.	202.	291.	396.	517.	655	808	977
			1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	5 1/2	6
10	23.1	38.6	213	289	378	479	591	714	851	1158	1510	1915	2365	2855	3405
15	34.6	47.25	260	354	463	585	723	874	1041	1418	1850	2345	2890	3490	4165
20	46.2	54.55	301	409	535	676	835	1009	1203	1638	2135	2710	3340	4040	4810
25	57.7	61.0	336	458	598	756	934	1128	1345	1830	2385	3025	3730	4510	5380
30	69.3	66.85	368	501	655	828	1023	1236	1473	2005	2615	3315	4090	4940	5895
35	80.8	72.2	398	541	708	895	1106	1335	1591	2168	2825	3580	4415	5340	6370
40	92.4	77.2	425	578	756	957	1182	1428	1701	2315	3020	3830	4725	5280	6380
45	103.9	81.8	451	613	801	1015	1252	1512	1802	2455	3200	4055	5000	6050	7210
50	115.5	86.25	475	647	845	1070	1320	1595	1900	2590	3375	4275	5280	6380	7600
55	127.0	90.4	498	678	886	1121	1385	1671	1991	2710	3540	4480	5530	6690	7970
60	138.6	94.5	521	708	926	1172	1447	1748	2085	2835	3700	4685	5790	6980	8330
65	150.1	98.3	542	737	964	1220	1506	1819	2165	2950	3850	4875	6020	7270	8670
70	161.7	102.1	563	765	1001	1267	1565	1888	2250	3065	4000	5060	6250	7560	9000
75	173.2	105.7	582	792	1037	1310	1619	1955	2330	3170	4135	5240	6475	7820	9320
80	184.8	109.1	602	818	1070	1354	1672	2020	2405	3280	4270	5410	6690	8080	9630
85	196.3	112.5	620	844	1103	1395	1723	2080	2480	3375	4440	5575	6890	8320	9920
90	207.9	115.8	638	868	1136	1436	1773	2140	2550	3475	4530	5740	7090	8560	10210
95	219.4	119.0	656	892	1168	1476	1824	2200	2625	3570	4655	5900	7290	8800	10500
100	230.9	122.0	672	915	1196	1512	1870	2255	2690	3660	4775	6050	7470	9030	10770
105	242.4	125.0	689	937	1226	1550	1916	2312	2755	3750	4855	6150	7590	9170	10950
110	254.0	128.0	705	960	1255	1588	1961	2366	2820	3840	5010	6350	7840	9470	11300
115	265.5	130.9	720	980	1282	1621	2005	2420	2885	3930	5120	6490	8010	9680	11500
120	277.1	133.7	736	1002	1310	1659	2050	2470	2945	4015	5225	6630	8180	9900	11800
125	288.6	136.4	751	1022	1338	1690	2090	2520	3005	4090	5340	6760	8350	10100	12030
130	300.2	139.1	767	1043	1365	1726	2132	2575	3070	4175	5450	6900	8530	10300	12290
135	311.7	141.8	780	1063	1390	1759	2173	2620	3125	4250	5550	7030	8680	10490	12510
140	323.3	144.3	795	1082	1415	1790	2212	2670	3180	4330	5650	7160	8850	10690	12730
145	334.8	146.9	809	1100	1440	1820	2250	2715	3235	4410	5740	7280	8990	10880	12960
150	346.4	149.5	824	1120	1466	1853	2290	2760	3295	4485	5850	7410	9150	11070	13200
175	404.1	161.4	890	1210	1582	2000	2473	2985	3560	4840	6310	8000	9890	11940	14250
200	461.9	172.6	950	1294	1691	2140	2645	3190	3800	5175	6750	8550	10580	12770	15220
250	577.4	193.0	1063	1447	1891	2392	2955	3570	4250	5795	7550	9570	11820	14290	17020
300	692.8	211.2	1163	1582	2070	2615	3235	3900	4650	6330	8260	10480	12940	15620	18610

NOTE: - The actual quantities will vary from these figures, the amount of variation depending upon the shape of nozzle and size of pipe at the point where the pressure is determined. With smooth taper nozzles the actual discharge is about 94% of the figures given in the tables.

Fig. 5

e.) Orifice

An orifice is a thin plate containing an opening of specific shape and dimensions. The plate is installed in a pipe and the flow is a function of the pressure upstream of the orifice. There are numerous types of orifices available and their descriptions and applications are covered in the Hydraulic Institute Standards and the ASME Fluid Meters Report. Orifices are not recommended for permanent installations due to the inherent high head loss across the plate.

f.) Weir

A weir is particularly well suited to measuring flows in open conduits and can be adapted to extremely large capacity systems. For best accuracy, a weir should be calibrated in place. However, when this is impractical, there are formulas which can be used for the various weir configurations. The most common types are the rectangular contracted weir and the 90 V-notch weir. These are shown in Fig. 6 with the applicable flow formulas.

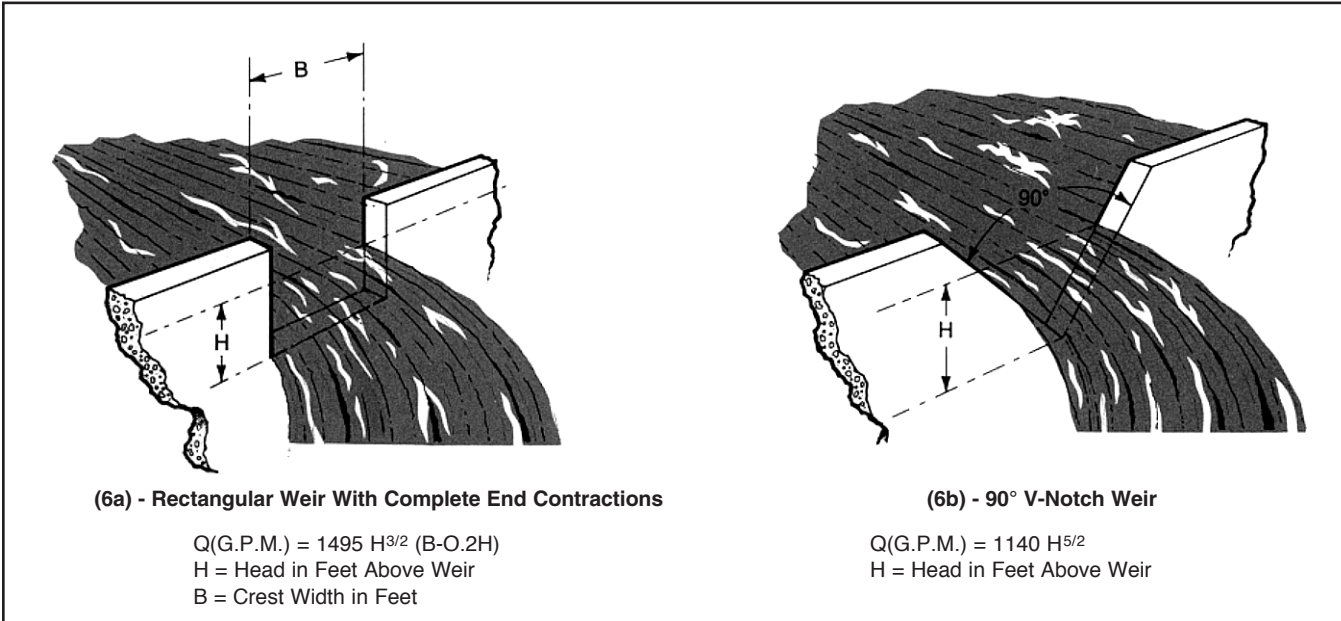


Fig. 6 Weirs

g.) Pilot tube

A pilot tube measures fluid velocity. A small tube placed in the flow stream gives two pressure readings: one receiving the full impact of the flowing stream reads static head + velocity head, and the other reads the static head only (Fig. 7). The difference between the two readings is the velocity head. The velocity and the flow are then determined from the following well known formulas.

$V = C \sqrt{2gh_v}$ where C is a coefficient for the meter determined by calibration, and h_v = velocity head,

Capacity = Area x Average Velocity

Since the velocity varies across the pipe, it is necessary to obtain a velocity profile to determine the average velocity. This involves some error, but when properly applied a calibrated pilot tube is within plus or minus 2% accuracy.

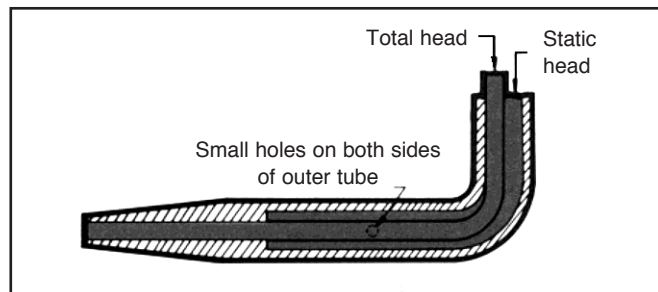


Fig. 7 Pilot Tube

TECH-B-6 Vibration Analysis

Vibration analysis equipment enables you to tell when "normal" vibration becomes "problem" vibration or exceeds acceptable levels. It may also allow you to determine the source and cause of the vibration, thus becoming an effective preventive maintenance and troubleshooting aid.

A vibration analyser measures the amplitude, frequency and phase of vibration. Also when vibration occurs at several frequencies, it separates one frequency from another so that each individual vibration characteristic can be measured.

The vibration pickup senses the velocity of the vibration and converts it into an electrical signal. The analyzer receives this signal, converting it to the corresponding amplitude and frequency.

The amplitude is measured in terms of peak-to-peak displacement in mils (1 mil = .001") and is indicated on the amplitude meter.

Some instruments are equipped with a frequency meter which gives a direct readout of the predominant frequency of the vibration. Other instruments have tunable filters which allow scanning the frequency scale and reading amplitude at any particular frequency, all others being filtered out.

A strob light is used to determine the phase of vibration. It can be made to flash at the frequency of the vibration present or at any arbitrary frequency set on an internal oscillator.

A reference mark on a rotating part viewed under the strob light flashing at the vibration frequency may appear as a single frozen (or rotating) mark, or as several frozen (or rotating) marks. The number of marks viewed is useful in determining the source of the vibration. The location of the mark or marks is used in balancing rotating parts.

The first step in vibration analysis is to determine the severity of the vibration, then, if the vibration is serious, a complete set of vibration readings should be taken before attempting to analyze the cause. Fig. 1 is the general guide for horizontal centrifugal pumps as published by the Hydraulic Institute. The amplitudes shown are the overall maximum obtained without filtering to specific frequencies. Amplitudes at specific frequencies, such as vane pass frequency with multi-vane impellers, should be less than 75% of the unfiltered amplitudes allowed in Fig. 1 at the operating RPM. For horizontal non-clog and vertical submerged pumps, refer to Hydraulic Institute standards or pump manufacturer.

Severity of vibration is a function of amplitude and pump speed; however, it should be noted that a change in severity over a period of time is usually a warning of impending failure. This change is often more important than vibration in the "slightly rough" or "rough" ranges which does not change with time.

Complete pump vibration analysis requires taking vibration readings at each bearing in three planes (horizontal, vertical and axial). Readings at the pump suction and discharge flanges may also be useful in some cases.

After all data has been tabulated, it can be analyzed to determine the most likely cause or causes of vibration and the identifying characteristics of each.

By analyzing the tabulated vibration data one or several causes may be found. Each must be checked, starting with the most likely cause or easiest to check.

For example, assume the axial vibration is 50% or more of the radial vibration and the predominant frequency is the same as the RPM of the pump. The chart indicates probable misalignment or bent shaft. Coupling misalignment is probably the most common single cause of pump vibration and is one of the easiest to check. If after checking, the alignment proves to be good, then inspect for flange loading. Finally, check for a bent shaft. Cavitation in a pump can cause serious vibration. Vibration at random frequencies can also be caused by hydraulic disturbances in poorly designed suction or discharge systems.

The use of vibration equipment in preventative maintenance involves keeping a vibration history on individual pieces of equipment in a plant. A form similar to that shown in Fig 3 can be used to record the vibration data on a periodic routine basis. Abrupt changes are a sign of impending failure. A gradual increase in vibration can also be detected and corrective measures can be taken before it reaches a dangerous level.

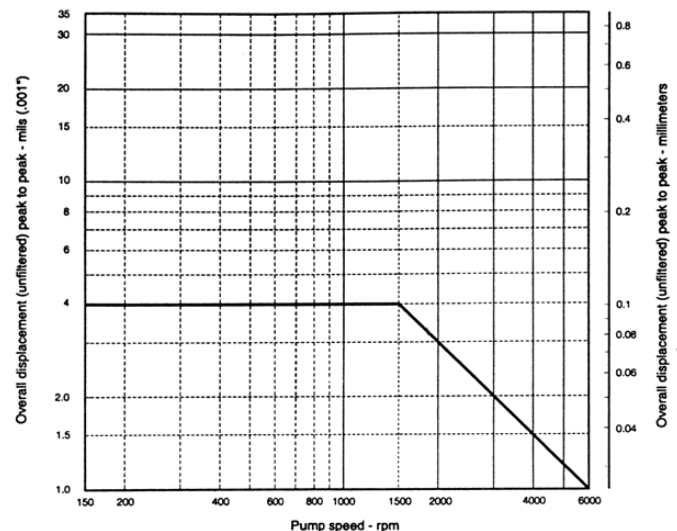


Fig. 1 Acceptable field vibration limits for horizontal or vertical in-line pumps (Figures 1.107 to 1.109) - clear liquids

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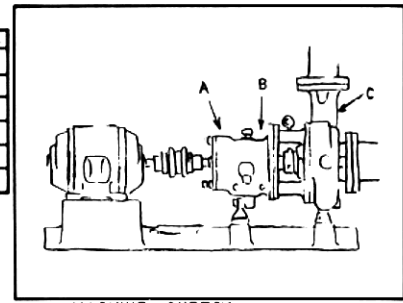
Vibration Analysis – Continued

Cause	Amplitude	Frequency	Phase	Remarks
Unbalance	Largest in radial direction. Proportional to unbalance	1 x RPM	Single reference mark	Unbalance
Misalignment of coupling or bearings and bent shaft	Axial direction vibration 50% or more of radial	1 x RPM normally	single, double, or triple	Easily recognized by large axial vibration. Excessive flange loading can contribute to misalignment
Bad Anti-friction bearings	Unsteady	Very high. Several time RPM	Erratic	Largest high-frequency vibration near the bad bearing.
Mechanical looseness		2 x RPM	Two reference marks. Slightly erratic.	Check grouting and bed plate bolting.
Bad drive belts	Erratic or pulsing	1, 2, 3 & 4 x RPM of belts	Unsteady	Use strobe light to freeze faulty belt.
Electrical	Disappears when power is turned off.	1 or 2 x synchronous frequency	Single or rotating double mark	3600 or 7200 cps for 60 cycle current.
Hydraulic forces		No. of impeller vanes x RPM		Rarely a cause of serious vibration

Fig. 3 Vibration Identification Chart

VIBRATION HISTORY

DATE	10/4
MACHINE	GOULDS MODEL 3196
LOCATION	
NAME	
INSTRUMENT	MODEL 600
S N	222 8078
OPERATOR	WETZ HIGGINS



MACHINE SKETCH

PICKUP POSITION	FILTER OUT		FILTER IN							
	MILS	CPM	MILS	CPM	MILS	CPM	MILS	CPM	IN/SEC	CPM
A	H									
	V									
	A									
B	H									
	V									
	A									

Fig. 4 Vibration Data Sheet

TECH-B-7 Vertical Turbine Pumps

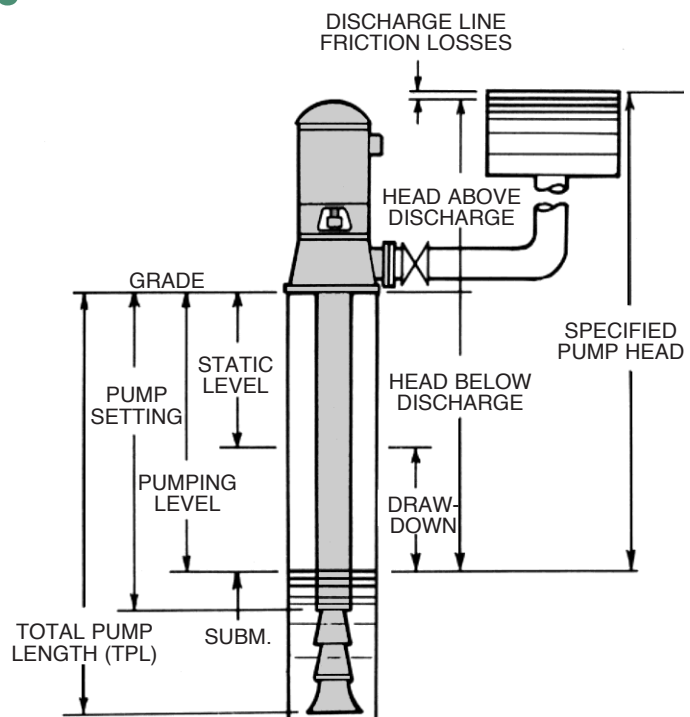
Turbine Nomenclature

- DATUM OR GRADE** - The elevation of the surface from which the pump is supported.
- STATIC LIQUID LEVEL** - The vertical distance from grade to the liquid level when no liquid is being drawn from the well or source.
- DRAWDOWN** - The distance between the static liquid level and the liquid level when pumping at required capacity.
- PUMPING LIQUID LEVEL** - The vertical distance from grade to liquid level when pumping at rated capacity. Pumping liquid level equals static water level plus drawdown.
- SETTING** - The distance from grade to the top of the pump bowl assembly.
- TPL (TOTAL PUMP LENGTH)** - The distance from grade to lowest point of pump.
- RATED PUMP HEAD** - Lift below discharge plus head above discharge plus friction losses in discharge line. This is the head for which the customer is responsible and does not include any losses within the pump.
- COLUMN AND DISCHARGE HEAD FRICTION LOSS** - Head loss in the pump due to friction in the column assembly and discharge head. Friction loss is measured in feet and is dependent upon column size, shaft size, setting, and discharge head size. Values given in appropriate charts in Data Section.

- BOWL HEAD** - Total head which the pump bowl assembly will deliver at the rated capacity. This is curve performance.
- BOWL EFFICIENCY** - The efficiency of the bowl unit only. This value is read directly from the performance curve.
- BOWL HORSEPOWER** - The horsepower - required by the bowls only to deliver a specified capacity against bowl head.

$$\text{BOWL HP} = \frac{\text{Bowl Head} \times \text{Capacity}}{3960 \times \text{Bowl Efficiency}}$$

- TOTAL PUMP HEAD** - Rated pump head plus column and discharge head loss. Note: This is new or final bowl head.
- SHAFT FRICTION LOSS** - The horsepower required to turn the lineshaft in the bearings. These values are given in appropriate table in Data Section.



- PUMP BRAKE HORSEPOWER** - Sum of bowl horsepower plus shaft loss (and the driver thrust bearing loss under certain conditions).

- TOTAL PUMP EFFICIENCY (WATER TO WATER)** - The efficiency of the complete pump less the driver, with all pump losses taken into account.

$$\text{Efficiency} = \frac{\text{Specified Pump Head} \times \text{Capacity}}{3960 \times \text{Brake Horsepower}}$$

- OVERALL EFFICIENCY (WIRE TO WATER)** - The efficiency of the pump and motor complete. Overall efficiency = total pump efficiency x motor efficiency.

- SUBMERGENCE** - Distance from liquid level to suction bell.

Vertical Turbine Pumps - Calculating Axial Thrust

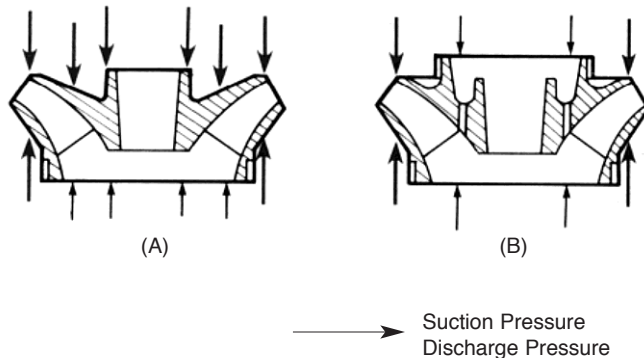
Under normal circumstances Vertical Turbine Pumps have a thrust load acting parallel to the pump shaft. This load is due to unbalanced pressure, dead weight and liquid direction change. Optimum selection of the motor bearing and correct determination of required bowl lateral for deep setting pumps require accurate knowledge of both the magnitude and direction (usually down) of the resultant of these forces. In addition, but with a less significant role, thrust influences shaft H.P. rating and shaft critical speeds.

IMPELLER THRUST

Impeller Thrust in the downward direction is due to the unbalanced discharge pressure across the eye area of the impeller. See diagram A.

Counteracting this load is an upward force primarily due to the change in direction of the liquid passing through the impeller. The resultant of these two forces constitutes impeller thrust. Calculating this thrust using a thrust constant (K) will often produce only an approximate thrust value because a single constant cannot express the upthrust component which varies with capacity.

To accurately determine impeller thrust, thrust-capacity curves based on actual tests are required. Such curves now exist for the "A" Line. To determine thrust, the thrust factor "K" is read from the thrust-capacity curve at the required capacity and given RPM. "K" is then multiplied by the Total Pump Head (Final Lab Head) times Specific Gravity of the pumped liquid. If impeller thrust is excessively high, the impeller can usually be hydraulically balanced. This reduces the value of "K". Balancing is achieved by reducing the discharge pressure above the impeller eye by use of balancing holes and rings. See diagram B.



NOTE:

Although hydraulic balancing reduces impeller thrust, it also decreases efficiency by one to five points by providing an additional path for liquid recirculation. Of even greater concern is that should the hydraulic balancing holes become clogged, (unclean fluids, fluids with solid content, intermittent services, etc.), the impeller thrust will increase and possibly cause the driver to fail. Hydraulically balanced impellers cannot be used in applications requiring rubber bowl bearings because the flutes on the inside diameter of the bearings provide an additional path to the top side of the impeller, thus creating an additional down thrust.

Hydraulically balanced impellers should be used as a "last resort" for those situations where the pump thrust exceeds the motor thrust bearing capabilities.

DEAD WEIGHT

In addition to the impeller force, dead weight (shaft plus impeller weight less the weight of the liquid displaced) acts downward. On pumps with settings less than 50 feet, dead weight may be neglected on all but the most critical applications as it represents only a small part of the total force. On deeper setting pumps, dead weight becomes significant and must be taken into account.

NOTE:

We normally only take shaft weight into consideration as dead weight, the reason being that impeller weight less its liquid displacement weight is usually a small part of the total.

SHAFT SLEEVES

Finally, there can be an upward force across a head shaft sleeve or mechanical seal sleeve. In the case of can pumps with suction pressure, there can be an additional upward force across the impeller shaft area. Again for most applications these forces are small and can be neglected; however, when there is a danger of upthrusts or when there is high discharge pressure (above 600 psi) or high suction pressure (above 400 psi) these forces should be considered.

MOTOR BEARING SIZING

Generally speaking a motor for a normal thrust application has as standard, a bearing adequate for shutoff thrust. When practical, motor bearings rated for shutoff conditions are preferred. For high thrust applications (when shutoff thrust exceeds the standard motor bearing rating) the motor bearing may be sized for the maximum anticipated operating range of the pump.

Should the pump operate to the left of this range for a short period of time, anti-friction bearings such as angular contact or spherical roller can handle the overload. It should be remembered, however, that bearing life is approximately inversely proportional to the cube of the load. Should the load double, motor bearing life will be cut to $\frac{1}{8}$ of its original value. Although down thrust overloading is possible, the pump must never be allowed to operate in a continuous up thrust condition even for a short interval without a special motor bearing equipped to handle it. Such upthrust will tail the motor bearing.

CALCULATING MOTOR BEARING LOAD

As previously stated, for short setting non-hydraulic balanced pumps below 50 feet with discharge pressures below 600 psi and can pumps with Suction pressures below 100 psi only impeller thrust need be considered.

Under these conditions:	Where:
Motor Bearing Load (lbs.)	Impeller Thrust (lbs.)
$T_{imp} = KH_L \times SG$	$K = \text{Thrust factors (lbs./ft.)}$
	$H_L = \text{Lab Head (ft.)}$
	$SG = \text{Specific Gravity}$

For more demanding applications, the forces which should be considered are impeller thrust plus dead weight minus any sleeve or shaft area force.

In equation form:

$$\text{Motor Bearing Load} = T_{imp} + Wt^{(1)} - \text{sleeve force}^{(2)} - \text{shaft area force}^{(3)} = T_t$$

CALCULATING AXIAL THRUST - CONTINUED

Shaft Dia (in)	Shaft Dead Wt. (lbs/ft.)		Shaft Area (in ²)	Sleeve Area (in ²)
	Open Lineshaft	Closed Lineshaft		
1	2.3	2.6	.78	1.0
1 ³ / ₁₆	3.3	3.8	1.1	1.1
1 ¹ / ₂	5.3	6.0	1.8	1.1
1 ¹¹ / ₁₆	6.7	7.6	2.2	1.5
1 ¹⁵ / ₁₆	8.8	10.0	2.9	1.8
2 ³ / ₁₆	11.2	12.8	3.7	2.0

(1) Wt.= Shaft Dead Wt. x Setting In Ft.

(2) Sleeve Force=Sleeve area x Discharge pressure

(3) Shaft Area Force = Shaft area x Suction pressure

*Oil Lube shaft does not displace liquid above the pumping water level and therefore has a greater net weight.

THRUST BEARING LOSS

Thrust bearing loss is the loss of horsepower delivered to the pump at the thrust bearings due to thrust. In equation form:

$$L_{TB} = .0075 \left(\frac{BHP}{100} \right) \left(\frac{T_t}{1000} \right)$$

where:

L_{TB} = Thrust bearing loss (HP)

BHP = Brake horsepower

T_t = Motor Bearing Load (Lbs.)

= $T_{imp} + Wt^{(1)} - \text{sleeve force}^{(2)} - \text{shaft area force}^{(3)}$

Vertical Turbine Bearing Material Data

Material Description	Temp. and S.G. Limits	Remarks
1. Bronze-SAE 660 (Standard) #1104 ASTM-B-584-932	-50 to 250°F. Min S.G. of 0.6	General purpose material for non- abrasive, neutral pH service. 7% Tin/7% Lead/3% Zinc/83% Cu.
2. Bronze-SAE 64 (Zincless) #1107 ASTM-B-584-937	-50 to 180°F. Min. S.G. of 0.6	Similar to std. Bronze. Used for salt water services. 10% Tin/ 10% Lead/80% Cu.
3. Carbon Graphite Impregnated with Babbit ²	-450 to 300o F. All Gravities	Corrosion resistant material not suitable for abrasive services. Special materials available for severe acid services and for temp. as high as 650°. Good for low specific gravity fluids because the carbon is self-lubricating.
4. Teflon 25% Graphite with 75% Teflon	-50 to 250° F. All Gravities	Corrosion resistant except for highly oxidizing solutions. Not suitable for abrasive services. Glass filled Teflon also available.
5. Cast Iron3 ASTM-A-48 CL30 Flash Chrome Coated	32 to 180° F. Min. S.G of 0.6	Used on non abrasive caustic services and some oil products. Avoid water services as bearings can rust to shaft when idle. Test with bronze bearings.
6. Lead Babbit	32 to 300° F.	Excellent corrosion resistance to pH of 2. Good in mildly abrasive sevice. 80% Lead/3% Tin/17% Antimony.
7. Rubber w/Phenolic backing (Nitrite Butadiene or Neoprene)	32 to 150° F.	Use in abrasive water services. Bearings must be wet prior to start-up for TPL 50'. Do not use: For oily services, for stuffing box bushing, or with hydraulically balanced impellers. For services that are corrosive, backing material other than Phenolic must be specified.
8. Hardened Metals: Sprayed on stainless steel shell (Tungsten Carbide)	All temperatures All Gravities	Expensive alternate for abrasive services. Hardfaced surfaces typically in the range of Rc72. Other coatings are chromium oxide, tungsten carbide, colmonoy, etc. Consult factory for pricing and specific recommendation.

TECH-B-8 Self Priming Pump System Guidelines

Self-priming pumps are inherently designed to allow the pump to re-prime itself typically under lift conditions. These pumps are very effective to the end user in that they will eliminate the need for foot valves, vacuum and ejector pumps which can become clogged or be impractical to use for prolonged or remote operation. Although the pump itself is designed to accomplish this task, it is important to understand the principle of how self-priming is achieved so that the piping system can be designed so as not to conflict with this function.

A self-priming pump, by definition, is a pump which will clear its passages of air if it becomes air bound and resume delivery of the pumpage without outside attention. To accomplish this, a charge of

liquid sufficient to prime the pump must be retained in the casing (See Fig. A) or in an accessory priming chamber. When the pump starts, the rotating impeller creates a partial vacuum; air from the suction piping is then drawn into this vacuum and is entrained in the liquid drawn from the priming chamber. This air-liquid mixture is then pumped into the air separation chamber (within the casing) where the air is separated from the liquid with the air being expelled out the discharge piping (Fig. B) and the liquid returning to the priming chamber. This cycle is repeated until all of the air from the suction piping has been expelled and replaced by pumpage and the prime has been established (Fig. C).

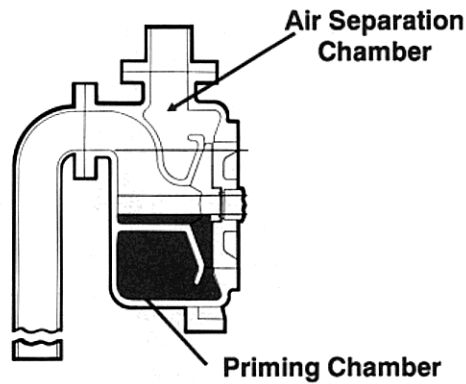


Fig. A

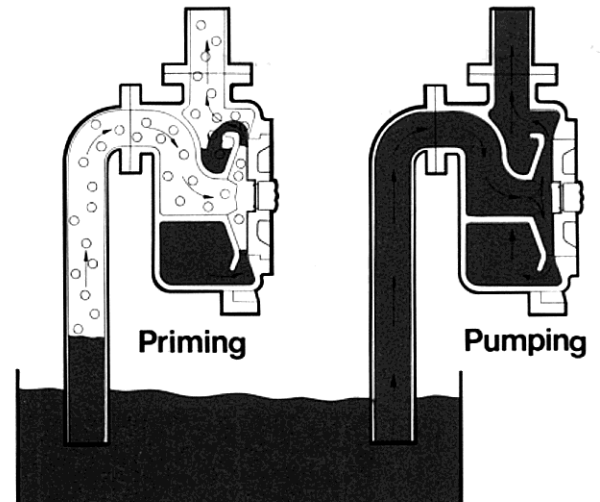


Fig. B

Fig. C

The following considerations should be made when designing a piping system for which a self-priming pump is to be used:

- Care should be exercised to insure that adequate liquid is retained in the priming chamber. For outdoor/remote installations a heating element may be required to prevent freezing. For dirty services a strainer may be required to keep solids from accumulating in the priming chamber, thus displacing priming liquid.
- The static lift and suction piping should be minimized to keep priming time to a minimum. Excessive priming time can cause liquid in the priming chamber to vaporize before prime is achieved.
- All connections in the suction piping should be leak-free as air could be sucked in, thus extending/compromising priming of the pump. (Pumps sealed with packing should be flushed to prevent air from being introduced.)
- A priming bypass line (See Fig. D) should be installed so that back pressure is not created in the discharge piping during priming which would prevent the pump from priming itself. (Self-priming pumps are not good air compressors!)
- The suction piping should be designed such that no high points are created where air can be trapped/accumulate which can prevent priming. Historically this has been problematic on top unloading of rail cars. (See Fig. E)

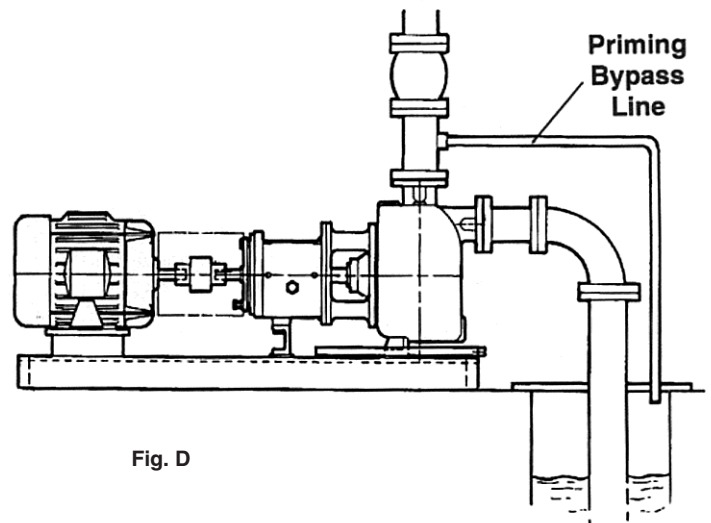


Fig. D

NOTE: Goulds Model 3796 self-priming process pump is outlined in Section 1F.

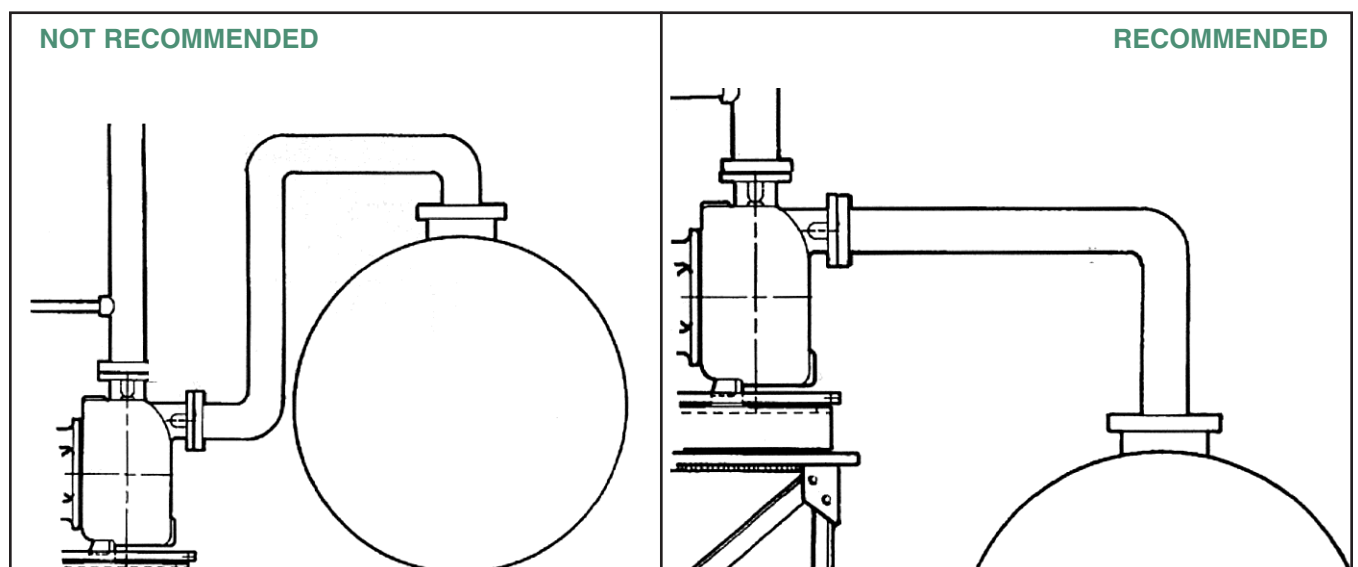


Fig. E Tank Car Unloading

TECH-B-9 Priming Time Calculations

Priming time data for each Model 3796 pump size and speed is displayed on the individual performance curves where priming time is plotted versus effective static lift for maximum, minimum and intermediate impeller diameters. This data is for suction piping of the same nominal diameter as the pump suction, i.e. 3" piping and 3" pump suction, and must be corrected for suction pipe diameters different from the pump suction and for suction pipe lengths greater than the effective static lift.

To calculate the total priming time for a given system:

1. Select the correct size and speed pump from the performance curve for the given rating.
2. Calculate the **NPSH Available** for the system. The available NPSH must be **equal to or greater than the NPSH Required** by the selected pump at the rating point.

$$NPSH_A = P - (L_s + V_p + h_f)$$

where: P = Pressure on surface of liquid in feet absolute

L_s = Maximum static lift in feet from free surface of the liquid to the centerline of the impeller.

V_p = Vapor pressure of the liquid at maximum pumping temperature in feet absolute.

h_f = Suction pipe friction loss in feet at the required capacity.

3. Determine the effective static lift.

$$L_{es} = L_s \times \text{Sp. Gr.}$$

where: L_{es} = Effective static lift in feet.

L_s = Maximum static lift in feet from free surface of the liquid to the centerline of the pump suction, or the highest point in the suction piping, whichever is greater.

Sp. Gr. = Specific gravity of the liquid.

4. Enter the priming time curve at the effective static lift calculated in Step 3. Proceed across to the impeller diameter selected for the specified rating and then downward to the bottom coordinate to determine the priming time (PT_{Les}) to achieve the given lift.
5. Insert the priming time from Step 4 into the following formula to calculate the total system priming time:

Priming Time - Seconds

$$PT_T = PT_{Les} \times \frac{SPL}{L_{es}} \times \left(\frac{D_p}{D_s} \right)^2$$

where: PT_T = Total system priming time.

PT_{Les} = Priming time in seconds for the effective static lift (Step 4.)

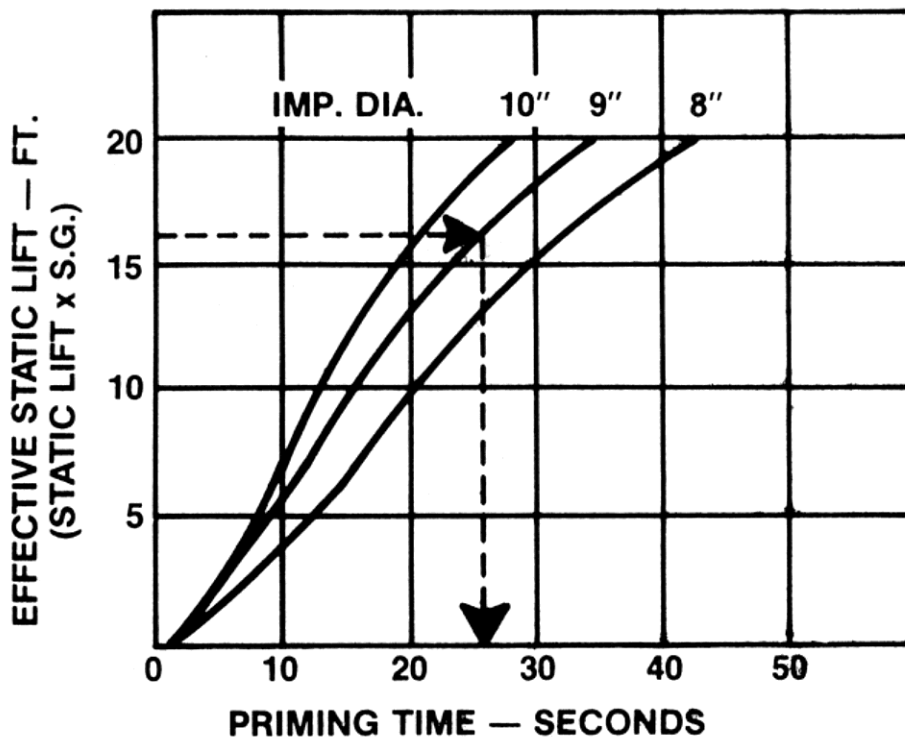
SPL = Total suction pipe length above the free surface of the liquid in feet.

L_{es} = Effective static lift.

D_p = Nominal pipe diameter.

D_s = Nominal pump suction diameter.

Sample Priming Time Curve





Section TECH-C

Water Data

TECH-C-1 Friction Loss for Water – Sched 40 Steel Pipe

U.S. Gallons per Minute	1/8 In. (0.269" I.D.)			1/4 In. (0.364" I.D.)			3/8 In. (0.493" I.D.)			1/2 In. (0.622" I.D.)			U.S. Gallons per Minute
	V (Ft./Sec.)	V ² 2g	h _f (Ft./100 ft.)	V	V ² 2g	h _f	V	V ² 2g	h _f	V	V ² 2g	h _f	
0.2	1.13	0.020	2.72										0.2
0.4	2.26	0.079	16.2	1.23	0.024	3.7							0.4
0.6	3.39	0.178	33.8	1.85	0.053	7.6	1.01	0.016	1.74				0.6
0.8	4.52	0.317	57.4	2.47	0.095	12.7	1.34	0.028	2.89				0.8
1.0	5.65	0.495	87.0	3.08	0.148	19.1	1.68	0.044	4.30	1.06	0.017	1.86	1.0
1.5	8.48	1.12	188	4.62	0.332	40.1	2.52	0.099	8.93	1.58	0.039	2.85	1.5
2.0	11.3	1.98	324	6.17	0.591	69.0	3.36	0.176	15.0	2.11	0.069	4.78	2.0
2.5				7.17	0.923	105	4.20	0.274	22.6	2.64	0.108	7.16	2.5
3.0				9.25	1.33	148	5.04	0.395	31.8	3.17	0.156	10.0	3.0
3.5				10.79	1.81	200	5.88	0.538	42.6	3.70	0.212	13.3	3.5
4.0				12.33	2.36	259	6.72	0.702	54.9	4.22	0.277	17.1	4.0
4.5				13.87	2.99	326	7.56	0.889	68.4	4.75	0.351	21.3	4.5
5				15.42	3.69	398	8.40	1.10	83.5	5.28	0.433	25.8	5
6							10.1	1.58	118	6.34	0.624	36.5	6
7							11.8	2.15	158	7.39	0.849	48.7	7
8							13.4	2.81	205	8.45	1.11	62.7	8
9							15.1	3.56	258	9.50	1.40	78.3	9
10							16.8	4.39	316	10.6	1.73	95.9	10
12										12.7	2.49	136	12
14										14.8	3.40	183	14

U.S. Gallons per Minute	3/4 In. (0.824" I.D.)			1 In. (1.049" I.D.)			1 1/4 In. (1.3880" I.D.)			1 1/2 In. (1.610" I.D.)			U.S. Gallons per Minute
	V	V ² 2g	h _f	V	V ² 2g	h _f	V	V ² 2g	h _f	V	V ² 2g	h _f	
4	2.41	0.090	4.21	1.48	0.034	1.29							4
5	3.01	0.141	6.32	1.86	0.053	1.93							5
6	3.61	0.203	8.87	2.23	0.077	2.68	1.29	0.026	0.70				6
7	4.21	0.276	11.8	2.60	0.105	3.56	1.50	0.035	0.93				7
8	4.81	0.360	15.0	2.97	0.137	4.54	1.72	0.046	1.18	1.26	0.025	0.56	8
9	5.42	0.456	18.8	3.34	0.173	5.65	1.93	0.058	1.46	1.42	0.031	0.69	9
10	6.02	0.563	23.0	3.71	0.214	6.86	2.15	0.071	1.77	1.58	0.039	0.83	10
12	7.22	0.810	32.6	4.45	0.308	9.62	2.57	0.103	2.48	1.89	0.056	1.16	12
14	8.42	1.10	43.5	5.20	0.420	12.8	3.00	0.140	3.28	2.21	0.076	1.53	14
16	9.63	1.44	56.3	5.94	0.548	16.5	3.43	0.183	4.20	2.52	0.099	1.96	16
18	10.8	1.82	70.3	6.68	0.694	20.6	3.86	0.232	5.22	2.84	0.125	2.42	18
20	12.0	2.25	86.1	7.42	0.857	25.1	4.29	0.286	6.34	3.15	0.154	2.94	20
25	15.1	3.54	134	9.29	1.34	37.4	5.37	0.448	9.66	3.94	0.241	4.50	25
30	18.1	5.06	187	11.1	1.93	54.6	6.44	0.644	13.6	4.73	0.347	6.26	30
35				13.0	2.62	73.3	7.52	0.879	18.5	5.52	0.473	8.38	35
40				14.8	3.43	95.0	8.58	1.14	23.5	6.30	0.618	10.8	40
45				16.7	4.33	119	9.66	1.45	29.5	7.10	0.783	13.5	45
50				18.6	5.35	146	10.7	1.79	36.0	7.88	0.965	16.4	50
60				22.3	7.71	209	12.9	2.57	51.0	9.46	1.39	23.2	60
70				26.0	10.5	283	15.0	3.50	68.8	11.0	1.89	31.3	70
80							17.2	4.58	89.2	12.6	2.47	40.5	80
90							19.3	5.79	112	14.2	3.13	51.0	90
100							21.5	7.15	138	15.8	3.86	62.2	100
120							25.7	10.3	197	18.9	5.56	88.3	120
140										22.1	7.56	119	140

U.S. Gallons per Minute	2 In. (2.067" I.D.)			2½ In. (2.469" I.D.)			3 In. (3.068" I.D.)			3½ In. (3.548" I.D.)			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
30	2.87	0.128	1.82	2.01	0.063	0.75							30
35	3.35	0.174	2.42	2.35	0.085	1.00							35
40	3.82	0.227	3.10	2.68	0.112	1.28							40
50	4.78	0.355	4.67	3.35	0.174	1.94	2.17	0.073	0.66				50
60	5.74	0.511	6.59	4.02	0.251	2.72	2.60	0.105	0.92	1.95	0.059	0.45	60
80	7.65	0.909	11.4	5.36	0.447	4.66	3.47	0.187	1.57	2.60	0.105	0.77	80
100	9.56	1.42	17.4	6.70	0.698	7.11	4.34	0.293	2.39	3.25	0.164	1.17	100
120	11.5	2.05	24.7	8.04	1.00	10.0	5.21	0.421	3.37	3.89	0.236	1.64	120
140	13.4	2.78	33.2	9.38	1.37	13.5	6.08	0.574	4.51	4.54	0.321	2.18	140
160	15.3	3.64	43.0	10.7	1.79	17.4	6.94	0.749	5.81	5.19	0.419	2.80	160
180	17.2	4.60	54.1	12.1	2.26	21.9	7.81	0.948	7.28	5.84	0.530	3.50	180
200	19.1	5.68	66.3	13.4	2.79	26.7	8.68	1.17	8.90	6.49	0.655	4.27	200
220	21.0	6.88	80.0	14.7	3.38	32.2	9.55	1.42	10.7	7.14	0.792	5.12	220
240	22.9	8.18	95.0	16.1	4.02	38.1	10.4	1.69	12.6	7.79	0.943	6.04	240
260	24.9	9.60	111	17.4	4.72	44.5	11.3	1.98	14.7	8.44	1.11	7.04	260
280	26.8	11.1	128	18.8	5.47	51.3	12.2	2.29	16.9	9.09	1.28	8.11	280
300	28.7	12.8	146	20.1	6.28	58.5	13.0	2.63	19.2	9.74	1.47	9.26	300
350				23.5	8.55	79.2	15.2	3.57	26.3	11.3	2.00	12.4	350
400				26.8	11.2	103	17.4	4.68	33.9	13.0	2.62	16.2	400
500				33.5	17.4	160	21.7	7.32	52.5	16.2	4.09	25.0	500
600							26.0	10.5	74.8	19.5	5.89	35.6	600
700							30.4	14.3	101	22.7	8.02	48.0	700
800							34.7	18.7	131	26.0	10.5	62.3	800
1000										32.5	16.44	96.4	1000

U.S. Gallons per Minute	4 In. (4.026" I.D.)			5 In. (5.047" I.D.)			6 In. (6.065" I.D.)			8 In. (7.981" I.D.)			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
140	3.53	0.193	1.16	2.25	0.078	0.38							140
160	4.03	0.253	1.49	2.57	0.102	0.49							160
180	4.54	0.320	1.86	2.89	0.129	0.61							180
200	5.04	0.395	2.27	3.21	0.160	0.74	2.22	0.077	0.30				200
240	6.05	0.569	3.21	3.85	0.230	1.03	2.66	0.110	0.42				240
280	7.06	0.774	4.30	4.49	0.313	1.38	3.11	0.150	0.56				280
320	8.06	1.01	5.51	5.13	0.409	1.78	3.55	0.196	0.72				320
360	9.07	1.28	6.92	5.77	0.518	2.22	4.00	0.240	0.90				360
400	10.1	1.58	8.47	6.41	0.639	2.72	4.44	0.307	1.09	2.57	0.102	0.28	400
450	11.3	2.00	10.5	7.23	0.811	3.42	5.00	0.388	1.37	2.89	0.129	0.35	450
500	12.6	2.47	13.0	8.02	0.999	4.16	5.55	0.479	1.66	3.21	0.160	0.42	500
600	15.1	3.55	18.6	9.62	1.44	5.88	6.66	0.690	2.34	3.85	0.230	0.60	600
700	17.6	4.84	25.0	11.2	1.96	7.93	7.77	0.939	3.13	4.49	0.313	0.80	700
800	20.2	6.32	32.4	12.8	2.56	10.2	8.88	1.23	4.03	5.13	0.409	1.02	800
900	22.7	8.00	40.8	14.4	3.24	12.9	9.99	1.55	5.05	5.77	0.518	1.27	900
1000	25.2	9.87	50.2	16.0	4.00	15.8	11.1	1.92	6.17	6.41	0.639	1.56	1000
1200	30.2	14.2	72.0	19.2	5.76	22.5	13.3	2.76	8.76	7.70	0.920	2.20	1200
1400	35.3	19.3	97.6	22.5	7.83	30.4	15.5	3.76	11.8	8.98	1.25	2.95	1400
1600				25.7	10.2	39.5	17.8	4.91	15.4	10.3	1.64	3.82	1600
1800				28.8	12.9	49.7	20.0	6.21	19.4	11.5	2.07	4.79	1800
2000				32.1	16.0	61.0	22.2	7.67	23.8	12.8	2.56	5.86	2000
2400							26.6	11.0	34.2	15.4	3.68	8.31	2400
2800							31.1	15.0	46.1	18.0	5.01	11.2	2800
3200							35.5	19.6	59.9	20.5	6.55	14.5	3200
3600										23.1	8.28	18.4	3600
4000										25.7	10.2	22.6	4000

U.S. Gallons per Minute	10 In. (10.020" I.D.)			12 In. (11.938" I.D.)			14 In. (13.124" I.D.)			16 In. (15.000" I.D.)			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
800	3.25	0.165	0.328										800
900	3.66	0.208	0.410	2.58	0.103	0.173							900
1000	4.07	0.257	0.500	2.87	0.128	0.210	2.37	0.087	0.131				1000
1200	4.88	0.370	0.703	3.44	0.184	0.296	2.85	0.126	0.185				1200
1400	5.70	0.504	0.940	4.01	0.250	0.395	3.32	0.171	0.247				1400
1600	6.51	0.659	1.21	4.59	0.327	0.609	3.79	0.224	0.317	2.90	0.131	0.163	1600
1800	7.32	0.834	1.52	5.16	0.414	0.636	4.27	0.283	0.395	3.27	0.166	0.203	1800
2000	8.14	1.03	1.86	5.73	0.511	0.776	4.74	0.349	0.483	3.63	0.205	0.248	2000
2500	10.2	1.62	2.86	7.17	0.799	1.19	5.93	0.546	0.738	4.54	0.320	0.377	2500
3000	12.2	2.32	4.06	8.60	1.15	1.68	7.11	0.786	1.04	5.45	0.461	0.535	3000
3500	14.2	3.13	5.46	10.0	1.55	2.25	8.30	1.07	1.40	6.35	0.627	0.718	3500
4000	16.3	4.12	7.07	11.5	2.04	2.92	9.48	1.40	1.81	7.26	0.820	0.921	4000
4500	18.3	5.21	8.88	12.9	2.59	3.65	10.7	1.77	2.27	8.17	1.04	1.15	4500
5000	20.3	6.43	10.9	14.3	3.19	4.47	11.9	2.18	2.78	9.08	1.28	1.41	5000
6000	24.4	9.26	15.6	17.2	4.60	6.39	14.2	3.14	3.95	10.9	1.84	2.01	6000
7000	28.5	12.6	21.1	20.1	6.26	8.63	16.6	4.28	5.32	12.7	2.51	2.69	7000
8000	32.5	16.5	27.5	22.9	8.17	11.2	19.0	5.59	6.90	14.5	3.28	3.498	8000
9000	36.6	20.8	34.6	25.8	10.3	14.1	21.3	7.08	8.7	16.3	4.15	4.38	9000
10,000				28.7	12.8	17.4	23.7	8.74	10.7	18.2	5.12	5.38	10,000
12,000				34.4	18.3	24.8	28.5	12.6	15.2	21.8	7.38	7.69	12,000
14,000				40.1	25.0	33.5	33.2	17.1	20.7	25.4	10.0	10.4	14,000
16,000							37.9	22.4	26.8	29.0	13.1	13.5	16,000
18,000							42.7	28.3	33.9	32.7	16.6	17.2	18,000
20,000										36.3	20.5	21.2	20,000

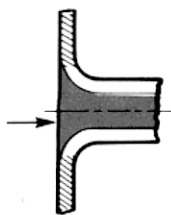
U.S. Gallons per Minute	18 In. (16.876" I.D.)			20 In. (18.812" I.D.)			24 In. (22.624" I.D.)			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
2000	2.87	0.128	0.139							2000
3000	4.30	0.288	0.297	3.46	0.186	0.174				3000
4000	5.74	0.512	0.511	4.62	0.331	0.298	3.19	0.158	0.120	4000
5000	7.17	0.799	0.781	5.77	0.517	0.455	3.99	0.247	0.181	5000
6000	8.61	1.15	1.11	6.92	0.745	0.645	4.79	0.356	0.257	6000
8000	11.5	2.05	1.93	9.23	1.32	1.11	6.38	0.633	0.441	8000
10,000	14.3	3.20	2.97	11.5	2.07	.70	7.98	0.989	0.671	10,000
12,000	17.2	4.60	4.21	13.8	2.98	2.44	9.58	1.42	0.959	12,000
14,000	20.1	6.27	5.69	16.2	4.06	3.29	11.2	1.94	1.29	14,000
16,000	22.9	8.19	7.41	18.5	5.30	4.26	12.8	2.53	1.67	16,000
18,000	25.8	10.4	9.33	20.8	6.71	5.35	14.4	3.21	2.10	18,000
20,000	28.7	12.8	11.5	23.1	8.28	6.56	16.0	3.96	2.58	20,000
22,000	31.6	15.5	13.9	25.4	10.0	7.91	17.6	4.79	3.10	22,000
24,000	34.4	18.4	16.5	27.7	11.9	9.39	19.2	5.70	3.67	24,000
26,000	37.3	21.6	19.2	30.0	14.0	11.0	20.7	6.69	4.29	26,000
28,000	40.2	25.1	22.2	32.3	16.2	12.7	22.3	7.76	4.96	28,000
30,000	43.0	28.8	25.5	34.6	18.6	14.6	23.9	8.91	5.68	30,000
34,000				39.2	23.9	18.7	27.1	11.4	7.22	34,000
38,000				43.9	29.9	23.2	30.3	14.3	9.00	38,000
42,000							33.5	17.5	11.0	42,000
46,000							36.7	20.9	13.2	46,000
50,000							39.9	24.7	15.5	50,000

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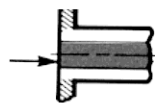
U.S. Gallons per Minute	30 In.			36 In.			42 In.			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
5,000	2.43	0.0917	0.0535							5,000
6,000	2.91	0.132	0.0750							6,000
7,000	3.40	0.180	0.100							7,000
8,000	3.89	0.235	0.129	2.52	0.0988	0.0442				8,000
9,000	4.37	0.297	0.161	2.84	0.125	0.0551				9,000
10,000	4.86	0.367	0.196	3.15	0.154	0.0670				10,000
12,000	5.83	0.528	0.277	3.78	0.222	0.0942	2.78	0.120	0.0441	12,000
14,000	6.80	0.719	0.371	4.41	0.303	0.126	3.24	0.163	0.0591	14,000
16,000	7.77	0.939	0.478	5.04	0.395	0.162	3.71	0.213	0.0758	16,000
18,000	8.74	1.19	0.598	5.67	0.500	0.203	4.17	0.270	0.0944	18,000
20,000	9.71	1.47	0.732	6.30	0.618	0.248	4.63	0.333	0.115	20,000
25,000	12.1	2.29	1.13	7.88	0.965	0.378	5.79	0.521	0.176	25,000
30,000	14.6	3.30	1.61	9.46	1.39	0.540	6.95	0.750	0.250	30,000
35,000	17.0	4.49	2.17	11.03	1.89	0.724	8.11	1.02	0.334	35,000
40,000	19.4	5.87	2.83	12.6	2.47	0.941	9.26	1.33	0.433	40,000
45,000	21.9	7.42	3.56	14.1	3.13	1.18	10.42	1.69	0.545	45,000
50,000	24.3	9.17	4.38	15.8	3.86	1.45	11.6	2.08	0.668	50,000
60,000	29.1	13.2	6.23	18.9	5.56	2.07	13.9	3.00	0.946	60,000
70,000	34.0	18.0	8.43	22.1	7.56	2.81	16.2	4.08	1.27	70,000
80,000	38.9	23.5	11.0	25.2	9.88	3.66	18.5	5.33	1.66	80,000
90,000				28.4	12.5	4.59	20.8	6.75	2.08	90,000
100,000				31.5	15.4	5.64	23.2	8.33	2.57	100,000
120,000				37.8	22.2	8.05	27.8	12.0	3.67	120,000
140,000							32.4	16.3	4.98	140,000
160,000							37.1	21.3	6.46	160,000
180,000							41.7	27.0	8.12	180,000

U.S. Gallons per Minute	48 In.			54 In.			60 In.			U.S. Gallons per Minute
	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	V	$\frac{V^2}{2g}$	h_f	
16,000	2.84	0.125	0.0391							16,000
18,000	3.19	0.158	0.0488							18,000
20,000	3.55	0.195	0.0598	2.80	0.122	0.0333				20,000
25,000	4.43	0.305	0.0910	3.50	0.191	0.0504	2.84	0.125	0.0301	25,000
30,000	5.32	0.440	0.128	4.20	0.274	0.0713	3.40	0.180	0.0424	30,000
35,000	6.21	0.598	0.172	4.90	0.374	0.0958	3.97	0.245	0.0567	35,000
40,000	7.09	0.782	0.222	5.60	0.488	0.124	4.54	0.320	0.0730	40,000
45,000	7.98	0.989	0.278	6.30	0.618	0.155	5.11	0.405	0.0916	45,000
50,000	8.87	1.221	0.341	7.00	0.762	0.189	5.67	0.500	0.112	50,000
60,000	10.64	1.76	0.484	8.40	1.098	0.267	6.81	0.720	0.158	60,000
70,000	12.4	2.39	0.652	9.81	1.49	0.358	7.94	0.980	0.213	70,000
80,000	14.2	3.13	0.849	11.21	1.95	0.465	9.08	1.28	0.275	80,000
90,000	16.0	3.96	1.06	12.6	2.47	0.586	10.21	1.62	0.344	90,000
100,000	17.7	4.89	1.30	14.0	3.05	0.715	11.3	2.00	0.420	100,000
120,000	21.3	7.03	1.87	16.8	4.39	1.02	13.6	2.88	0.600	120,000
140,000	24.8	9.57	2.51	19.6	5.98	1.38	15.9	3.92	0.806	140,000
160,000	28.4	12.5	3.26	22.4	7.81	1.80	18.2	5.12	1.04	160,000
180,000	31.9	15.8	4.11	25.2	9.88	2.26	20.4	6.48	1.32	180,000
200,000	35.5	19.5	5.05	28.0	12.2	2.77	22.7	8.00	1.62	200,000
250,000				35.0	19.1	4.32	28.4	12.5	2.52	250,000
300,000				42.0	27.4	6.19	34.0	18.0	3.60	300,000
350,000							39.7	24.5	4.88	350,000

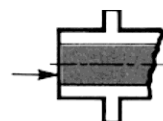
TECH-C-2 Resistance Coefficients for Valves and Fittings



BELL-MOUTH
INLET OR REDUCER
 $K = 0.05$

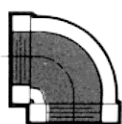


SQUARE EDGED INLET
 $K = 0.5$

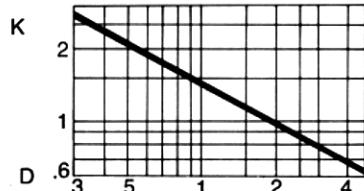


INWARD PROJECTING PIPE
 $K = 1.0$

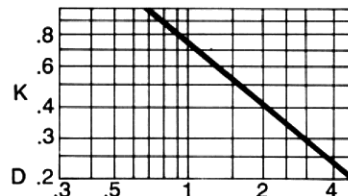
NOTE: K DECREASES WITH
INCREASING WALL THICKNESS OF
PIPE AND ROUNDING OF EDGES



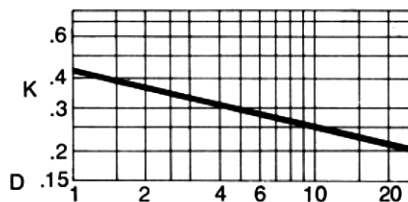
REGULAR
SCREWED
90° ELL.



LONG
RADIUS
SCREWED
90° ELL.



REGULAR
FLANGED
90° ELL.



LONG
RADIUS
FLANGED
90° ELL.

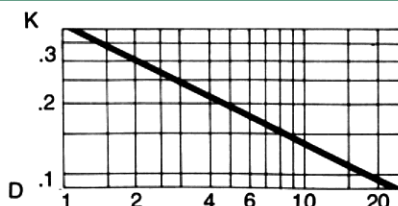


Chart 1

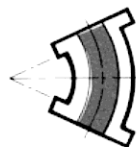
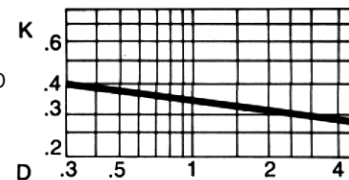
Where:

h = Frictional Resistance in Feet of Liquid

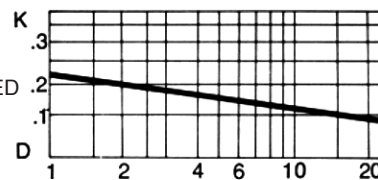
V = Average Velocity in Feet/Second in a Pipe of Corresponding Diameter



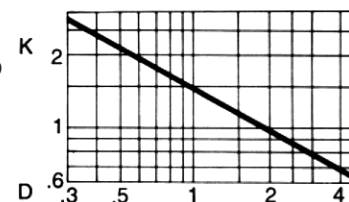
REGULAR
SCREWED
45° ELL.



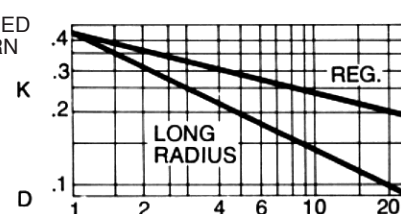
LONG
RADIUS
FLANGED
45° ELL.



SCREWED
RETURN
BEND

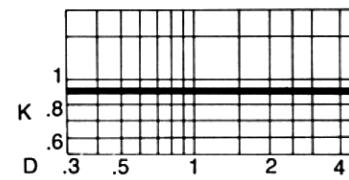


FLANGED
RETURN
BEND

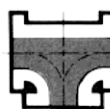


SCREWED
TEE

LINE
FLOW

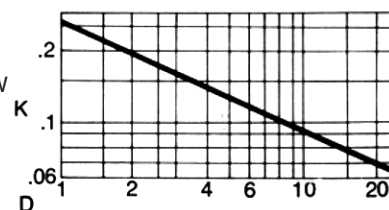


BRANCH
FLOW

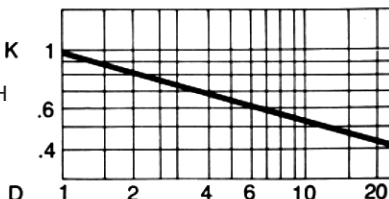


FLANGED
TEE

LINE
FLOW



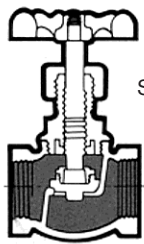
BRANCH
FLOW



$$h = K \frac{V^2}{2g}$$

$g = 32.17$ Feet/Second/Second

K = Resistance Coefficient For Valve or Fitting

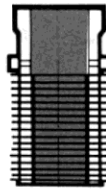


SCREWED

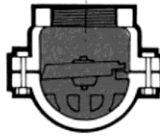
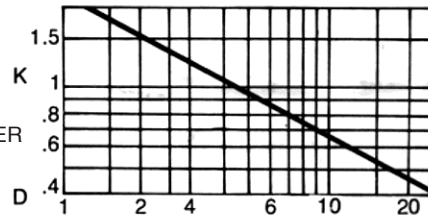


GLOBE VALVE

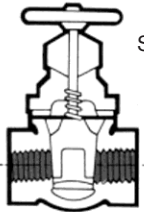
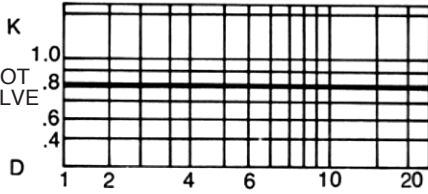
FLANGED



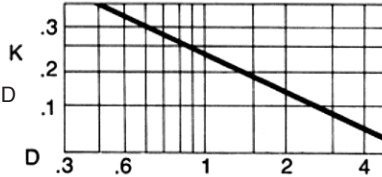
BASKET STRAINER



FOOT VALVE

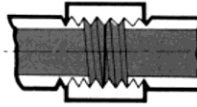
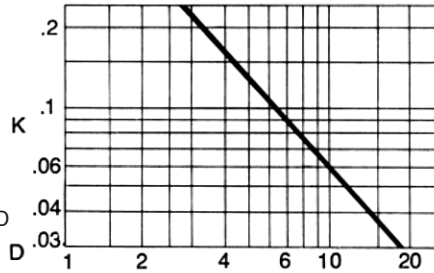


SCREWED

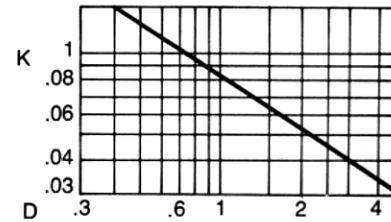


GATE VALVE

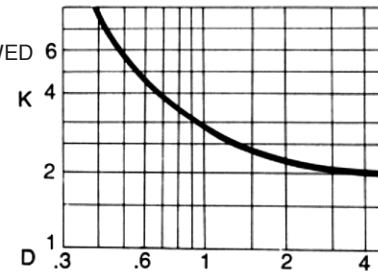
FLANGED



COUPLINGS AND UNIONS

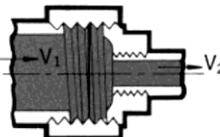
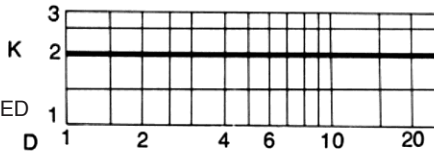


SCREWED



SWING CHECK VALVE

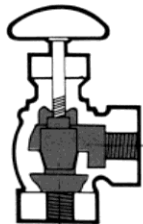
FLANGED



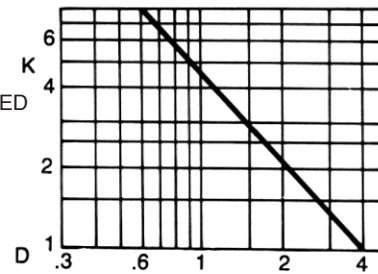
REDUCING BUSHING AND COUPLING

$$h = K \frac{V_2^2}{2g}$$

USED AS A REDUCER $K = 0.05 - 2.0$
SEE ALSO FIG. 3
USED AS INCREASER LOSS IS UP TO 40% MORE THAN THAT CAUSED BY A SUDDEN ENLARGEMENT

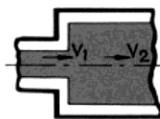
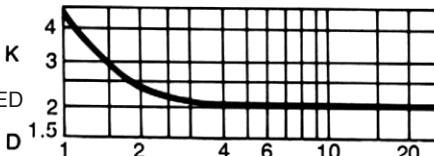


SCREWED



ANGLE VALVE

FLANGED



SUDDEN ENLARGEMENT

$$h = \frac{(V_1 - V_2)^2}{2g} \text{ FEET OF FLUID}$$

SEE ALSO EQUATION(5)

IF $A_2 = \infty$ SO THAT $V_2 = 0$

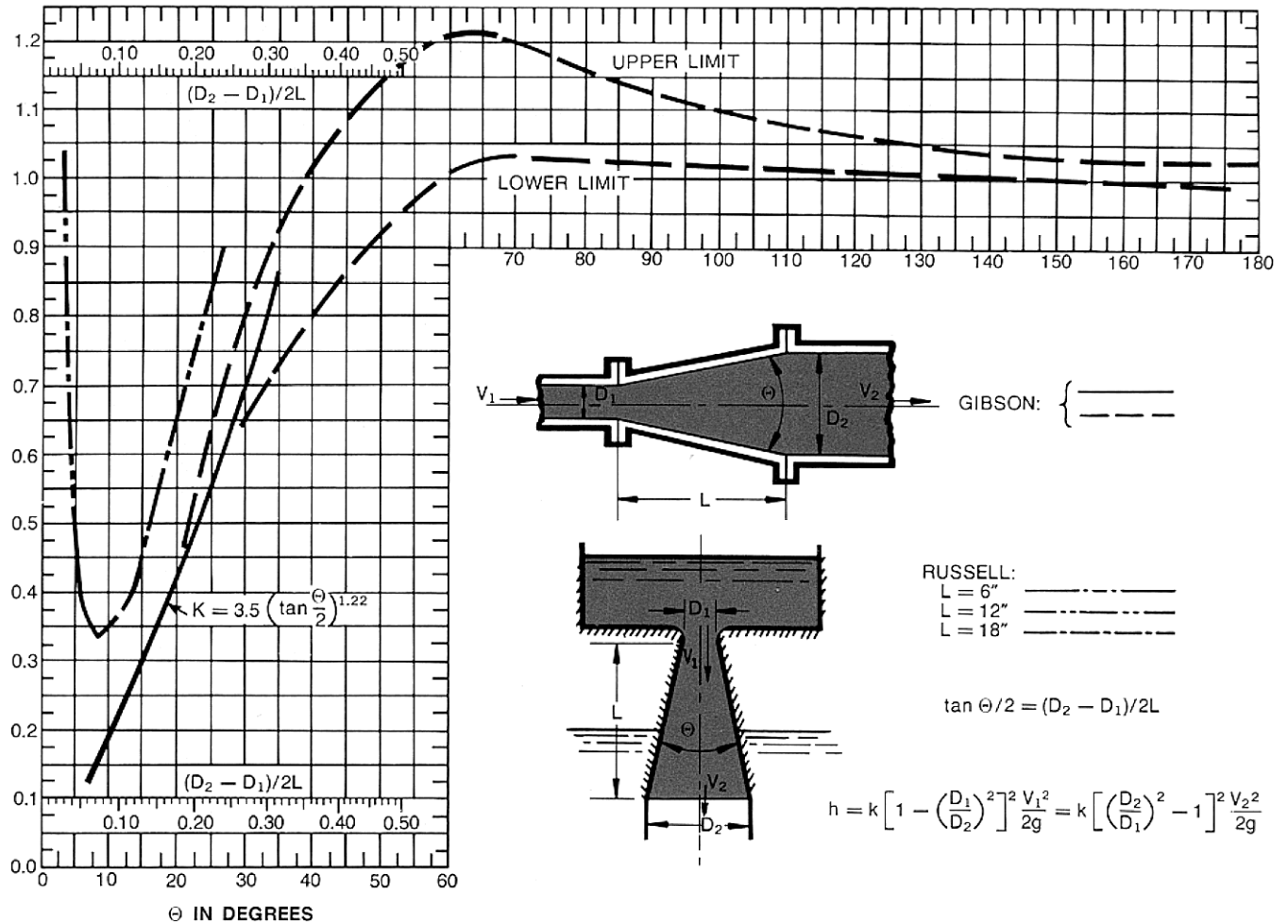
$$h = \frac{V_1^2}{2g} \text{ FEET OF FLUID}$$

$$h = K \frac{V_2^2}{2g}$$

Chart 2

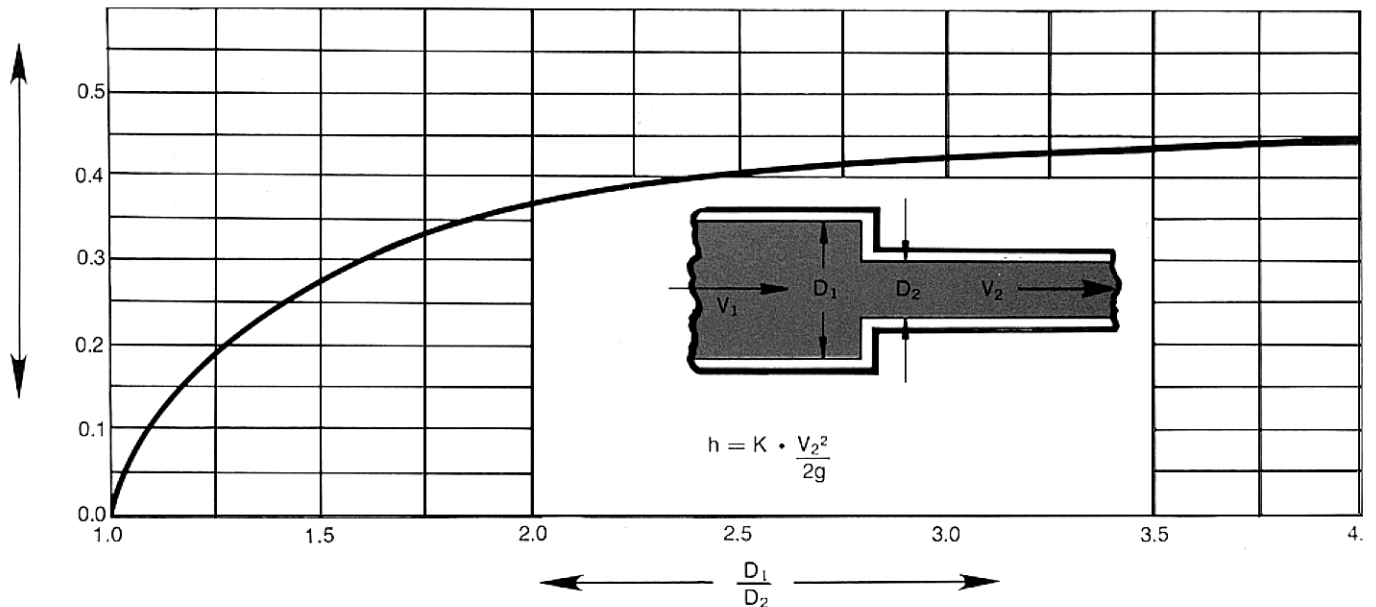
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TECH-C-3 Resistance Coefficients for Increasers and Diffusers



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TECH-C-4 Resistance Coefficients for Reducers



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TECH-C-5 Properties of Water at Various Temperatures from 32° to 705.4°F

Temp. F	Temp. C	SPECIFIC GRAVITY 60 F Reference	Wt. in Lb/Cu Ft	Vapor Pressure Psi Abs	Vapor Pressure* Feet Abs. (At Temp.)
32	0	1.002	62.42	0.0885	0.204
40	4.4	1.001	62.42	0.1217	0.281
45	7.2	1.001	62.40	0.1471	0.340
50	10.0	1.001	62.38	0.1781	0.411
55	12.8	1.000	62.36	0.2141	0.494
60	15.6	1.000	62.34	0.2653	0.591
65	18.3	.999	62.31	0.3056	0.706
70	21.1	.999	62.27	0.3631	0.839
75	23.9	.998	62.24	0.4298	0.994
80	26.7	.998	62.19	0.5069	1.172
85	29.4	.997	62.16	0.5959	1.379
90	32.2	.996	62.11	0.9682	1.167
95	35.0	.995	62.06	0.8153	1.890
100	37.8	.994	62.00	0.9492	2.203
110	43.3	.992	61.84	1.275	2.965
120	48.9	.990	61.73	1.692	3.943
130	54.4	.987	61.54	2.223	5.196
140	60.0	.985	61.39	2.889	6.766
150	65.5	.982	61.20	3.718	8.735
160	71.1	.979	61.01	4.741	11.172
170	76.7	.975	60.79	5.992	14.178
180	82.2	.972	60.57	7.510	17.825
190	87.7	.968	60.35	9.339	22.257
200	93.3	.966	60.13	11.526	27.584
212	100.0	.959	59.81	14.696	35.353
220	104.4	.956	59.63	17.186	41.343
240	115.6	.948	59.10	24.97	60.77
260	126.7	.939	58.51	35.43	87.05
280	137.8	.929	58.00	49.20	122.18
300	148.9	.919	57.31	67.01	168.22
320	160.0	.909	56.66	89.66	227.55
340	171.1	.898	55.96	118.01	303.17
360	182.2	.886	55.22	153.04	398.49
380	193.3	.874	54.47	195.77	516.75
400	204.4	.860	53.65	247.31	663.42
420	215.6	.847	52.80	308.83	841.17
440	226.7	.833	51.92	381.59	1056.8
460	237.8	.818	51.02	466.9	1317.8
480	248.9	.802	50.00	566.1	1628.4
500	260.0	.786	49.02	680.8	1998.2
520	271.1	.766	47.85	812.4	2446.7
540	282.2	.747	46.51	962.5	2972.5
560	293.3	.727	45.3	1133.1	3595.7
580	304.4	.704	43.9	1325.8	4345.
600	315.6	.679	42.3	1524.9	5242.
620	326.7	.650	40.5	1786.6	6341.
640	337.8	.618	38.5	2059.7	7689.
660	348.9	.577	36.0	2365.4	9458.
680	360.0	.526	32.8	2708.1	11878.
700	371.1	.435	27.1	3039.7	16407.
705.4	374.1	.319	19.9	3206.2	23187.

* Vapor pressure in feet of wate (Abs.) Converted from PSIA using sp. gr. at temperature.

TECH-C-6 Atmospheric Pressure, Barometer Reading and Boiling Point of Water at Various Altitudes

Altitude		Barometric Reading		Atmos. Pressure		Boiling Pt. Of Water °F
Feet	Meters	In. Hg.	Mm. Hg.	psia	Ft. Water	
— 1000	— 304.8	31.0	788	15.2	35.2	213.8
— 500	— 152.4	30.5	775	15.0	34.6	212.9
0	0.0	29.9	760	14.7	33.9	212.0
+ 500	+ 152.4	29.4	747	14.4	33.3	211.1
+ 1000	304.8	28.9	734	14.2	32.8	210.2
1500	457.2	28.3	719	13.9	32.1	209.3
2000	609.6	27.8	706	13.7	31.5	208.4
2500	762.0	27.3	694	13.4	31.0	207.4
3000	914.4	26.8	681	13.2	30.4	206.5
3500	1066.8	26.3	668	12.9	29.8	205.6
4000	1219.2	25.8	655	12.7	29.2	204.7
4500	1371.6	25.4	645	12.4	28.8	203.8
5000	1524.0	24.9	633	12.2	28.2	202.9
5500	1676.4	24.4	620	12.0	27.6	201.9
6000	1828.8	24.0	610	11.8	27.2	201.0
6500	1981.2	23.5	597	11.5	26.7	200.1
7000	2133.6	23.1	587	11.3	26.2	199.2
7500	2286.0	22.7	577	11.1	25.7	198.3
8000	2438.4	22.2	564	10.9	25.2	197.4
8500	2590.8	21.8	554	10.7	24.7	196.5
9000	2743.2	21.4	544	10.5	24.3	195.5
9500	2895.6	21.0	533	10.3	23.8	194.6
10000	3048.0	20.6	523	10.1	23.4	193.7
15000	4572.0	16.9	429	8.3	19.2	184.0

TECH-C-7 Saturation: Temperatures

Steam Data

Temp. F t	Abs. press. Lb Sq. In. p	Specific Volume			Enthalpy			Entropy			Temp F t
		Sat. Liquid v _f	Evap v _{fg}	Sat. Vapor v _g	Sat. Liquid h _f	Evap h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Sfg s _{fg}	Sat Vapor s _g	
32	0.08854	0.01602	3306	3306	0.00	1075.8	1075.8	0.0000	2.1877	2.1877	32
35	0.09995	0.01602	2947	2947	3.02	1074.1	1077.1	0.0061	2.1709	2.1770	35
40	0.12170	0.01602	2444	2444	8.05	1071.3	1079.3	0.0162	2.1435	2.1597	40
45	0.14752	0.01602	2036.4	2036.4	13.06	1068.4	1081.5	0.0262	2.1167	2.1429	45
50	0.17811	0.01603	1703.2	1703.2	18.07	1065.6	1083.7	0.0361	2.0903	2.1264	50
60	0.2563	0.01604	1206.6	1206.7	28.06	1059.9	1088.0	0.0555	2.0393	2.0948	60
70	0.3631	0.01606	867.8	867.9	38.04	1054.3	1092.3	0.0745	1.9902	2.0647	70
80	0.5069	0.01608	633.1	633.1	48.02	1048.6	1096.6	0.0932	1.9428	2.0360	80
90	0.6982	0.01610	468.0	468.0	57.99	1042.9	1100.9	0.1115	1.8972	2.0087	90
100	0.9492	0.01613	350.3	350.4	67.97	1037.2	1105.2	0.1295	1.8531	1.9826	100
110	1.2748	0.01617	265.3	265.4	77.94	1031.6	1109.5	0.1471	1.8106	1.9577	110
120	1.6924	0.01620	203.25	203.27	87.92	1025.8	1113.7	0.1645	1.7694	1.9339	120
130	2.2225	0.01625	157.32	157.34	97.90	1020.0	1117.9	0.1816	1.7296	1.9112	130
140	2.8886	0.01629	122.99	123.01	107.89	1014.1	1122.0	0.1984	1.6910	1.8894	140
150	3.718	0.01634	97.06	97.07	117.89	1008.2	1126.1	0.2149	1.6537	1.8685	150
160	4.741	0.01639	77.27	77.29	127.89	1002.3	1130.2	0.2311	1.6174	1.8485	160
170	5.992	0.01645	62.04	62.06	137.90	996.3	1134.2	0.2472	1.5822	1.8293	170
180	7.510	0.01651	50.21	50.23	147.92	990.2	1138.1	0.2630	1.5480	1.8109	180
190	9.339	0.01657	40.94	40.96	157.95	984.1	1142.0	0.2785	1.5147	1.7932	190
200	11.526	0.01663	33.62	33.64	167.99	977.9	1145.9	0.2938	1.4824	1.7762	200
210	14.123	0.01670	27.80	27.82	178.05	971.6	1149.7	0.3090	1.4508	1.7598	210
212	14.696	0.01672	26.78	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	212
220	17.186	0.01677	23.13	23.15	188.13	965.2	1153.4	0.3239	1.4201	1.7440	220
230	20.780	0.01684	19.365	19.382	198.23	958.8	1157.0	0.3387	1.3901	1.7288	230
240	24.969	0.01692	16.306	16.323	208.34	952.2	1160.5	0.3531	1.3609	1.7140	240
250	29.825	0.01700	13.804	13.821	218.48	945.5	1164.0	0.3675	1.3323	1.6998	250
260	35.429	0.01709	11.746	11.763	228.64	938.7	1167.3	0.3817	1.3043	1.6860	260
270	41.858	0.01717	10.044	10.061	238.84	931.8	1170.6	0.3958	1.2769	1.6727	270
280	49.203	0.01726	8.628	8.645	249.06	924.7	1173.8	0.4096	1.2501	1.6597	280
290	57.556	0.01735	7.444	7.461	259.31	917.5	1176.8	0.4234	1.2238	1.6472	290
300	67.013	0.01745	6.449	6.446	269.59	910.1	1179.7	0.4369	1.1980	1.6350	300
320	89.66	0.01765	4.896	4.914	290.28	894.9	1185.2	0.4637	1.1478	1.6115	320
340	118.01	0.01787	3.770	3.788	311.13	879.0	1190.1	0.4900	1.0992	1.5891	340
360	153.04	0.01811	2.939	2.957	332.18	862.2	1194.4	0.5158	1.0519	1.5677	360
380	195.77	0.01836	2.317	2.335	353.45	844.6	1198.1	0.5413	1.0059	1.5471	380
400	247.31	0.01864	1.8447	1.8633	374.97	826.0	1201.0	0.5664	0.9608	1.5272	400
420	308.83	0.01894	1.4811	1.5000	396.77	806.3	1203.1	0.5912	0.9166	1.5078	420
440	381.59	0.01926	1.1979	1.2171	418.90	785.4	1204.3	0.6158	0.8730	1.4887	440
460	466.9	0.0196	0.9748	0.9944	441.4	763.2	1204.6	0.6402	0.8298	1.4700	460
480	566.1	0.0200	0.7972	0.8172	464.4	739.4	1203.7	0.6645	0.7868	1.4513	480
500	680.8	0.0204	0.6545	0.6749	487.8	713.9	1201.7	0.6887	0.7438	1.4325	500
520	812.4	0.0209	0.5385	0.5594	511.9	686.4	1198.2	0.7130	0.7006	1.4136	520
540	962.5	0.0215	0.4434	0.4649	536.6	656.6	1193.2	0.7374	0.6568	1.3942	540
560	1133.1	0.0221	0.3647	0.3868	562.2	624.2	1186.4	0.7621	0.6121	1.3742	560
580	1325.8	0.0228	0.2989	0.3217	588.9	588.4	1177.3	0.7872	0.5659	1.3532	580
600	1542.9	0.0236	0.2432	0.2668	617.0	548.5	1165.5	0.8131	0.5176	1.3307	600
620	1786.6	0.0247	0.1955	0.2201	646.7	503.6	1150.3	0.8398	0.4664	1.3062	620
640	2059.7	0.0260	0.1538	0.1798	678.6	452.0	1130.5	0.8679	0.4110	1.2789	640
660	2365.4	0.0278	0.1165	0.1442	714.2	390.2	1104.4	0.8987	0.3485	1.2472	660
680	2708.1	0.0305	0.0810	0.1115	757.3	309.9	1067.2	0.9351	0.2719	1.2071	680
700	3093.7	0.0369	0.0392	0.0761	823.3	172.1	995.4	0.9905	0.1484	1.1389	700
705.4	3206.2	0.0503	0	0.0503	902.7	0	902.7	1.0680	0	1.0580	705.4

TECH-C-7 Saturation: Pressures

Steam Data

Abs. press. Lb Sq. In. p	Temp. Liquid t	Specific Volume		Enthalpy			Entropy			Internal Energy			Abs. press. Lb Sq. In. p
		Sat. Vapor v _f	Sat. Liquid v _g	Sat. Liquid h _r	Evap h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap s _{fg}	Sat Vapor s _g	Sat. Liquid u _f	Evap u _{fg}	Sat Vapor u _g	
1.0	101.74	0.01614	333.6	69.70	1036.3	1106.0	0.1326	1.8456	1.9782	69.70	974.6	1044.2	1.0
2.0	126.08	0.01623	173.73	93.99	1022.2	1116.2	0.1749	1.7451	1.9200	93.98	957.9	1051.9	2.0
3.0	141.48	0.01630	118.71	109.37	1031.2	1122.6	0.2008	1.6855	1.8863	109.36	947.3	1056.7	3.0
4.0	152.97	0.01636	90.63	120.86	1006.4	1127.3	0.2198	1.6427	1.8625	120.85	939.3	1060.2	4.0
5.0	162.24	0.01640	73.52	130.13	1001.0	1131.1	0.2347	1.6094	1.8441	130.12	933.0	1063.1	5.0
6.0	170.06	0.01645	61.98	137.96	996.2	1134.2	0.2472	1.5820	1.8292	137.94	927.5	1065.4	6.0
7.0	176.85	0.01649	53.64	144.76	992.1	1136.9	0.2581	1.5586	1.8167	144.74	922.7	1067.4	7.0
8.0	182.86	0.01653	47.34	150.79	988.5	1139.3	0.2674	1.5383	1.8057	150.77	918.4	1069.2	8.0
9.0	188.28	0.01656	42.40	156.22	985.2	1141.4	0.2759	1.5203	1.7962	156.19	914.6	1070.8	9.0
10	193.21	0.01659	38.42	161.17	982.1	1143.3	0.2835	1.5041	1.7876	161.14	911.1	1072.2	10
14.696	212.00	0.01672	26.80	180.07	970.3	1150.4	0.3120	1.4446	1.7566	180.02	897.5	1077.5	14.696
15	213.03	0.01672	26.29	181.11	969.7	1150.8	0.3135	1.4115	1.7549	181.06	896.7	1077.8	15
20	227.96	0.01683	20.089	196.16	960.1	1156.3	0.3356	1.3962	1.7319	196.10	885.8	1081.9	20
30	250.33	0.01701	13.746	218.82	945.3	1164.1	0.3680	1.3313	1.6993	218.73	869.1	1087.8	30
40	267.25	0.01715	10.498	236.03	933.7	1169.7	0.3919	1.2844	1.6763	235.90	856.1	1092.0	40
50	281.01	0.01727	8.515	250.09	924.0	1174.1	0.4110	1.2474	1.6585	249.93	845.4	1095.3	50
60	292.71	0.01738	7.175	262.09	915.5	1177.6	0.4270	1.2168	1.6438	261.90	836.0	1097.9	60
70	302.92	0.01748	6.206	272.61	907.9	1180.6	0.4409	1.1906	1.6315	272.38	827.8	1100.2	70
80	312.03	0.01757	5.472	282.02	901.1	1183.1	0.4531	1.1676	1.6207	281.76	820.3	1102.1	80
90	320.27	0.01766	4.896	290.56	894.7	1185.3	0.4641	1.1471	1.6112	290.27	813.4	1103.7	90
100	327.81	0.01774	4.432	298.40	888.8	1187.2	0.4740	1.1286	1.6026	298.08	807.1	1105.2	100
120	341.25	0.01789	3.728	312.44	877.9	1190.4	0.4916	1.0962	1.5878	312.05	795.6	1107.6	120
140	353.02	0.01802	3.220	324.82	868.2	1193.0	0.5069	1.0682	1.5751	324.35	785.2	1109.6	140
160	363.53	0.01815	2.834	335.93	859.2	1195.1	0.5204	1.0436	1.5640	335.39	775.8	1111.2	160
180	373.06	0.01827	2.532	346.03	850.8	1196.9	0.5325	1.0217	1.5542	345.42	767.1	1112.5	180
200	381.79	0.01839	2.288	355.36	843.0	1198.4	0.5435	1.0018	1.5453	354.68	759.0	1113.7	200
250	400.95	0.01865	1.8438	376.00	825.1	1201.1	0.5676	0.9588	1.5263	375.14	740.7	1115.8	250
300	417.33	0.01890	1.5433	393.84	809.1	1202.8	0.5879	0.9225	1.5104	392.79	724.3	1117.1	300
350	431.72	0.01913	1.3260	409.69	794.2	1203.9	0.6056	0.8910	1.4966	408.55	709.6	1118.0	350
400	444.59	0.0193	1.1613	424.0	780.5	1204.5	0.6214	0.8630	1.4844	422.6	695.9	1118.5	400
450	456.28	0.0195	1.0320	437.2	767.4	1204.6	0.6356	0.8378	1.4734	435.5	683.2	1118.7	450
500	467.01	0.0197	0.9278	449.4	755.0	1204.4	0.6487	0.8147	1.4634	447.6	671.0	1118.6	500
550	476.93	0.0199	0.8422	460.8	743.1	1203.9	0.6608	0.7934	1.4542	458.8	659.4	1118.2	550
600	486.21	0.0201	0.7698	471.6	731.6	1203.2	0.6720	0.7734	1.4454	469.4	648.3	1117.7	600
700	503.10	0.0205	0.6554	491.5	709.7	1201.2	0.6925	0.7371	1.4296	488.8	627.5	1116.3	700
800	518.23	0.0209	0.5687	509.7	688.9	1198.6	0.7108	0.7054	1.4235	506.6	607.8	1114.4	800
900	531.98	0.0212	0.5006	526.6	668.8	1195.4	0.7275	0.6744	1.4020	523.1	589.0	1112.1	900
1000	544.61	0.0216	0.4456	542.4	649.4	1191.8	0.7430	0.6467	1.3897	538.4	571.0	1109.4	1000
1100	556.31	0.0220	0.4001	557.4	630.4	1187.8	0.7575	0.6205	1.3780	552.9	553.5	1106.4	1100
1200	567.22	0.0223	0.3619	571.7	611.7	1183.4	0.7711	0.5956	1.3667	566.7	536.3	1103.0	1200
1300	577.46	0.0227	0.3293	585.4	593.2	1178.6	0.7840	0.5719	1.3559	580.0	519.4	1099.4	1300
1400	587.10	0.0231	0.3012	598.7	574.7	1173.4	0.7963	0.5491	1.3454	592.7	502.7	1095.4	1400
1500	596.23	0.0235	0.2765	611.6	556.3	1167.9	0.8082	0.5269	1.3351	605.1	486.1	1091.2	1500
2000	635.82	0.0257	0.1878	671.7	463.4	1135.1	0.8619	0.4230	1.2849	662.2	403.4	1065.6	2000
2500	668.13	0.0287	0.1307	730.6	360.5	1091.1	0.9126	0.3197	1.2322	717.3	313.3	1030.6	2500
3000	695.36	0.0346	0.0858	802.5	217.8	1020.3	0.9731	0.1885	1.1615	783.4	189.3	972.7	3000
3206.2	705.40	0.0503	0.0503	902.7	0	902.7	1.0580	0	1.0580	872.9	0	872.9	3206.2



Section TECH-D

Properties of Liquids

TECH-D-1 Viscosity

The viscosity of a fluid is that property which tends to resist a shearing force. It can be thought of as the internal friction resulting when one layer of fluid is made to move in relation to another layer.

Consider the model shown in Fig. 1, which was used by Isaac Newton in first defining viscosity. It shows two parallel planes of fluid of area A separated by a distance dx and moving in the same direction at different velocities V_1 and V_2 .

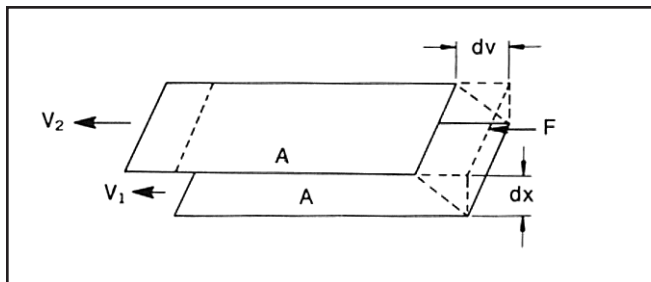


Fig. 1

The velocity distribution will be linear over the distance dx , and experiments show that the velocity gradient, $\frac{dv}{dx}$, is directly proportional to the force per unit area, $\frac{F}{A}$.

$$\frac{F}{A} = n \times \frac{dv}{dx} \quad \text{Where } n \text{ is constant for a given liquid and is called its viscosity.}$$

The velocity gradient, $\frac{dv}{dx}$, describes the shearing experienced by the intermediate layers as they move with respect to each other. Therefore, it can be called the "rate of shear", S . Also, the

force per unit area, $\frac{F}{A}$, can be simplified and called the "shear force" or "shear stress," F . With these simplified terms, viscosity can be defined as follows:

$$F = n \times S$$

$$\text{Viscosity} = n = \frac{F}{S} = \frac{\text{shear stress}}{\text{rate of shear}}$$

Isaac Newton made the assumption that all materials have, at a given temperature, a viscosity that is independent of the rate of shear. In other words, a force twice as large would be required to move a liquid twice as fast. Fluids which behave this way are called Newtonian fluids. There are, of course, fluids which do not behave this way, in other words their viscosity is dependent on the rate of shear. These are known as Non-Newtonian fluids.

Fig. 2 shows graphically the relationships between shear Stress (F), rate of shear (S), and viscosity (n) for a Newtonian liquid. The viscosity remains constant as shown in sketch 2, and in absolute units, the viscosity is the inverse slope of the line in sketch 1. Water and light oils are good examples of Newtonian liquids.

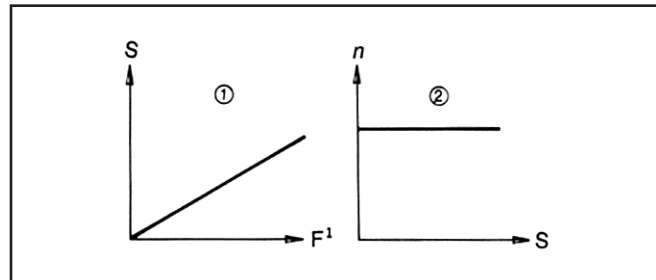


Fig. 2 Newtonian Liquid

Fig. 3 shows graphically the three most common types of Non-Newtonian liquids. Group A shows a decreasing viscosity with an increasing rate of shear. This is known as a **pseudo-plastic** material. Examples of this type are grease, molasses, paint, soap, starch, and most emulsions. They present no serious pumping problems since they tend to thin out with the high rates of shear present in a pump.

Group B shows a **dilatant** material or one in which the viscosity increases with an increasing rate of shear. Clay slurries and candy compounds are examples of dilatant liquids. Pumps must be selected with extreme care since these liquids can become almost solid if the shear rate is high enough. The normal procedure would be to oversize the pump somewhat and open up the internal clearances in an effort to reduce the shear rate.

Group C shows a **plastic** material. The viscosity decreases with increasing rate of shear. However, a certain force must be applied before any movement is produced. This force is called the yield value of the material. Tomato catsup is a good example of this type of material. It behaves similar to a pseudo-plastic material from a pumping standpoint.

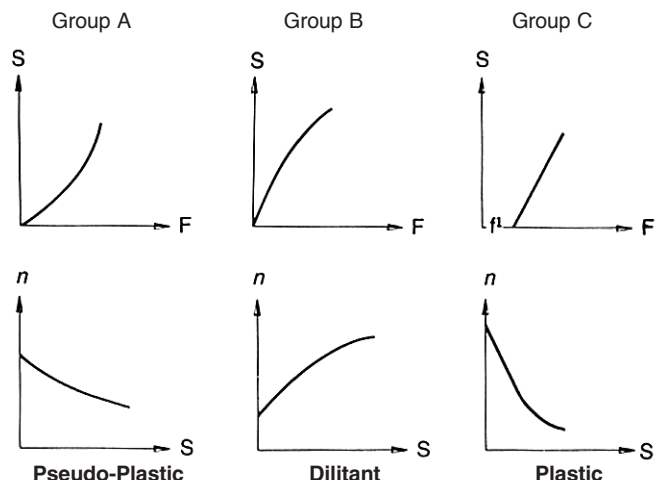


Fig. 3 Non-Newtonian Liquids

The viscosity of some Non-Newtonian liquids is dependent upon time as well as shear rate. In other words, the viscosity at any particular time depends upon the amount of previous agitation or shearing of the liquid. A liquid whose viscosity decreases with time at a given shear rate is called a thixotropic liquid. Examples are asphalts, glues, molasses, paint, soap, starch, and grease. Liquids whose viscosity increases with time are called rheopectic liquids, but they are seldom encountered in pumping applications.

There are two basic viscosity parameters: dynamic (or absolute) viscosity and kinematic viscosity. Dynamic viscosities are given in terms of force required to move a unit area a unit distance. This is usually expressed in pound-seconds per square foot in the English system which is equal to slugs per foot-second. The Metric system is more commonly used, however, in which the unit is the dyne-second per square centimeter called the Poise. This is numerically equal to the gram per centimeter-second. For convenience, numerical values are normally expressed in centipoise, which are equal to one-hundredth of a poise.

Most pipe friction charts and pump correction charts list kinematic

viscosity. The basic unit of kinematic viscosity is the stoke which is equal to a square centimeter per second in the Metric system. The corresponding English unit is square foot per second. The centistoke which is one-hundredth of a stoke is normally used in the charts. The following formula is used to obtain the kinematic viscosity when the dynamic or absolute viscosity is known:

$$\text{centistokes} = \frac{\text{centipoise}}{\text{sp. gr.}}$$

There are numerous types of viscometers available for determining liquid viscosities, most of which are designed for specific liquids or viscosity ranges. The Saybolt viscometers are probably the most widely used in the United States. The Saybolt Universal Viscometer measures low to medium viscosity, and the Saybolt Furol Viscometer measures high viscosities. The corresponding units are the SSU (Seconds Saybolt Universal) and the SSF (Seconds Saybolt Furol.) These units are found on most pipe friction and pump correction charts in addition to centistokes. A conversion chart for these and other units is shown in Fig. 4.

TECH-D-2A Viscosity Conversion Table

The following table will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal ssu	Kine-matic Viscosity Centi-stokes*	Seconds Saybolt Furol ssf	Seconds Red-wood 1 (Standard)	Seconds Red-wood 2 (Admiralty)	Degrees Engler	Degrees Barbey	Seconds Parlin Cup #7	Seconds Parlin Cup #10	Seconds Parlin Cup #15	Seconds Parlin Cup #20	Seconds Ford Cup #3	Seconds Ford Cup #4
31	1.00	-	29	-	1.00	6200	-	-	-	-	-	-
35	2.56	-	32.1	-	1.16	2420	-	-	-	-	-	-
40	4.30	-	36.2	5.10	1.31	1440	-	-	-	-	-	-
50	7.40	-	44.3	5.83	1.58	838	-	-	-	-	-	-
60	10.3	-	52.3	6.77	1.88	618	-	-	-	-	-	-
70	13.1	12.95	60.9	7.60	2.17	483	-	-	-	-	-	-
80	15.7	13.70	69.2	8.44	2.45	404	-	-	-	-	-	-
90	18.2	14.44	77.6	9.30	2.73	348	-	-	-	-	-	-
100	20.6	15.24	85.6	10.12	3.02	307	-	-	-	-	-	-
150	32.1	19.30	128	14.48	4.48	195	-	-	-	-	-	-
200	43.2	23.5	170	18.90	5.92	144	40	-	-	-	-	-
250	54.0	28.0	212	23.45	7.35	114	46	-	-	-	-	-
300	65.0	32.5	254	28.0	8.79	95	52.5	15	6.0	3.0	30	20
400	87.60	41.9	338	37.1	11.70	70.8	66	21	7.2	3.2	42	28
500	110.0	51.6	423	46.2	14.60	56.4	79	25	7.8	3.4	50	34
600	132	61.4	508	55.4	17.50	47.0	92	30	8.5	3.6	58	40
700	154	71.1	592	64.6	20.45	40.3	106	35	9.0	3.9	67	45
800	176	81.0	677	73.8	23.35	35.2	120	39	9.8	4.1	74	50
900	198	91.0	762	83.0	26.30	31.3	135	41	10.7	4.3	82	57
1000	220	100.7	896	92.1	29.20	28.2	149	43	11.5	4.5	90	62
1500	330	150	1270	138.2	43.80	18.7	-	65	15.2	6.3	132	90
2000	440	200	1690	184.2	58.40	14.1	-	86	19.5	7.5	172	118
2500	550	250	2120	230	73.0	11.3	-	108	24	9	218	147
3000	660	300	2540	276	87.60	9.4	-	129	28.5	11	258	172
4000	880	400	3380	368	117.0	7.05	-	172	37	14	337	230
5000	1100	500	4230	461	146	5.64	-	215	47	18	425	290
6000	1320	600	5080	553	175	4.70	-	258	57	22	520	350
7000	1540	700	5920	645	204.5	4.03	-	300	67	25	600	410
8000	1760	800	6770	737	233.5	3.52	-	344	76	29	680	465
9000	1980	900	7620	829	263	3.13	-	387	86	32	780	520
10000	2200	1000	8460	921	292	2.82	-	430	96	35	850	575
15000	3300	1500	13700	-	438	2.50	-	650	147	53	1280	860
20000	4400	2000	18400	-	584	1.40	-	860	203	70	1715	1150

***Kinematic Viscosity (in centistokes)**
= Absolute Viscosity (in centipoises)
Density

When the Metric System terms centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following expression can be used which will be sufficiently accurate for most calculations:

***Kinematic Viscosity (in centistokes)**
= Absolute Viscosity (in centipoises)
Specific Gravity

When the English System units are used, the density must be used rather than the specific gravity.

For values of 70 centistokes and above, use the following conversion:

$$SSU = \text{centistokes} \times 4.635$$

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor	Viscosimeter	Factor
Saybolt Furoil	10.	Parlin cup #15	98.2
Redwood Standard	1.095	Parlin cup #20	187.0
Redwood Admiralty	10.87	Ford cup #4	17.4
Engler – Degrees	34.5		

TECH-D-2B Viscosity Conversion Table

The following table will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal ssu	Kine-matic Viscosity Centi-stokes*	Approx. Seconds Mac Michael	Approx. Gardner Holt Bubble	Seconds Zahn Cup #1	Seconds Zahn Cup #2	Seconds Zahn Cup #3	Seconds Zahn Cup #4	Seconds Zahn Cup #5	Seconds Demmier Cup #1	Seconds Demmier Cup #10	Approx. Seconds Stormer 100 gpm Load	Seconds Pratt and Lambert "F"
31	1.00	-	-	-	-	-	-	-	-	-	-	-
35	2.56	-	-	-	-	-	-	-	-	-	-	-
40	4.30	-	-	-	-	-	-	-	1.3	-	-	-
50	7.40	-	-	-	-	-	-	-	2.3	-	2.6	-
60	10.3	-	-	-	-	-	-	-	3.2	-	3.6	-
70	13.1	-	-	-	-	-	-	-	4.1	-	4.6	-
80	15.7	-	-	-	-	-	-	-	4.9	-	5.5	-
90	18.2	-	-	-	-	-	-	-	5.7	-	6.4	-
100	20.6	125	-	38	18	-	-	-	6.5	-	7.3	-
150	32.1	145	-	47	20	-	-	-	10.0	1.0	11.3	-
200	43.2	165	A	54	23	-	-	-	13.5	1.4	15.2	-
250	54.0	198	A	62	26	-	-	-	16.9	1.7	19	-
300	65.0	225	B	73	29	-	-	-	20.4	2.0	23	-
400	87.0	270	C	90	37	-	-	-	27.4	2.7	31	7
500	110.0	320	D	-	46	-	-	-	34.5	3.5	39	8
600	132	370	F	-	55	-	-	-	41	4.1	46	9
700	154	420	G	-	63	22.5	-	-	48	4.8	54	9.5
800	176	470	-	-	72	24.5	-	-	55	5.5	62	10.8
900	198	515	H	-	80	27	18	-	62	6.2	70	11.9
1000	220	570	I	-	88	29	20	13	69	6.9	77	12.4
1500	330	805	M	-	-	40	28	18	103	10.3	116	16.8
2000	440	1070	Q	-	-	51	34	24	137	13.7	154	22
2500	550	1325	T	-	-	63	41	29	172	17.2	193	27.6
3000	660	1690	U	-	-	75	48	33	206	20.6	232	33.7
4000	880	2110	V	-	-	-	63	43	275	27.5	308	45
5000	1100	2635	W	-	-	-	77	50	344	34.4	385	55.8
6000	1320	3145	X	-	-	-	-	65	413	41.3	462	65.5
7000	1540	3760	-	-	-	-	-	75	481	48	540	77
8000	1760	4170	Y	-	-	-	-	86	550	55	618	89
9000	1980	4700	-	-	-	-	-	96	620	62	695	102
10000	2200	5220	Z	-	-	-	-	-	690	69	770	113
15000	3300	7720	Z2	-	-	-	-	-	1030	103	1160	172
20000	4400	10500	Z3	-	-	-	-	-	1370	137	1540	234

Fig. 4B

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor
Mac Michael	1.92 (approx.)
Demmier #1	14.6
Demmier #10	146.
Stormer	13. (approx.)

TECH-D-3 Determination of Pump Performance When Handling Viscous Liquids

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The performance of centrifugal pumps is affected when handling viscous liquids. A marked increase in brake horsepower, a reduction in head, and some reduction in capacity occur with moderate and high viscosities.

Fig. 5 provides a means of determining the performance of a conventional centrifugal pump handling a viscous liquid when its performance on water is known. It can also be used as an aid in selecting a pump for a given application. The values shown in Fig. 5 are averaged from tests of conventional single stage pumps of 2-inch to 8-inch size, handling petroleum oils. The correction curves are, therefore, not exact for any particular pump.

When accurate information is essential, performance tests should be conducted with the particular viscous liquid to be handled.

Limitations on Use of Viscous Liquid Performance Correction Chart

Reference is made to Fig. 5. This chart is to be used only within the scales shown. Do not extrapolate.

Use only for pumps of conventional hydraulic design, in the normal operating range, with open or closed impellers. Do not use for mixed flow or axial flow pumps or for pumps of special hydraulic design for either viscous or non-uniform liquids.

Use only where adequate NPSH is available in order to avoid the effect of cavitation.

Use only on Newtonian (uniform) liquids. Gels, slurries, paper stock and other non-uniform liquids may produce widely varying results, depending on the particular characteristics of the liquids.

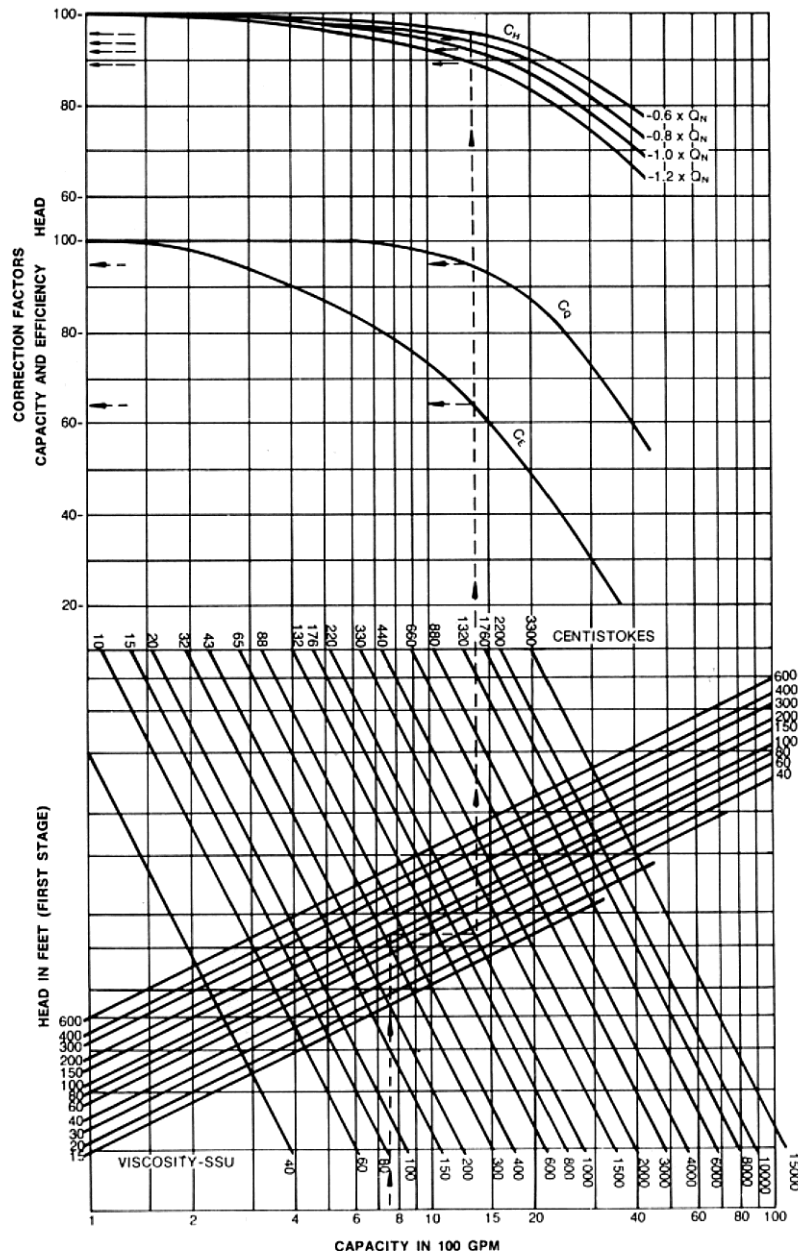


Fig. 5
Performance Correction
Chart

Symbols and Definitions Used in Determination of Pump Performance When Handling Viscous Liquids.

These symbols and definitions are:

Q_{vis}	= Viscous Capacity, gpm The capacity when pumping a viscous liquid.
H_{vis}	= Viscous Head, feet The head when pumping a viscous liquid.
E_{vis}	= Viscous Efficiency, per cent The efficiency when pumping a viscous liquid.
bhp_{vis}	= Viscous Brake Horsepower The horsepower required by the pump for the viscous conditions.
Q_w	= Water Capacity, gpm The capacity when pumping water.
H_w	= Water Head, feet The head when pumping water.
sp gr	= Specific Gravity
C_Q	= Capacity correction factor
C_H	= Head correction factor
C_E	= Efficiency correction factor
$1.0 Q_w$	= Water Capacity at which maximum efficiency is obtained.

The following equations are used for determining the viscous performance when the water performance of the pump is known:

$$\begin{aligned} Q_{vis} &= C_Q \times Q_w \\ H_{vis} &= C_H \times H_w \\ E_{vis} &= C_E \times E_w \end{aligned}$$

$$bhp_{vis} = \frac{Q_{vis} \times H_{vis} \times \text{sp gr}}{3960 \times E_{vis}}$$

C_Q , C_H and C_E are determined from Fig. 5 which is based on the water performance.

The following equations are used for approximating the water performance when the desired viscous capacity and head are given and the values of C_Q and C_H must be estimated from Fig. 5 using Q_{vis} and H_{vis} , as:

$$\begin{aligned} Q_w(\text{approx.}) &= \frac{Q_{vis}}{C_Q} \\ H_w(\text{approx.}) &= \frac{H_{vis}}{C_H} \end{aligned}$$

Instructions for Preliminary Selection of a Pump for a Given Head-Capacity-Viscosity Condition

Given the desired capacity and head of the viscous liquid to be pumped and the viscosity and specific gravity at the pumping temperature, Fig. 5 can be used to find approximate equivalent capacity and head when pumping water.

Enter the chart (Fig. 5) at the bottom with the desired viscous capacity, (Q_{vis}) and proceed upward to the desired viscous head (H_{vis}) in feet of liquid. For multistage pumps, use head per stage. Proceed horizontally (either left or right) to the fluid viscosity, and then go upward to the correction curves. Divide the viscous capacity (Q_{vis}) by the capacity correction factor (C_Q) to get the approximate equivalent water capacity (Q_w approximately).

Divide the viscous head (H_{vis}) by the head correction factor (C_H) from the curve marked " $1.0 \times Q_w$ " to get the approximate equivalent water head (H_w approximately). Using this new equivalent water head-capacity point, select a pump in the usual manner.

The viscous efficiency and the viscous brake horsepower may then be calculated.

This procedure is approximate as the scales for capacity and head on the lower half of Fig. 5 are based on the water performance. However, the procedure has sufficient accuracy for most pump selection purposes. Where the corrections are appreciable, it is desirable to check the selection by the method described below.

EXAMPLE. Select a pump to deliver 750 gpm at 100 feet total head of a liquid having a viscosity of 1000 SSU and a specific gravity of 0.90 at the pumping temperature.

Enter the chart (Fig. 5) with 750 gpm, go up to 100 feet head, over to 1000 SSU, and then up to the correction factors:

$$\begin{aligned} C_Q &= 0.95 \\ C_H &= 0.92 \text{ (for } 1.0 Q_w) \\ C_E &= 0.635 \\ Q_w &= \frac{750}{0.95} = 790 \text{ gpm} \\ H_w &= \frac{100}{0.92} = 108.8 \approx 109 \text{ feet head} \end{aligned}$$

Select a pump for a water capacity of 790 gpm at 109 feet head. The selection should be at or close to the maximum efficiency point for water performance. If the pump selected has an efficiency on water of 81 per cent at 790 gpm, then the efficiency for the viscous liquid will be as follows:

$$E_{vis} = 0.635 \times 81\% = 51.5 \text{ per cent}$$

The brake horsepower for pumping the viscous liquid will be:

$$bhp_{vis} = \frac{750 \times 100 \times 0.90}{3960 \times 0.515} = 33.1 \text{ hp}$$

For performance curves of the pump selected, correct the water performance as shown below.

Instructions for Determining Pump Performance on a Viscous Liquid When Performance on Water is Known

Given the complete performance characteristics of a pump handling water, determine the performance when pumping a liquid of a specified viscosity.

From the efficiency curve, locate the water capacity ($1.0 \times Q_w$) at which maximum efficiency is obtained.

From this capacity, determine the capacities ($0.6 \times Q_w$), ($0.8 \times Q_w$) and ($1.2 \times Q_w$).

Enter the chart at the bottom with the capacity at best efficiency ($1.0 \times Q_w$), go upward to the head developed (in one stage) (H_w) at this capacity, then horizontally (either left or right) to the desired viscosity, and then proceed upward to the various correction curves.

Read the values of (C_E) and (C_Q), and of (C_H) for all four capacities.

Multiply each head by its corresponding head correction factor to obtain the corrected heads. Multiply each efficiency value by (C_E) to obtain the corrected efficiency values which apply at the corresponding corrected capacities.

Plot corrected head and corrected efficiency against corrected capacity. Draw smooth curves through these points. The head at shut-off can be taken as approximately the same as that for water.

Calculate the viscous brake horsepower (bhp_{vis}) from the formula given above.

Plot these points and draw a smooth curve through them which should be similar to and approximately parallel to the brake horsepower (bhp) curve for water.

EXAMPLE. Given the performance of a pump (Fig. 6) obtained by test on water, plot the performance of this pump when handling oil with a specific gravity of 0.90 and a viscosity of 1000 SSU at pumping temperature.

On the performance curve (Fig. 6) locate the best efficiency point which determines (Qw). In this sample this is 750 gpm. Tabulate capacity, head and efficiency for (0.6 x 750), (0.8 x 750) and (1.2 x 750).

Using 750 gpm, 100 feet head and 1000 SSU, enter the chart and determine the correction factors. These are tabulated in Table of Sample Calculations. Multiply each value of head, capacity and efficiency by its correction factor to get the corrected values. Using the corrected values and the specific gravity, calculate brake horsepower. These calculations are shown on Table 6. Calculated points are plotted in Fig. 6 and corrected performance is represented by dashed curves.

TECH-D-4 Viscosity Corrections for Capacities of 100 GPM or Less

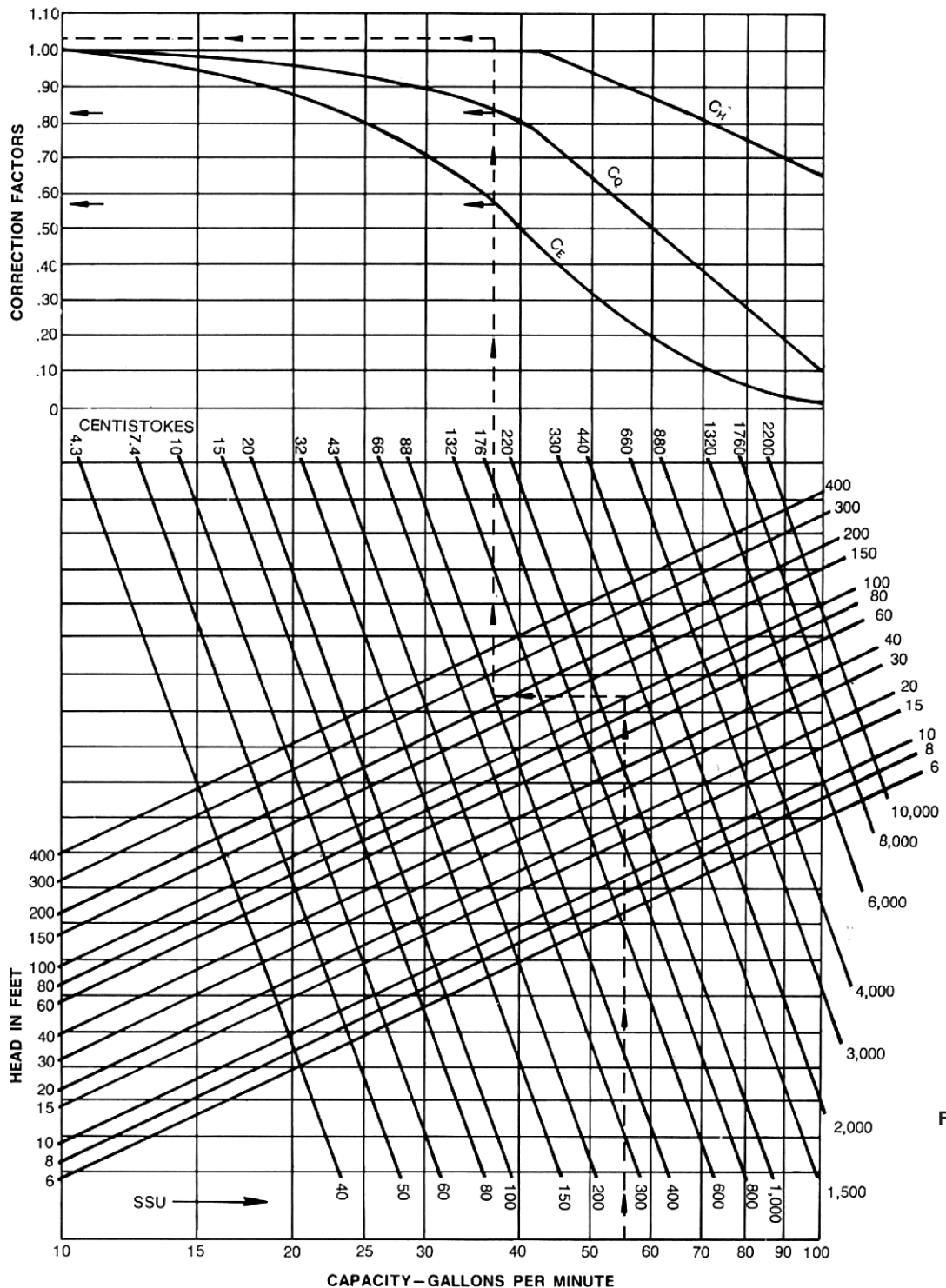


Fig. 5A

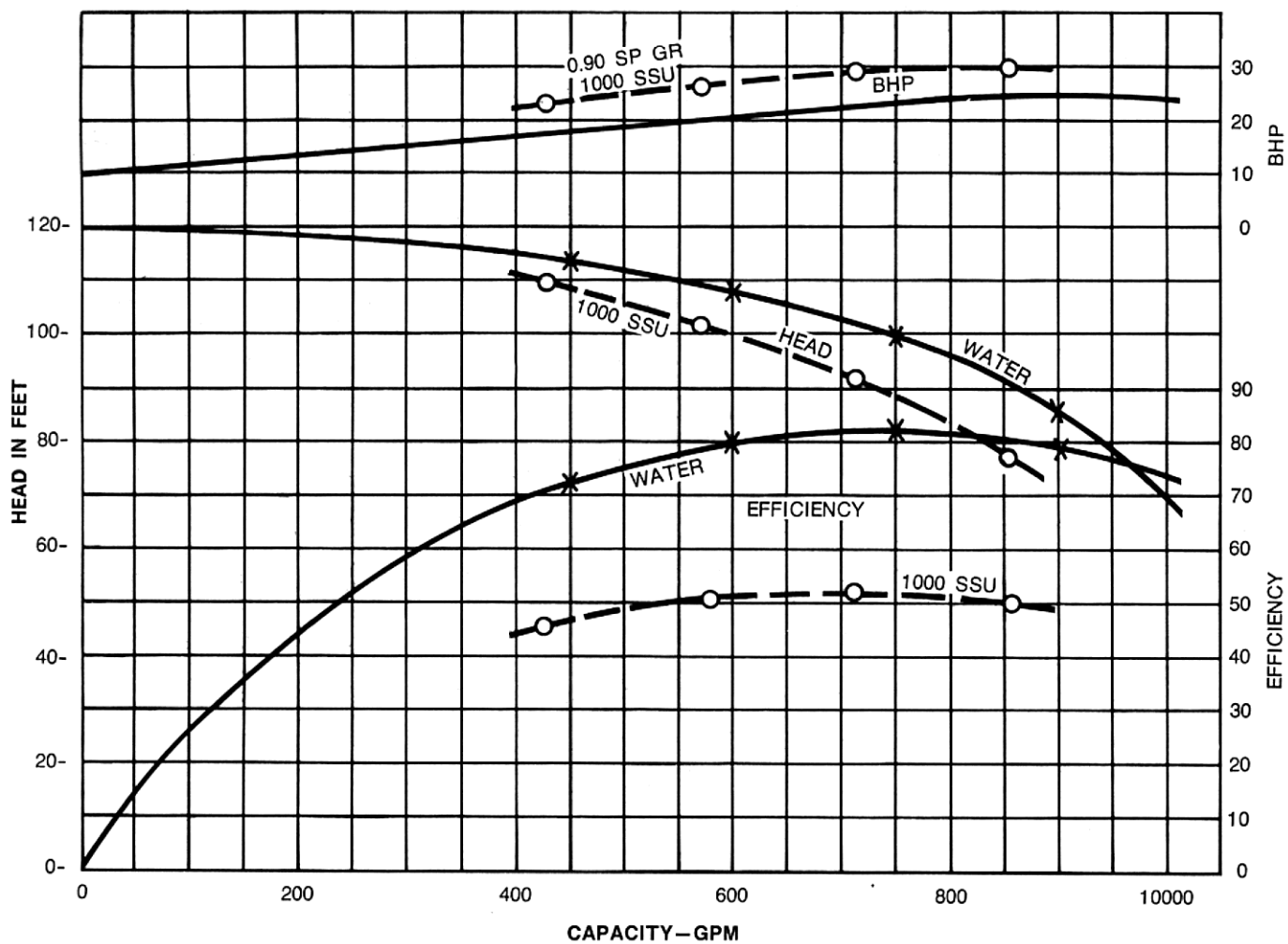


Fig. 6 Sample Performance Chart

TABLE 6

SAMPLE CALCULATIONS

	$0.6 \times Q_{NW}$	$0.8 \times Q_{NW}$	$1.0 \times Q_{NW}$	$1.2 \times Q_{NW}$
Water capacity (Q_w).....	450	600	750	900
Water head in feet (H_w).....	114	108	100	86
Water efficiency (E_w).....	72.5	80	82	79.5
Viscosity of liquid.....	1000 SSU	1000 SSU	1000 SSU	1000 SSU
C_Q —from chart.....	0.95	0.95	0.95	0.95
C_H —from chart.....	0.96	0.94	0.92	0.89
C_E —from chart.....	0.635	0.635	0.635	0.635
Viscous capacity— $Q_w \times C_Q$	427	570	712	855
Viscous head— $H_w \times C_H$	109.5	101.5	92	76.5
Viscous efficiency— $E_w \times C_E$	46.0	50.8	52.1	50.5
Specific gravity of liquid.....	0.90	0.90	0.90	0.90
bhp viscous.....	23.1	25.9	28.6	29.4

TECH-D-5A Viscosity of Common Liquids

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Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Freon	1.37 to 1.49 @ 70 F		.27-.32	70
Glycerine (100%)	1.26 @ 68F	2,950 813	648 176	68.6 100
Glycol:				
Propylene	1.038 @ 68F	240.6	52	70
Triethylene	1.125 @ 68 F	185.7	40	70
Diethylene	1.12	149.7	32	70
Ethylene	1.125	88.4	17.8	70
Hydrochloric Acid(31.5)	1.05 @ 68 F		1.9	68
Mercury	13.6		.118 .11	70 100
Phenol (Carbonic Acid)	.95 to 1.08	65	11.7	65
Silicate of soda	40 Baume 42 Baume	365 637.6	79 138	100 100
Sulfuric Acid (100%)	1.83	75.7	14.6	68
FISH AND ANIMAL OILS:				
Bone Oil	.918	220 65	47.5 11.6	130 212
Cod Oil	.928	150 95	32.1 19.4	100 130
Lard	.96	287 160	62.1 34.3	100 130
Lard Oil	.912 to .925	190 to 220 112 to 128	41 to 47.5 23.4 to 27.1	100 130
Menhadden Oil	.933	140 90	29.8 18.2	100 130
Neatsfoot Oil	.917	230 130	49.7 27.5	100 130
Sperm Oil	.883	110 78	23.0 15.2	100 130
Whale Oil	.925	163 to 184 97 to 112	35 to 39.6 19.9 to 23.4	100 130
Mineral Oils:				
Automobile Crankcase Oils (Average Midcontinent Paraffin Base)				
SAE 10	** .880 to .935	165 to 240 90 to 120	35.4 to 51.9 18.2 to 25.3	100 130
SAE 20	** .880 to .935	240 to 400 120 to 185	51.9 to 86.6 25.3 to 39.9	100 130
SAE 30	** .880 to .935	400 to 580 185 to 255	86.6 to 125.5 39.9 to 55.1	100 130
SAE 40	** .880 to .935	580 to 950 255 to 80	125.5 to 205.6 55.1 to 15.6	100 130 210
SAE 50	** .880 to .935	950 to 1,600 80 to 105	205.6 to 352 15.6 to 21.6	100 210
SAE 60	** .880 to .935	1,600 to 2,300 105 to 125	352 to 507 21.6 to 26.2	100 210
SAE 70	** .880 to .935	2,300 to 3,100 125 to 150	507 to 682 26.2 to 31.8	100 210
SAE 10W	** .880 to .935	5,000 to 10,000	1,100 to 2,200	0
SAE 20W	** .880 to .935	10,000 to 40,000	2,200 TO 8,800	0
Automobile Transmission Lubricants:				
SAE 80	** .880 to .935	100,000 max	22,000 max	0
SAE 90	** .880 to .935	800 To 1,500 300 to 500	173.2 to 324.7 64.5 to 108.2	100 130
SAE 140	** .880 to .935	950 to 2,300 120 to 200	205.6 to 507 25.1 to 42.9	130 210
SAE 250	** .880 to .935	Over 2,300 Over 200	Over 507 Over 42.9	130 210
Crude Oils:				
Texas, Oklahoma	.81 to .916	40 to 783 34.2 to 210	4.28 to 169.5 2.45 to 4.53	60 100
Wyoming, Montana	.86 to .88	74 to 1,215 46 to 320	14.1 to 263 6.16 to 69.3	60 100
California	.78 to .92	40 to 4,480 34 to 700	4.28 to 1,063 2.4 to 151.5	60 100
Pennsylvania	.8 to .85	46 to 216 38 to 86	6.16 to 46.7 3.64 to 17.2	60 100
Diesel Engine Lubricating Oils (Based on Average Midcontinent Paraffin Base): Federal Specification No. 9110	** .880 to .935	165 to 240 90 to 120	35.4 to 51.9 18.2 to 25.3	100 130

* Unless otherwise noted.

** Depends on origin or percent and type of solvent.

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Diesel Engine Lubricating Oils (Based on Average Midcontinent Paraffin Base): Federal Specification No.9170	** .880 to .935	300 to 410 140 to 180	64.5 to 88.8 29.8 to 38.8	100 130
Federal Specification No. 9250	** .880 to .935	470 to 590 200 to 255	101.8 to 127.8 43.2 to 55.1	100 130
Federal Specification No. 9370	** .880 to .935	800 to 1,100 320 to 430	173.2 to 238.1 69.3 to 93.1	100 130
Federal Specification No. 9500	** .880 to .935	490 to 600 92 to 105	106.1 to 129.9 18.54 to 21.6	130 210
Diesel Fuel Oils: No. 2 D	** .82 to .95	32.6 to 45.5 39	2 to 6 1 to 3.97	100 130
No.3 D	** .82 to .95	45.5 to 65 39 to 48	6 to 11.75 3.97 to 6.78	100 130
No.4 D	** .82 to .95	140 max 70 max	29.8 max 13.1 max.	100 130
No.5 D	** .82 to .95	400 max 165 max	86.6 max 35.2 max	122 160
Fuel Oils: No. 1	** .82 to .95	34 to 40 32 to 35	2.39 to 4.28 2.69	70 100
No. 2	** .82 to .95	36 to 50 33 to 40	3.0 to 7.4 2.11 to 4.28	70 100
No.3	** .82 to .95	35 to 45 32.8 to 39	2.69 to .584 2.06 to 3.97	100 130
No.5A	** .82 to .95	50 to 125 42 to 72	7.4 to 26.4 4.91 to 13.73	100 130
No.5B	** .82 to .95	125 to 400 72 to 310	26.4 to 86.6 13.63 to 67.1	100 122 130
No.6	** .82 to .95	450 to 3,000 175 to 780	97.4 to 660 37.5 to 172	122 160
Fuel Oil – Navy Specification	** .989 max	110 to 225 63 to 115	23 to 48.6 11.08 to 23.9	122 160
Fuel Oil – Navy II	1.0 max	1,500 max 480 max	324.7 max 104 max	122 160
Gasoline	.68 to .74		.46 to .88 .40 to .71	60 100
Gasoline (Natural)	76.5 degrees API		.41	68
Gas Oil	28 degrees Api	73 50	13.9 7.4	70 100
Insulating Oil: Transformer, switches and Circuit breakers		65 max	11.75 max	100
Kerosene	.78 to .82	35 32.6	2.69 2	68 100
Machine Lubricating Oil (Average Pennsylvania Paraffin Base): Federal Specification No.8	** .880 to .935	112 to 160 70 to 90	23.4 to 34.3 13.1 to 18.2	100 130
Federal Specification No. 10	** .880 to .935	160 to 235 90 to 120	34.3 to 50.8 18.2 to 25.3	100 130
Federal Specification No. 20	** .880 to .935	235 to 385 120 to 185	50.8 to 83.4 25.3 to 39.9	100 130
Federal Specification No. 30	** .880 to .935	385 to 550 185 to 255	83.4 to 119 39.9 to 55.1	100 130
Mineral Lard Cutting Oil: Federal Specefication Grade 1		140 to 190 86 to 110	29.8 to 41 17.22 to 23	100 130
Federal Specification Grade 2		190 to 220 110 to 125	41 to 47.5 23 to 26.4	100 130
Petrolatum	.825	100 77	20.6 14.8	130 160
Turbine Lubricating Oil: Federal Specification (Penn Base)	.91 Average	400 to 440 185 to 205	86.6 to 95.2 39.9 to 44.3	100 130
VEGETABLE OILS: Castor Oil	.96 @ 68 F	1,200 to 1,500 450 to 600	259.8 to 324.7 97.4 to 129.9	100 130
China Wood Oil	.943	1,425 580	308.5 125.5	69 100
Cocoanut Oil	.925	140 to 148 76 to 80	29.8 to 31.6 14.69 to 15.7	100 130
Corn Oil	.924	135 54	28.7 8.59	130 212
Cotton Seed Oil	.88 to .925	176 100	37.9 20.6	100 130

* Unless otherwise noted.

** Depends on origin or percent and type of solvent.

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
VEGETABLE OILS: Linseed Oil, Raw	.925 to .939	143 93	30.5 18.94	100 130
Olive oil	.912 to .918	200 115	43.2 24.1	100 130
Palm oil	.924	221 125	47.8 26.4	100 130
Peanut Oil	.920	195 112	42 23.4	100 130
Rape Seed Oil	.919	250 145	54.1 31	100 130
Rosin Oil	.980	1,500 600	324.7 129.9	100 130
Rosin (Wood)	1.09 Avg	500 to 20,000 1,000 to 50,000	108.2 to 4,400 216.4 to 11,000	200 190
Sesame Oil	.923	184 110	39.6 23	100 130
Soja Bean Oil	.927 to .98	165 96	35.4 19.64	100 130
Turpentine	.86 to .87	33 32.6	2.11 2.0	60 100
SUGARS, SYRUPS, MOLASSES, ETC. Corn Syrups	1.4 TO 1.47	5,000 to 500,000 1,500 to 60,000	1,100 to 110,000 324.7 to 13,200	100 130
Glucose	1.35 to 1.44	35,000 to 100,000 4,000 to 11,000	7,700 to 22,000 880 to 2420	100 150
Honey (Raw)		340	73.6	100
Molasses "A" (First)	140.6 to 146	1,300 to 23,00 700 to 8,000	281.1 to 5,070 151.5 to 1,760	100 130
Molasses "B" (Second)	1.43 to 1.48	6,500 to 60,000 3,000 to 15,000	1,410 to 13,200 660 to 3,300	100 130
Molasses "C" (Blackstrap or final)	1.46 to 1.49	17,00 to 250,000 6,000 to 75,00	2,630 to 5,500 1,320 to 16,500	100 130
Sucrose Solutions(Sugar Syrups) 60 Brix	1.29	230 92	49.7 18.7	70 100
62 Brix	1.30	310 111	67.1 23.2	70 100
64 Brix	1.31	440 148	95.2 31.6	70 100
66 Brix	1.326	650 195	140.7 42.0	70 100
68 Brix	1.338	1,000 275	216.4 59.5	70 100
70 Brix	1.35	1,650 400	364 86.6	70 100
72 Brix	1.36	2,700 640	595 138.6	70 100
74 Brix	1.376	5,500 1,100	1,210 238	70 100
76 Brix	1.39	10,000 2,000	2,200 440	70 100
TARS: Tar Coke Oven	1.12+	3,000 to 8,000 650 to 1,400	600 to 1,760 140.7 to 308	71 100
Tar Gas House	1.16 to 1.30	15,000 to 300,000 2,000 to 20,000	3,300 to 66,000 440 to 4,400	70 100
Road Tar: Grade RT-2	1.07+	200 to 300 55 to 60	43.2 to 64.9 8.77 to 10.22	122 212
Grade RT-4	1.08+	400 to 700 65 to 75	86.6 to 154 11.63 to 14.28	122 212
Grade RT-6	109+	1,000 to 2,000 85 to 125	216.4 to 440 16.83 to 26.2	122 212
Grade RT-8	1.13+	3,000 to 8,000 150 to 225	660 to 1,760 31.8 to 48.3	122 212
Grade RT-10	1.14+	20,000 to 60,000 250 to 400	4,400 to 13,200 53.7 to 86.6	122 212
Grade RT-12	1.15+	114,000 to 456,000 500 to 800	25,000 to 75,000 108.2 to 173.2	122 212
Pine Tar	1.06	2,500 500	559 108.2	100 132
MISCELLANEOUS Corn Starch Solutions: 22 Baume	1.18	150 130	32.1 27.5	70 100
24 Baume	1.20	600 440	129.8 95.2	70 100

* Unless otherwise noted.

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
MISCELLANEOUS Corn Starch Solutions: 25 Baume	1.2	1400 800	303 17.2	70 100
Ink- Printers	1.00 to 1.38	2,500 to 10,000 1,100 to 3,000	550 to 2,200 238.1 to 660	100 130
Tallow	.918 Avg.	56	9.07	212
Milk	1.02 to 1.05		1.13	68
Varnish – Spar	.9	1425 650	313 143	68 100
Water- Fresh	1.0		1.13 .55	60 130

* Unless otherwise noted.

TECH-D-5B Physical Properties of Common Liquids

Liquid	Sp. Gr. 60° F (16°C)	Melting Point °F (°C)	Boiling Point °F (°C)	pH At 77° F (25°C)	VISCOSITY							
					SSU				Centipoise			
					40°F	80°F	120°F	160°F	4°C	27°C	49°C	71°C
Acetic Acid Glacial 8.8% (1N) .88% (.1N) .09 (.01N)	1.05 1.01	63 (17)	244 (118)	 2.4 2.9 3.4					1.6	1.2	.8	.6
Acetone	.79	- 137 (-94)	133 (56)						.4	.3	.3	.2
Alum, 0.6% (0.1N)				3.2								
Ammonia 100% 26% 1.7% (1N) .17% (0.1N) .02% (.01N)	.77 .91	-108 (-78)	-27 (-33)	 11.6 11.1 10.6					.14 1.8	.1 1.2	.08	.06
Asphalt Unblended RS1 RC2 RC5 Emulsion	1.1-1.5 1.0 1.0 1.0 1.0					155-1,000 500,000 1000-7,000	160 2,400-5,000 45,000	(12,000 at 250°F) 85 8,000		86	34	17
Benzene	.84	42 (6)	176 (80)						.8	.6	.5	.3
Benzoic Acid 0.1% (.01N)				3.1								
Black liquor, 50% 70%	1.3 1.5						5,000	(80-150 at 250°F) (6,300 at 250°F)				(15-37 at 121°C) 1,400 at 121°C)
Borax 1% (0.1N)	1.7 (75)	167		 9.2								
Boric Acid 0.2% (0.1N)		338 (171)		 5.2								
Butane	.59									.18		
Calcium Carbonate Sat.				9.4								
Calcium Chloride 25%	1.23	-21 (-29)							4.5	2.1	.9	.5
Calcium Hydroxide Sat. (Slaked Lime)				12.4								
Carbolic Acid (Phenol)	1.07	109 (43)	360 (182)			60			14.5	7.3	3.9	2.1

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Liquid	Sp. Gr. 60° F (16°C)	Melting Point °F (°C)	Boiling Point °F (°C)	pH At 77° F 25°C)	VISCOSITY							
					SSU				Centipoise			
					40°F	80°F	120°F	160°F	4°C	27°C	49°C	71°C
Carbonic Acid Sat.				3.8								
Carbon Tetrachloride	1.58	-95 (-71)	170 (77)						1.3	.9	.7	.6
Citric Acid .6% (1N)				2.2								
Corn Oil	.92						135					
Corn Starch, 22° Baume	1.18					150	130					
25° Baume	1.21					1,400	800					
Corn Syrup	1.4						5,000- 500,000					
Cotton Seed Oil	.9						176					
Crude Oil Pennsylvania	.8					200	86					
Wyoming	.9					1,100	320					
48° API	.79									2.8		
32.6° API	.86									20.0		
Dowtherm A	.99	54 (12)	500 (260)									
Dowtherm C	1.1	70 (21)	600 (316)									
Ethane	.37									(.05 at 16° C.)		
Ethyl Alcohol	.79	- 173 (- 144)	173 (78)						1.6	1.0	.7	.5
Ethyl Alcohol 95%	.81								2.0	1.3	.8	.6
Ethylene Glycol	1.1	9 (- 13)	387 (198)		185	86	53	39	44	19	9	4
Ethyl Acetate	.9									.49		
Formic Acid, 1.22 100%	122	47 (8)	213 (100)						2.4	1.5	-	.8
.5% (.1N)	1.0	-	-	2.3								
Fuel Oil												
No. 1 (Kerosene)	.81				40	36	31	30	3.3	2.1	1.4	0.9
No. 2	.86				43	36	33	32	4.6	2.6	1.6	1.2
No. 3	.89				84	52	41	37	15	7	4	3
No. 6 (Bunker C)	.96					4,500- 20,000	680- 1,900	180- 500		1,00	155	40
Diesel 2D	.82-.95				100	53	40	35				
3D	.82-.95				200	80	50	40				
5D	.82-.95				15,000	2,000	400	160				
Gasoline	.6-.7				30				.7	.6	.4	.3
Glucose	1.4						35,000- 100,000					
Glycerine (Glycerol) 50%	1.26 1.13	64 (18)	554 (290)		25,000	3,100	700	230	6,260 11	490 5.4	130 2.8	56 1.5
Hydrochloric Acid, 38%	1.20	- 13 (-25)										
31.5%	1.15	-115 (-46)							2.5	1.8	1.4	1.1
3.6% (1N)				0.1								
.36% (0.1N)				1.1								
.04% (.01N)				2.0								
Jet Fuel	.7-.8				35							
Lactic Acid		63 (17) —	252 (122) —	2.4								
Methyl Alcohol 80%	.80 .82	- 144 (-98)	149 (65)						.8 1.0	.5 .7	.4 .5	.4 .4
Milk, 3.5%	1.03			6.3-6.6						1.1		
Molasses A	1.40					10,000	2,600- 60,000					
Molasses C	1.49					300,000	25,000- 250,000					

Liquid	Sp. Gr. 60° F (16°C)	Melting Point °F (°C)	Boiling Point °F (°C)	pH At 77° F 25°C)	VISCOSITY							
					SSU				Centipoise			
					40°F	80°F	120°F	160°F	4°C	27°C	49°C	71°C
Nitric Acid, 95%	1.50	-44 (-44)	187 (86)						1.4	1.0	.8	.6
60%	1.37	-9 (-23)							3.4	2.2	1.5	1.0
Oil, 5W					550	160	74	51	110	30	12	7
10W	.9				1,500	265	120	64	170	50	22	11
20W	.9				2,900	500	170	80	580	98	33	14
30W	.9				5,000	870	260	110	1,200	200	60	25
50W	.9				23,000	3,600	720	225	-	400	100	45
70W	.9				120,000	10,000	1,800	500	-	4,000	-	-
Oleic Acid	0.89	13 (-11)	547 (286)							26		
Olive Oil	.9				1,500	320	150	80				
Palmetic Acid	0.85	146 (63)	520 (271)									
Parafin	.9	100 (38)	660 (349)									
Peanut Oil	.9				1,200	300	150	80				
Propane	.51										.12	
Propylene Glycol	1.0					241						
Potassium Hydroxide 5.7% (1N) 0.57% (0.1N) 0.06% (0.01N)				14.0 13.0 12.0								
Rosin	1.09							500-20,000				
Sodium Bicarbonate 0.4% (0.1N)				8.4								
Sodium Chloride, 25%	1.19								3.3	2.1	1.3	.9
Sodium Hydroxide, 50% 30% 4% (1N) 0.4% (0.1N) .04% (.01N)	1.53 1.33			14.0 13.0 12.0	950	240 58	84	46	250	77 10	26 4.5	10 2.5
Stearic Acid	.85	157 (69)	721 (383)									
Sucrose, 60%	1.29	10 (-12)	218 (103)		500	150	68		156	41	14	7
40 %	1.18	25 (-4)	214 (101)						120	5	2.5	1.6
Sugar Syrup 60 Brix 70 Brix 76 Brix	1.29 1.35 1.39					230 1,650 10,000	92 400 2,000					
Sulfur Molten	2.06	239 (115)	832 (445)						(11 at 123°C)	(9 at 159°C)	(22 at 160°C)	(16,000 at 184°C)
Sulfuric Acid 110% (Fuming, Oleum) 100%	1.83	92 50 (10)	342 (33)	(172)	280	100 75	55		82	41	22	12
98%	1.84	37 (3)	554 (290)		118	68	45	37	46	23	12	6
60%	1.50	-83 (-64)	282 (139)						8.9	5.8	3.9	2.7
20%	1.14	8 (-13)	218 (103)						2.5	1.4	0.8	0.6
4.9% (1N) .49% (.1N) .05 (.01N)					0.3 1.2 2.1							
Toluene	.86	-139 (-95)	231 (111)						.8	.6	.4	.4
Trichloroethylene	1.62	-99 (-72)	189 (87)						.7	.6	.5	.4
Turpentine	.86	140 (-10)	320 (160)		34	33	32	32	1.9	1.4	.9	.7
Vinegar				2.4-3.4								
Water	1.0	32 (0)	212 (100)	6.5-8.0	32				1.6	.9	.6	.4
Wines		1.03			2.8-3.8							

TECH-D-6 Friction Loss for Viscous Liquids. Loss in Feet of Liquid per 100 Feet of New Schedule 40 Steel Pipe

GPM	Nom. Pipe Size	Kinematic Viscosity – Seconds Saybolt Universal													
		Water	100	200	300	400	500	600	800	1000	1500	2000	3000	5000	10,000
3	1/2	10.0	25.7	54.4	83	108	135	162	218	273	411	545	820	1350	-
	3/4	2.50	8.5	17.5	26.7	35.5	44	53	71	88	131	176	265	440	880
	1	0.77	3.2	6.6	10.2	13.4	16.6	20.0	26.6	34	50	67	100	167	-
5	3/4	6.32	14.1	29.3	44	59	74	88	117	147	219	293	440	740	1470
	1	1.93	5.3	11.0	16.8	22.4	28	33	44	56	83	111	167	-	-
	1 1/4	0.51	1.8	3.7	5.5	7.6	9.5	11.1	14.8	18.5	28	37	56	94	187
10	1	6.86	11.2	22.4	33.5	45	56	66	89	112	165	223	-	-	-
	1 1/4	1.77	3.6	7.5	11.2	14.9	19.1	22.4	30	37	55	74	112	190	-
	1 1/2	0.83	1.9	4.2	6.0	8.1	10.2	12.3	16.5	20.3	31	41	62	102	207
15	1	14.6	26	34	50	67	85	104	137	172	-	-	-	-	-
	1 1/4	3.72	6.4	11.3	16.9	22.4	29	34	45	57	84	112	167	-	-
	1 1/2	1.73	2.8	6.2	9.2	12.4	15.3	18.4	25	30	46	61	92	152	-
20	1	25.1	46	46	67	90	111	133	180	220	-	-	-	-	-
	1 1/2	2.94	5.3	8.1	12.2	16.2	20.3	25	33	40	61	81	122	203	-
	2	0.87	1.5	3.0	4.4	6.0	7.4	9.0	11.9	14.8	22.4	30	45	74	147
30	1 1/2	6.26	11.6	12.2	18.2	24.3	30	37	50	61	91	122	182	-	-
	2	1.82	3.2	4.4	6.7	9.0	11.1	13.2	17.8	22.2	33	45	67	178	222
	2 1/2	0.75	1.4	2.2	3.2	4.4	5.5	6.5	8.8	10.9	16.6	22.0	33	55	110
40	1 1/2	10.8	19.6	20.8	24	32	40	50	65	81	121	162	243	400	810
	2	3.10	5.8	5.8	9.0	11.8	14.8	17.7	24	30	44	59	89	148	-
	2 1/2	1.28	2.5	3.0	4.4	5.8	7.4	8.8	11.8	14.6	22.2	29	44	73	145
60	2	6.59	11.6	13.4	13.4	17.8	22.2	27	36	45	67	89	134	220	-
	2 1/2	2.72	5.1	5.5	6.5	8.8	10.9	13.1	17.8	22.0	34	44	66	109	220
	3	0.92	1.8	1.8	2.8	3.7	4.6	5.6	7.3	9.2	13.8	18.5	27	46	92
80	2 1/2	4.66	8.3	9.7	9.7	11.8	14.6	17.6	24	29	44	58	87	145	-
	3	1.57	3.0	3.2	3.7	4.8	6.2	7.3	9.7	12.2	18.3	24	37	61	122
	4	0.41	0.83	0.83	1.2	1.7	2.1	2.5	3.3	4.2	6.2	8.3	12.5	20.6	41
100	2 1/2	7.11	12.2	14.1	14.8	14.8	18.5	22	29	36	55	73	109	183	-
	3	2.39	4.4	5.1	5.1	6.2	7.6	9.1	12.1	15.2	23	31	46	77	150
	4	0.62	1.2	1.3	1.5	2.1	2.5	3.1	4.1	5.1	7.8	10.4	15.5	26	51
125	3	3.62	6.5	7.8	8.1	8.1	9.7	11.5	15.3	19.4	29	39	58	97	193
	4	0.94	1.8	2.1	2.1	2.6	3.2	3.9	5.2	6.4	9.8	12.7	19.3	32	65
	6	0.12	0.25	0.28	0.39	0.52	0.63	0.78	1.0	1.3	1.9	2.6	3.9	6.4	13.0
150	3	5.14	9.2	10.4	11.5	11.5	11.5	13.7	18.4	23	35	46	69	115	230
	4	1.32	2.4	2.9	2.9	3.1	3.9	4.6	6.2	7.8	11.5	15.4	23	39	78
	6	0.18	0.34	0.39	0.46	0.62	0.77	0.9	1.2	1.5	2.3	3.0	4.6	7.6	15.2
175	3	6.9	11.7	13.8	15.8	15.8	15.8	15.9	21.4	27	40	53	80	133	-
	4	1.76	3.2	4.0	4.0	4.0	4.6	5.4	7.4	9.2	13.7	18.2	28	46	92
	6	0.23	0.44	0.52	0.54	0.7	0.9	1.1	1.4	1.8	2.6	3.5	5.3	8.8	17.8
200	3	8.90	15.0	17.8	20.3	20.3	20.3	20.3	25	31	46	61	91	152	-
	4	2.27	4.2	5.1	5.1	5.1	5.1	6.2	8.3	10.4	15.5	20.6	31	51	103
	6	0.30	0.58	0.69	0.69	0.81	1.0	1.2	1.6	2.0	3.0	3.9	6.2	9.9	20.1
250	4	3.46	6.0	7.4	8.0	8.0	8.0	8.0	10.2	12.9	19.4	26	39	64	130
	6	0.45	0.83	0.99	1.0	1.0	1.2	1.5	2.1	2.5	3.7	5.1	7.6	12.5	-
	8	0.12	0.21	0.28	0.28	0.35	0.42	0.51	0.67	0.83	1.2	1.7	2.5	4.2	8.3
300	6	1.09	8.5	9.9	11.6	11.6	11.6	11.6	12.4	15.5	23	31	46	77	155
	8	0.28	1.2	1.4	1.5	1.5	1.5	1.8	2.5	3.0	4.6	6.0	9.1	15.0	30
	10	0.09	0.30	0.39	0.39	0.42	0.51	0.61	0.82	1.0	1.5	2.0	3.0	5.1	9.9
400	6	1.09	1.9	2.3	2.5	2.8	2.8	2.8	3.2	3.9	6.0	8.1	12.1	20.1	-
	8	0.28	0.53	0.62	0.67	0.67	0.67	0.81	1.1	1.3	2.0	2.8	4.1	6.7	13.5
	10	0.09	0.18	0.21	0.23	0.23	0.28	0.32	0.43	0.53	0.81	1.1	1.6	2.8	5.3
600	6	2.34	4.2	5.1	5.3	5.5	6.0	6.2	6.2	6.2	9.0	12.0	18.5	-	-
	8	0.60	1.1	1.3	1.4	1.5	1.5	1.5	1.7	2.0	3.0	3.9	6.2	9.9	20
	10	0.19	0.37	0.42	0.46	0.51	0.51	0.51	0.65	0.81	1.2	1.6	2.4	4.2	8.1
800	6	4.03	6.5	8.1	8.5	9.2	9.7	11.1	11.1	11.1	12.0	16.0	-	-	-
	8	1.02	1.8	2.2	2.3	2.5	2.8	2.8	2.8	2.8	3.9	5.3	8.2	13.4	-
	10	0.33	0.60	0.69	0.78	0.88	0.92	0.92	0.92	1.1	1.6	2.1	3.2	5.3	10.9
1000	8	1.56	2.5	3.2	3.5	3.7	4.2	4.4	4.4	4.4	5.1	6.7	10.2	16.6	-
	10	0.50	0.88	1.0	1.2	1.3	1.4	1.4	1.4	1.4	2.0	2.8	4.0	6.7	13.4
	12	0.21	0.39	0.46	0.51	0.55	0.58	0.58	0.58	0.67	1.0	1.3	2.0	3.5	6.7

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TECH-D-7 Pumping Liquids with Entrained Gas

Pump applications in many industrial processes involve handling liquid and gas mixtures. The entrained gas may be an essential part of an industrial process, or it may be unwanted. The Pulp and Paper industry, for example, injects from between 4% and 10% air into a dilute pulp slurry as part of the ink removal process in a flote cell used in paper recycling. Many chemical and petrochemical processes also involve pumping a two phase flow. Unwanted entrained gas can result from excess agitation or vortexing due to inadequate submergence on the suction of a pump.

The proper selection of a centrifugal pump for liquid and gas (two phase) mixtures is highly dependent on the amount of gas and the characteristics of the liquid. The presence of entrained gases will reduce the output of centrifugal pumps and can potentially cause loss of prime. Conventional pump designs can be used for low percentages by volume (up to 4%), while special modified impellers can be used effectively for up to 10% gas by volume. Performance corrections are required in all cases with gas content above approximately 2%. Gas concentrations above 10% can also be handled, but only with special design pumps (pumps with inducers, vortex pumps, or pumps with gas extraction).

Virtually any type of centrifugal pump can handle some amount of entrained gas. The problem to be addressed is the tendency for the

gas to accumulate in the pump suction inhibiting flow and head generation. If gas continues to accumulate, the pump may lose prime. Fig. 1 shows how the performance of a standard end suction pump is affected by various amounts of air. With a minor performance correction, this type of pump is reasonably efficient in handling up to approximately 4% entrained gas.

As the percentage of gas exceeds 4% by volume, the performance of a conventional pump begins to degrade drastically (Fig. 1) until the pump becomes unstable, eventually losing prime. It has been found beneficial to increase the impeller running clearance (0.090 to 0.180 in.) allowing for greater leakage. This is effective in preventing loss of prime with gas concentrations up to 10%. Fig. 2 shows a standard end suction open impeller pump with clearances opened for gas handling.

Numerous tests have been conducted in an effort to quantify the performance corrections for various gas concentrations for both standard pumps and pumps with open clearances. The performance corrections are affected by many variables, including pump specific speed, operating speed, impeller design and number of vanes, operating point on the curve, and suction pressure. Performance correction charts are not presented here due to the numerous variables, but Goulds Applications Department can make recommendations and selections for most specific applications.

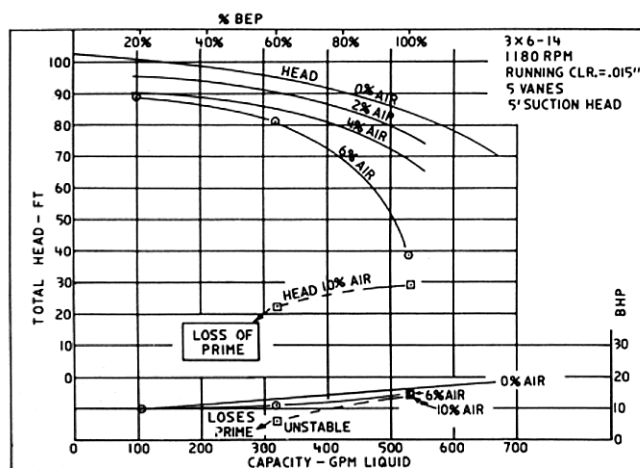


Fig. 1 Head and Power vs Capacity Zero to Ten Percent Air by Volume for Normal Running Clearance

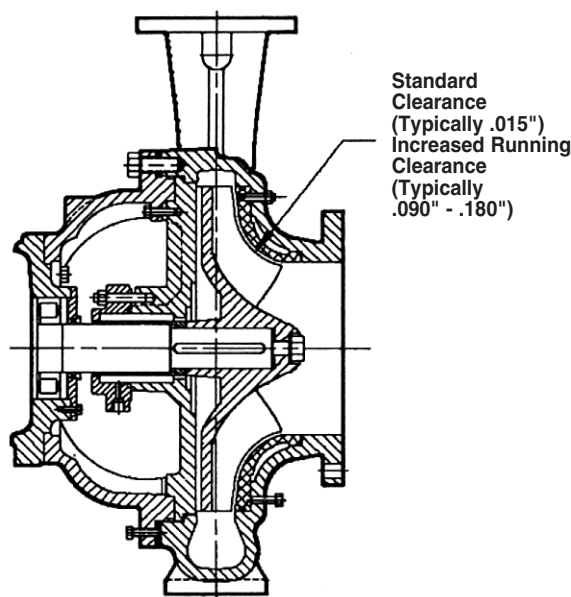


Fig. 2 Open Impeller End Suction Pump with Normal Running Clearance and Increased Running Clearance.

TECH-D-8A Solids and Slurries - Definition of Terms

APPARENT VISCOSITY

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

CRITICAL CARRYING VELOCITY

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

EFFECTIVE PARTICLE DIAMETER

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

FRICTION CHARACTERISTIC

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

HETEROGENEOUS MIXTURE

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

HOMOGENEOUS FLOW (FULLY SUSPENDED SOLIDS)

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

HOMOGENEOUS MIXTURE

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

NON-HOMOGENEOUS FLOW (PARTIALLY SUSPENDED SOLIDS)

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

NON-SETTLING SLURRY

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

PERCENT SOLIDS BY VOLUME

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

PERCENT SOLIDS BY WEIGHT

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

SALTATION

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

SETTLING SLURRY

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

SETTLING VELOCITY

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

SQUARE ROOT LAW

A rule used to calculate the approximate increase in critical carrying velocity for a given slurry when pipe size is increased. It states:

$$V_L = V_s = \left(\frac{D_L}{D_s} \right)^{1/2}$$

Where:

- V_L = Critical carrying velocity in larger pipe
- D_L = Diameter of larger pipe
- V_s = Critical carrying velocity in smaller pipe
- D_s = Diameter of smaller pipe

NOTE: This rule should not be used when pipe size is decreased.

VISCOSITY TYPES

(For definitions of the various types of viscosities applicable to slurries, see Rheological Definitions.)

YIELD VALUE (STRESS)

The stress at which many non-Newtonian slurries will start to deform and below which there will be no relative motion between adjacent particles in the slurry.

TECH-D-8B Solids and Slurries - Slurry Pump Applications

Determining the when to use a slurry style centrifugal pump can be a challenging decision. Often the cost of a slurry pump is many times that of a standard water pump and this can make the decision to use a slurry pump very difficult. One problem in selecting a pump type is determining whether or not the fluid to be pumped is actually a slurry. We can define a slurry as any fluid which contains more solids than that of potable water. Now, this does not mean that a slurry pump must be used for every application with a trace amount of solids, but at least a slurry pump should be considered.

Slurry pumping in its simplest form can be divided into three categories: the light, medium and heavy slurry. In general, **light slurries** are slurries that are not intended to carry solids. The presence of the solids occurs more by accident than design. On the other hand, **heavy slurries** are slurries that are designed to transport material from one location to another. Very often the carrying fluid in a heavy slurry is just a necessary evil in helping to transport the desired

material. The **medium slurry** is one that falls somewhere in between. Generally, the Percent solids in a medium slurry will range from 5% to 20% by weight.

After a determination has been made as to whether or not you are dealing with a heavy, medium, or light slurry, it is then time to match a pump to the application. Below is a general listing of the different characteristics of a light, medium, and heavy slurry.

Light Slurry Characteristics:

- Presence of solids is primarily by accident
- Solids Size < 200 microns
- Non-settling slurry
- The slurry specific gravity < 1.05
- Less than 5% solids by weight

Medium Slurry Characteristics:

- Solids size 200 microns to 1/4 inch (6.4mm)
- Settling or non-settling slurry
- The slurry specific gravity < 1.15
- 5% to 20% solids by weight

Heavy Slurry Characteristics:

- Slurry's main purpose is to transport material
- Solids > 1/4 inch (6.4mm)
- Settling or non-settling slurry
- The slurry specific gravity > 1.15
- Greater than 20% solids by weight

The previous listing is just a quick guideline to help classify various pump applications. Other considerations that need to be addressed when selecting a pump model are:

- Abrasive hardness
- Particle shape
- Particle size
- Particle velocity and direction
- Particle density
- Particle sharpness

The designers of slurry pumps have taken all of the above factors into consideration and have designed pumps to give the end user maximum expected life. Unfortunately, there are some compromises that are made in order to provide an acceptable pump life. The following short table shows the design feature, benefit, and compromise of the slurry pump.

SLURRY PUMP DESIGN		
Design Feature	Benefit	Compromise
Thick Wear Sections	Longer component life	Heavier, more expensive parts
Larger Impellers	Slower pump speeds longer component life	Heavier, more expensive parts
Specialty Materials	Longer component life	Expensive parts
Semi Volute or Concentric Casing	Improved pump life	Loss in efficiency
Extra Rigid Power Ends	Improved bearing lives	More expensive shafts and bearings

Although selecting the proper slurry pump for a particular application can be quite complex, the selection task can be broken down into a simplified three-step process:

1. Determine which group of possible pump selections best matches your specific application.
2. Plot the system curve depicting the required pump head at various capacities.
3. Match the correct pump performance curve with the system curve.

Slurry pumps can be broken down into two main categories. The rubber-lined pump and the hard metal pump. However, because of the elastomer lining, the rubber-lined pumps have a somewhat limited application range. Below is a general guideline which helps distinguish when to apply the rubber-lined pumps.

Rubber Lined	Hard Metal Pump
Solids < 1/2 inch (13mm) Temperature < 300° F (150°C) Low Head service < 150 feet (46m) Rounded particles Complete pH range	Solids > 1/4 inch (6.4mm) Temperature < 250° F (120°C) Heads above 150 feet (46m) Sharp/Jagged particles pH range from 4 to 12 Hydrocarbon based slurry

It should be noted, however, that a hard metal pump can also be used for services that are outlined for the rubber-lined pump. After a decision has been made whether to use a hard metal pump or a rubber-lined pump, it is then time to select a particular pump model. A pump model should be selected by reviewing the application and determining which model pump will work best in the service.

Slurry Pump Break Down		
Light Slurries	Medium Slurries	Heavy Slurries
AF	AF	5000
HS	HS	5150
HSU	HSU	RX
HSUL	HSUL	CKX
VHS	VHS	5500
JC	JC	SRL-C
JCU	JCU	SRL-XT
VJC	VJC	
5100	5100	
5800 Linapump	5000	
	5150	
	RX	
	SP	
	SRL	
	CW	

NOTES:

The Model HS pump is a unique pump in that it is a recessed impeller or "vortex" pump. This style pump is well suited to handle light pulpy or fibrous slurries. The recessed impeller used in the HS family of pumps will pass large stringy fibers and should be considered when pump plugging is a concern.

The Model AF is a specialized pump with an axial flow design. This design of pump is built specifically for high flow, low head applications.

In general, slurry pumps have been designed to handle fluids with abrasive solids, and will give extended lives over standard water or process pumps. Although many features have been designed into the slurry pump, there are still two factors which directly relate to the pump's life that can be determined. The first choice to make is determining the metallurgy of the pump. In most cases, a hard metal slurry pump will be constructed of some hardened metal with a Brinell hardness of at least 500. Goulds standard slurry pump material is a 28% chrome iron with a minimum hardness of 600 Brinell. This material is used for most abrasive services and can also be used in some corrosive fluids as well. If a more corrosive resistant material is required, then the pump may be constructed out of a duplex Stainless steel Such as CD4MCu. Please check with your nearest Goulds sales office if you are unsure what material will be best suited for a particular application.

PUMP RUNNING SPEED

The other factor that can be controlled by the sales or end user engineer is the pump running speed. The running speed of a slurry pump is one of the most important factors which determines the life of the pump. Through testing, it has been proven that a slurry pump's wear rate is proportional to the speed of the pump raised to the 2 1/2 power.

EXAMPLE:

If Pump (A) is running at 1000 RPM and
Pump (B) is running at 800 RPM,

then the life factor for Pump (B) as compared to Pump (A) is (1000/800)^{2.5} or Pump (B) will last 1.75 times as long as Pump (A).

With the above ratio in mind, it can be shown that by cutting a slurry pump speed in half, you get approximately 6 times the wear life. For this reason, most slurry pumps are V-belt driven with a full diameter impeller. This allows the pump to run at the slowest possible running speed and, therefore, providing the maximum pump life.

WHY USE A V-BELT DRIVE?

In most ANSI pump applications it is a reasonable practice to control condition point by trimming the impeller and direct connecting the motor. However, this is not always sound practice in slurry applications. The abrasive solids present, wear life is enhanced by applying the pump at the slowest speed possible.

Another situation where V-belts are beneficial is in the application of axial flow pumps. Axial flow pumps cannot be trimmed to reduce the condition point because they depend on close clearances between the vane tips and the casing for their function. The generally low RPM range for axial flow application also makes it beneficial to use a speed reduction from the point of view of motor cost.

The types of V-belt drives available for use in pump applications are termed fixed speed, or fixed pitch, and variable speed. The fixed pitch drive consists of two sheaves; each machined to a specific diameter, and a number of belts between them to transmit the torque. The speed ratio is roughly equal to the diameter ratio of the sheaves. The variable speed drive is similar to the fixed speed except that the motor sheave can be adjusted to a range of effective or pitch diameters to achieve a band of speed ratios. This pitch adjustment is made by changing the width of the Vgrooves on the sheave. Variable speed drives are useful in applications where an exact flow rate is required or when the true condition point is not well defined at the time that the pump is picked.

V-belt drives can be applied up to about 2000 horsepower, but, pump applications are usually at or below 350 HP.

TECH-D-8C Solids and Slurries - Useful Formulas

a. The formula for specific gravity of a solids-liquids mixture or slurry, S_m is:

$$S_m = \frac{S_s \times S_1}{S_s + C_w (S_1 - S_s)}$$

where,

S_m = specific gravity of mixture or slurry
 S_1 = specific gravity of liquid phase
 S_s = specific gravity of solids phase
 C_w = concentration of solids by weight
 C_v = concentration of solids by volume

EXAMPLE: if the liquid has a specific gravity of 1.2 and the concentration of solids by weight is 35% with the solids having a specific gravity of 2.2, then:

$$S_m = \frac{2.2 \times 1.2}{2.2 + .35 (1.2 - 2.2)} = 1.43$$

b. Basic relationships among concentration and specific gravities of solid liquid mixtures are shown below:

In Terms of	S_s, S_m, S_1	C_v	C_w
C_v	$\frac{S_m - S_1}{S_s - S_1}$		$C_w \frac{S_m}{S_s}$
C_w	$\frac{(S_m - S_1)}{(S_s - S_1)} \times \frac{S_s}{S_m}$	$C_v \frac{S_s}{S_m}$	

Where pumps are to be applied to mixtures which are both corrosive and abrasive, the predominant factor causing wear should be identified and the materials of construction selected accordingly. This often results in a compromise and in many cases can only be decided as a result of test or operational experience.

For any slurry pump application a complete description of the mixture components is required in order to select the correct type of pump and materials of construction.

$$C_w = \frac{\text{weight of dry solids}}{\text{weight of dry solids} + \text{weight of liquid phase}}$$

$$C_v = \frac{\text{volume of dry solids}}{\text{volume of dry solids} + \text{volume of liquid phase}}$$

See nomograph for the relationship of concentration to specific gravity of dry solids in water shown in Fig. B.

c. Slurry flow requirements can be determined from the expression:

$$Q_m = \frac{4 \times \text{dry solids (tons per hour)}}{C_w = S_m}$$

where,

Q_m = slurry flow (U.S. gallons per minute)
 1 ton = 2000 lbs.

EXAMPLE: 2,400 tons of dry solids is processed in 24 hours in water with a specific gravity of 1.0 and the concentration of solids by weight is 30% with the solids having a specific gravity of 2.7 then:

$$S_m = \frac{2.7 \times 1.0}{2.7 + .3 (1-2.7)} = .123$$

$$Q_m = \frac{4 \times 100}{.3 \times 1.23} = 1,084 \text{ U.S. GPM}$$

d. Abrasive wear: Wear on metal pumps increases rapidly when the particle hardness exceeds that of the metal surfaces being abraded. If an elastomer lined pump cannot be selected, always select metals with a higher relative hardness to that of the particle hardness. There is little to be gained by increasing the hardness of the metal unless it can be made to exceed that of the particles. The effective abrasion resistance of any metal will depend on its position on the mohs or knoop hardness scale. The relationships of various common ore minerals and metals is shown in Fig. A.

Wear increases rapidly when the particle size increases. The life of the pump parts can be extended by choosing the correct materials of construction.

Sharp angular particles cause about twice the wear of rounded particles.

Austenitic manganese steel is used when pumping large dense solids where the impact is high.

Hard irons are used to resist erosion and, to a lesser extent, impact wear.

Castable ceramic materials have excellent resistance to cutting erosion but impeller tip velocities are usually restricted to 100 ft./sec.

Elastomer lined pumps offer the best wear life for slurries with solids under 1/4" for the SRL/SRL-C and under 1/2" for the SRL-XT.

Several Elastomers are available for different applications. Hypalon is acceptable in the range of 1-14 pH. There is a single stage head limitation of about 150' due to tip speed limitations of elastomer impellers.

See the Classification of Pumps according to Solids Size chart (Fig. C) and Elastomer Quick Selection Guide (Section TECH-B-2) for more information.

Solids and Slurries

Approximate Comparison of Hardness Values of Common Ores and Minerals

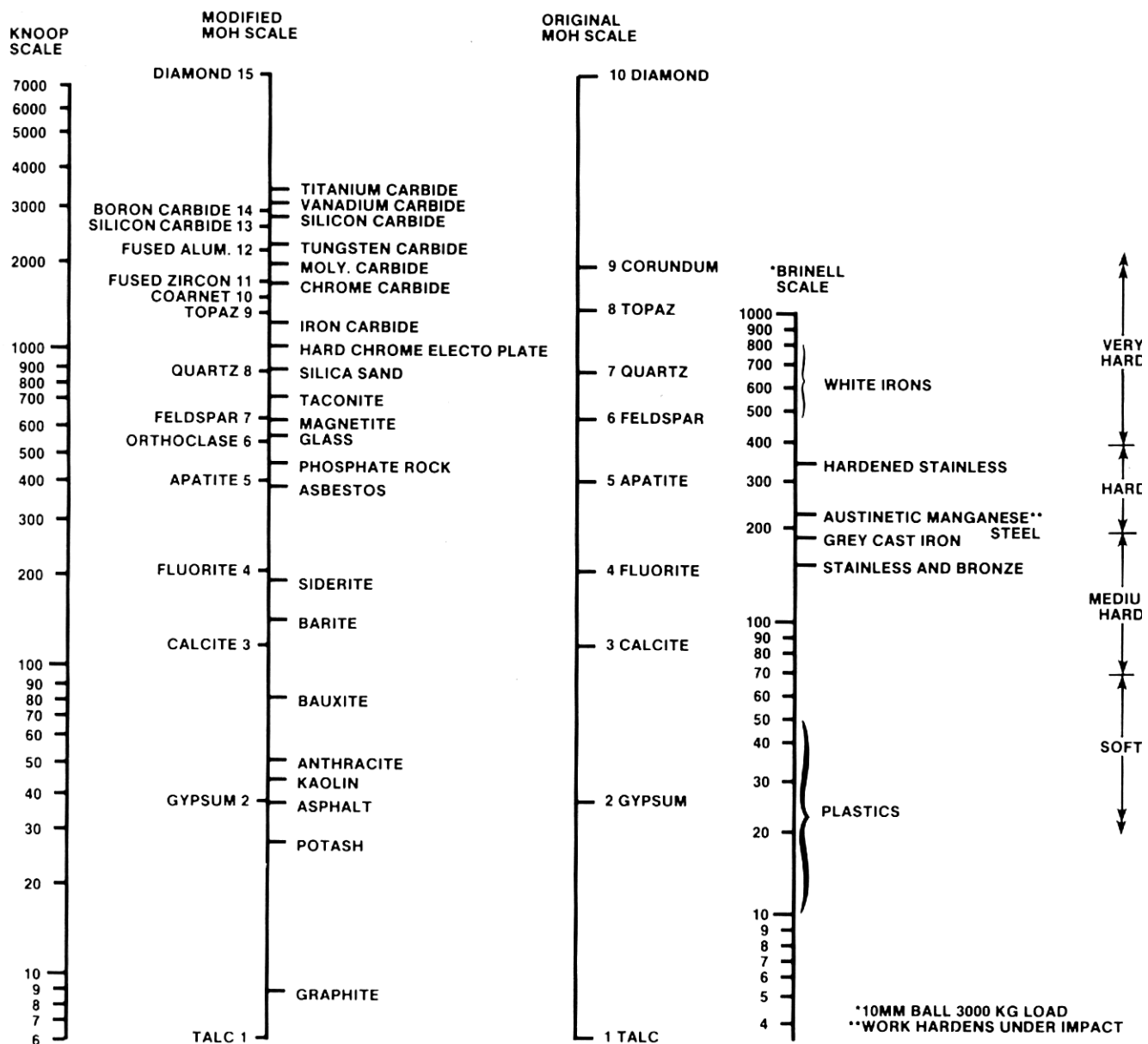


Fig. A

Solids and Slurries

Nomograph of the Relationship of Concentration to Specific Gravity in Aqueous Slurries

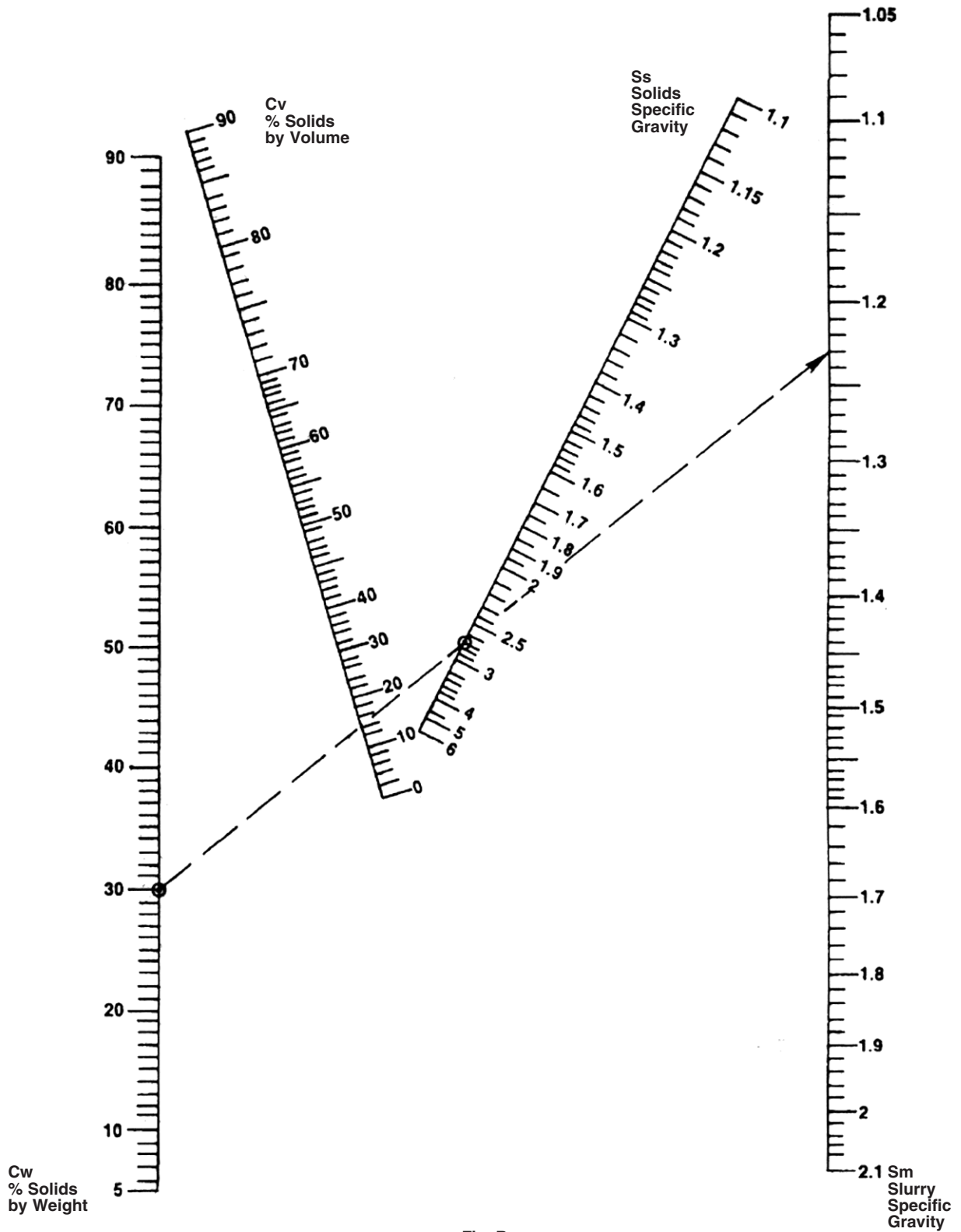
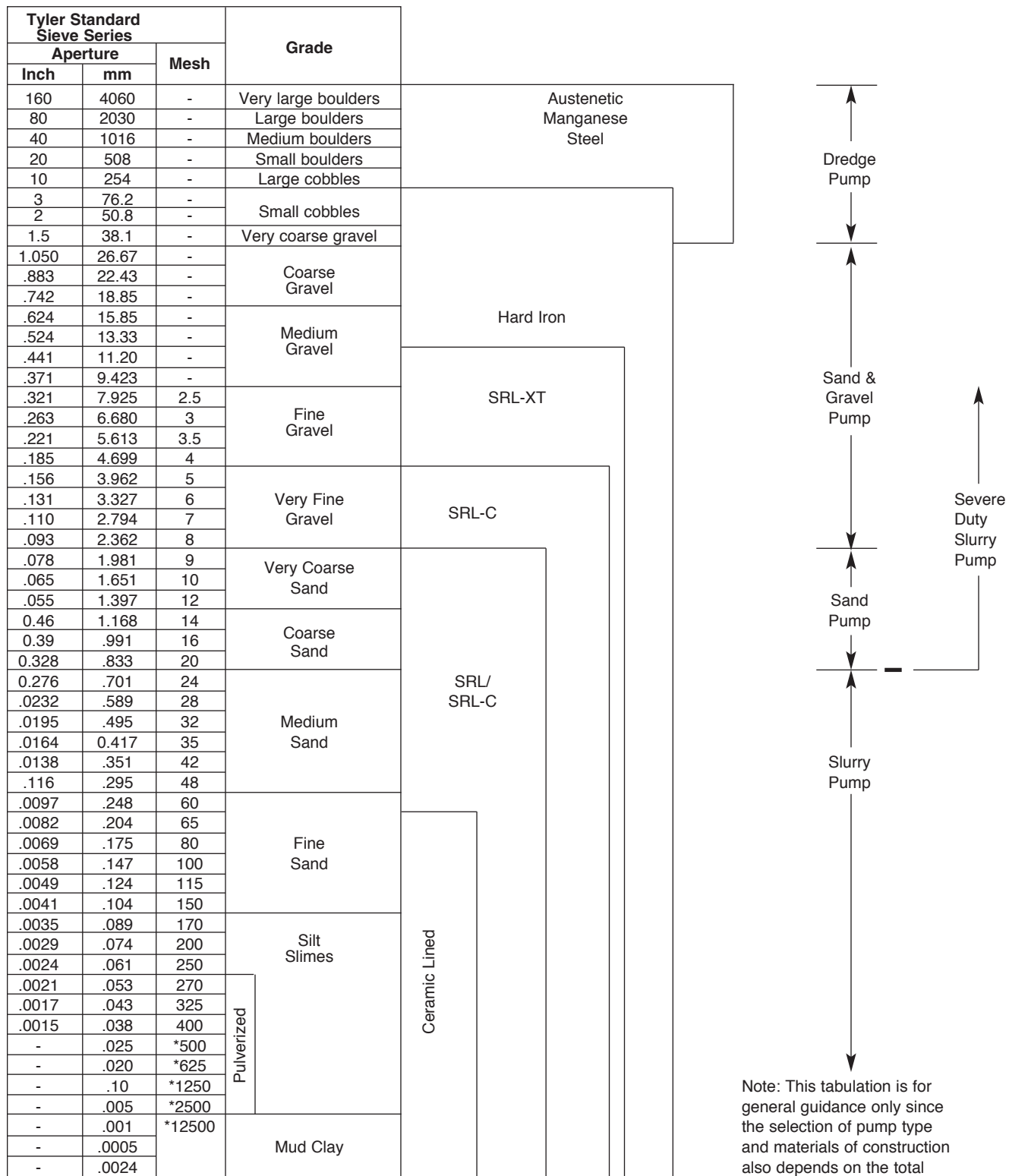


Fig. B

Solids and Slurries

Classification of Pumps According to Solid Size



* Theoretical values Micron = .001 mm

Fig. C

Solids and Slurries

Standard Screen Sizes Comparison Chart

U.S. Bureau of Standard Screens			Tyler Screens				British Standard Screens			I.M.M. Screens		
Mesh	Aperture		Mesh	Aperture		Mesh Double Tyler Series	Mesh	Aperture		Mesh	Aperture	
	Inches	mm		Inches	mm			Inches	mm		Inches	mm
			2½	.321	7.925							
3	.265	6.73	3	.263	6.680							
3½	.223	5.66		.221	5.613	3½						
4	.187	4.76	4	.185	4.699							
5	.157	4.00		.156	3.962	5						
6	.132	3.36	6	.131	3.327		5	.1320	3.34			
7	.111	2.83		.110	2.794	7	6	.1107	2.81			
8	.0937	2.38	8	.093	2.362		7	.0949	2.41	5	.100	2.54
10	.0787	2.00		.078	1.981	9	8	.0810	2.05			
12	.0661	1.68	10	.065	1.651		10	0.660	1.67			
										8	.062	1.574
14	.0555	1.41		.055	1.397	12	12	.0553	1.40			
										10	.050	1.270
16	.0469	1.19	14	.046	1.168		14	.0474	1.20			
										12	.0416	1.056
18	.0394	1.00		.039	.991	16	16	.0395	1.00			
20	.0331	.84	20	.0328	.883		18	.0336	.85			
										16	.0312	.792
25	.0280	.71		.0276	.701	24	22	.0275	.70			
										20	.025	.635
30	.0232	.59	28	.0232	.589		25	.0236	.60			
35	.0197	.50		.0195	.495	32	30	.0197	.50	25	.020	.508
40	.0165	.42	35	.0164	.417		36	.0166	.421	30	.0166	.421
45	.0138	.35		.0138	.351	42	44	.0139	.353	35	.0142	.361
										40	.0125	.317
50	.0117	.297	48	.0116	.295		52	.0166	.295			
60	.0098	.250		.0097	.246	60	60	.0099	.252	50	.01	.254
70	.0083	.210	65	.0082	.208		72	.0083	.211	60	.0083	.211
80	.0070	.177		.0069	.175	80	85	.0070	.177	70	.0071	.180
100	.0059	.149	100	.0058	.147		100	.0060	.152	80	.0062	.157
										90	.0055	.139
120	.0049	.125		.0049	.124	115	120	.0049	.125	100	.0050	.127
140	.0041	.105	150	.0041	.104		150	.0041	.105	120	.0042	.107
170	.0035	.088		.0035	.088	170	170	.0035	.088	150	.0033	.084
200	.0029	.074	200	.0029	.074		200	.0030	.076	170	.0029	.074
230	.0024	.062		.0024	.061	250	240	.0026	.065	200	.0025	.063
270	.0021	.053	270	.0021	.053		300	.0021	.053			
325	.0017	.044		.0017	.043	325						
			400	.0015	.037							

Fig. D

Solids and Slurries

Specific Gravities of Rocks, Minerals and Ores

Material	Specific Gravity	Mohs Hardness
Aluminum	2.55- 2.75	1-2
Amber	1.06-1.11	
Ambylgonite	3-3.1	5.5-6
Andesine	2.66- 2.94	6-6.5
Aragonite, CaCO ₃	2.94-2.95	3.5-4
Argentite	7.2-7.4	2-2.5
Asbestos	2.1-2.4	2
Asphaltum	1.1-1.5	
Asphalt Rock	2.41	
Barite	4.5	3-3.5
Basalt	2.4-3.1	8-9
Bauxite	2.55-2.73	
Bentonite	1.6	
Bertrandite	2.6	6
Beryl	2.66- 2.83	7.5-8
Biotite	2.7-3.1	2.5-3
Bone	1.7-2	
Borax	1.71-1.73	2-2.5
Bornite	5.06-5.08	3
Braggite	10	
Braunite	4.72- 4.83	6-6.5
Brick	1.4-2.2	
Calcite	2.72-2.94	3
Carnotite	2.47	1-2
Cassiterite	6.99-7.12	6-7
Carbon, Amorphous Graphitic	1.88-2.25	
Celluloid	1.4	
Cerussite	6.5- 6.57	3-3.5
Chalcocite	5.5-5.8	2.5-3
Chalcopyrite	4.1-4.3	3.5-4
Chalk	1.9-2.8	
Charcoal, Pine	0.28-0.44	
Charcoal, Oak	0.47-0.57	
Chromite	4.5	5.5
Chrysoberyl	3.65-3.85	8.5
Cinnabar	8.09	2-2.5
Clay	1.8-2.6	2
Coal, Anthracite	1.4-1.8	2
Coal, Bituminous	1.2-1.5	
Coal, Lignite	1.1-1.4	
Cobaltite	6.2	5.5
Coke	1-1.7	
Colemanite	1.73	4.5
Columbite	5.15-5.25	6
Copper	8.95	2.5-3
Cork	0.22-0.26	
Covellite	4.6-4.76	1.5-2
Cuprite	6	3.5-4
Diabase	2.94	
Diatomaceous Earth	0.4-0.72	
Diorite	2.86	
Dolomite	2.8-2.86	3.5-4
Enargite	4.4-4.5	3
Epidote	3.25-3.5	6
Feldspar	2.55-2.75	
Fluorite	3.18	4
Fly Ash	2.07	
Galena	7.3-7.6	2.5-2.75
Glass	2.4-2.8	7
Goethite	3.3-4.3	5-5.5
Gold	19.3	2.5-3
Granite	2.6-2.9	
Graphite	2.2-2.72	1-2
Gravel, Dry	1.55	4-5
Gravel, Wet	2	
Gypsum	2.3-2.37	2
Halite	2.2	2.5
Hausmannite	4.83-4.85	5.5
Helvite	3.2-3.44	6
Hematite	4.9-5.3	5-6

Material	Specific Gravity	Mohs Hardness
Hessite	8.24- 8.45	2-3
Ice	0.917	
Ilmenite	4.68-4.76	5-6
Iron, Slag	2.5-3	
Lepidolite	2.8-2.9	2.5-4
Lime, slaked	1.3- 1.4	
Limestone	2.4-2.7	2-5
Limonite	3.6-4	
Linnaeite	4.89	
Magnetite	4.9-5.2	5.5-6.5
Manganite	4.3-4.4	4
Marble	2.5-2.78	4
Marl	2.23	
Millerite	5.3-5.7	3-3.5
Monazite	5.1	5
Molybdenite	4.62-4.73	1-1.5
Muscovite	2.77- 2.88	2.5-3
Niccolite	7.784	5-5.5
Orpiment	3.5	1.5-2
Pentlandite	4.8	2.5-3
Petalite	2.412-2.422	6.5
Phosphite	3.21	
Phosphorus, white	1.83	
Polybasite	6-6.2	2.3
Porphyry		2.6-2.9
Potash	2.2	
Powellite	4.21-4.25	3.5-4
Proustite	5.57	2-2.5
Psilomelane	4.71	5-6
Pumice	0.37-0.9	
Pyragyrite	5.85	2.5
Pyrites	4.95-5.1	3.5-4.5
Pyrolusite	4.8	6-6.5
Quartz	2.5-2.8	7-8
Quartzite	2.68	7
Realgar	3.56	1.5-2
Rhodochrosite	3.7	3.5-4
Rhodonite	3.57-3.76	5.5-6.5
Rutile	4.2-5.5	6-6.5
Sand (see Quartz)	1.7-3.2	7
Sandstone	2-3.2	7
Scheelite	6.08-6.12	4.5-5
Schist	2.6-3	
Serpentine	2.5	2.5-3.5
Shale	1.6-2.9	
Siderite	3.9-4	4-4.5
Silica, fused trans.	2.21	
Slag, Furnace	2-3.9	
Slate	2.8-2.9	
Smaltite	6.48	
Soapstone, talc	2.6-2.8	2
Sodium Nitrate	2.2	
Sperrylite	10.58	6-7
Spodumene	3.03-3.22	6.5-7
Sphalerite	3.9-4.1	3.5-4
Stannite	4.3-4.5	4
Starch	1.53	
Stibnite	4.61-4.65	2
Sugar	1.59	
Sulfur	1.93-2.07	1.5-2.5
Sylvanite	8.161	1.5-2
Taconite	3.18	
Tallow, beef	0.94	
Tantalite	7.9-8	6.5
Tetrahedrite	4.6-5.1	3-4.5
Titanite	3.5	
Trap Rock	2.79	
Uraninite	8-11	5-6
Witherite	4.29-4.3	3.5
Wolframite	7.12-7.51	4-4.5
Zinc Blende	4.02	4
Zincite	5.64-5.68	4

Fig. E

Solids and Slurries

Hardness Conversion Table for Carbon and Alloy Steels

Brinell Hardness Number (Carbide Ball)	Rockwell Hardness Numbers					Tensile Strength	
	C Scale	A Scale	15N Scale Superficial	B Scale	30T Scale Superficial	ksi	MPa
	66	84.5	92.5				
722	64	83.4	91.8				
688	62	82.3	91.1				
654	60	81.2	90.2				
615	58	80.1	89.3				
577	56	79	88.3			313	2160
543	54	78	87.4			292	2010
512	52	76.8	86.4			273	1880
481	50	75.9	85.5			255	1760
455	48	74.7	84.5			238	1640
443	47	74.1	83.9			229	1580
432	46	73.6	83.5			221	1520
421	45	73.1	83			215	1480
409	44	72.5	82.5			208	1430
400	43	72	82			201	1390
309	42	71.5	81.5			194	1340
381	41	70.9	80.9			188	1300
371	40	70.4	80.4			182	1250
362	39	69.9	79.9			177	1220
353	38	69.4	79.4			171	1180
344	37	68.9	78.8			166	1140
336	36	68.4	78.3			161	1110
327	35	67.9	77.7			156	1080
319	34	67.4	77.2			152	1050
311	33	66.8	76.6			149	1030
301	32	66.3	76.1			146	1010
294	31	65.8	75.6			141	970
286	30	65.3	75			138	950
279	29	64.6	74.5			135	930
271	28	64.3	73.9			131	900
264	27	63.8	73.3			128	880
258	26	63.3	72.8			125	860
253	25	62.8	72.2			123	850
247	24	62.4	71.6			119	820
243	23	62	71			117	810
240				100	83.1	116	800
234				99	82.5	114	785
222				97	81.1	104	715
210				95	79.8	100	690
200				93	78.4	94	650
195				92	77.8	92	635
185				90	76.4	89	615
176				88	75.1	86	590
169				86	73.8	83	570

Fig. F

Solids and Slurries

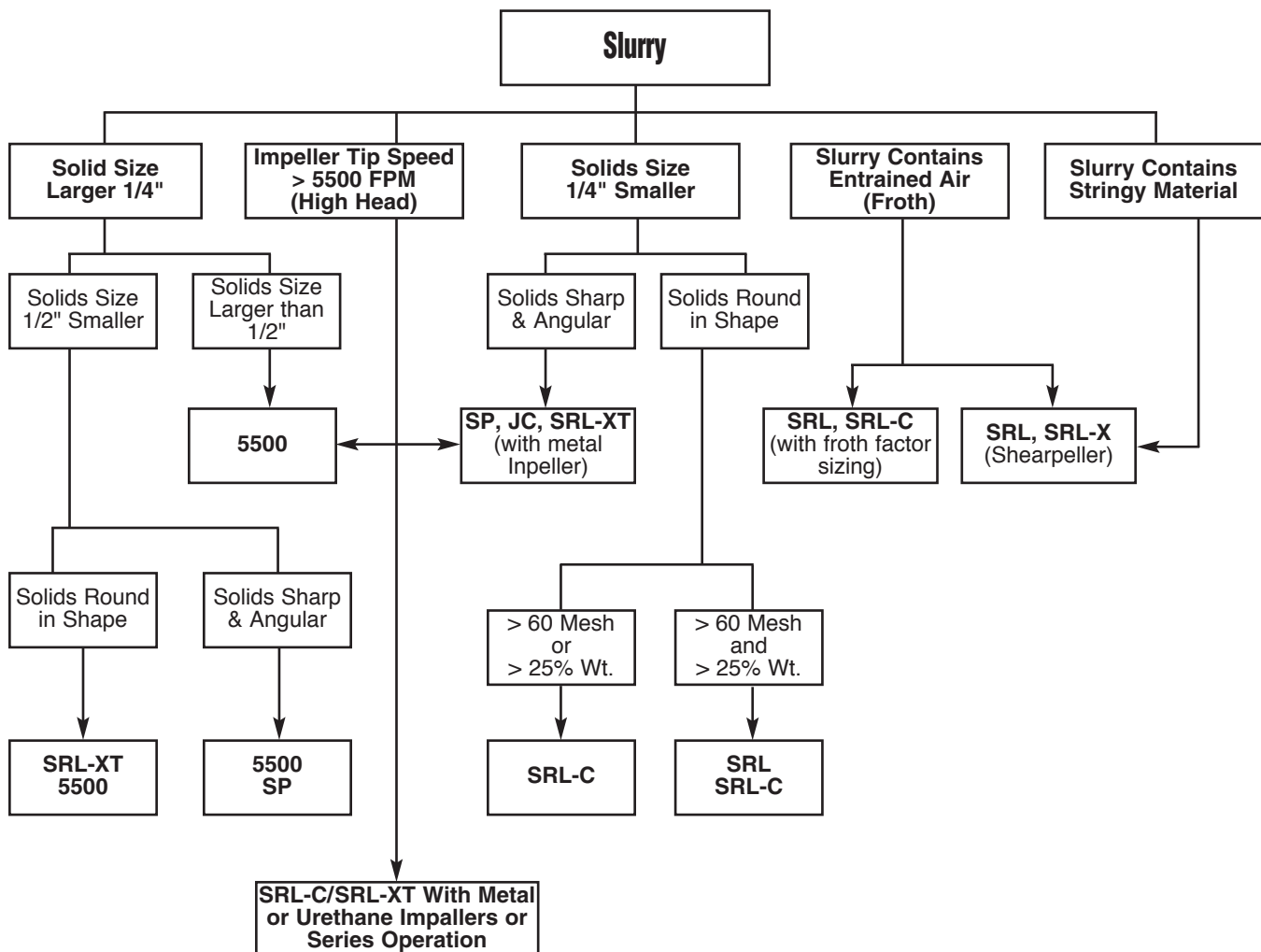
Slurry Pump Materials

MTL CODE	COMMON NAME	ASTM NUMBER	BRINELL HARDNESS	CHARACTERISTICS AND TYPICAL APPLICATIONS	pH RANGE
1002	Cast Iron	A48 Cl. 35B	196-228	Offers moderate resistance to abrasion and corrosion. It is suitable for light slurry applications, particularly those for intermittent service.	6-9
1228	HC600	A532 Cl. III Type A	550-650	Hardened HC600 (High Chromium Iron)	5-12
1245	316SS	A743 GR. CF-8M	159-190	Used for high corrosive, mildly abrasive applications.	3-11
1247	CD4MCu	A734 Gr. CD4MCu	224-325	This is a high strength corrosion resistant alloy for mildly abrasive applications.	

MTL CODE	PRINCIPAL ALLOYING ELEMENTS (% Bal Fe)						
	Cr	Ni	C	Mn	Si	Mo	Others
1002	-	-	3.25-3.35	0.45-0.70	1.70-1.90	-	-
1228	23.0-28.0	15 Max	2.3-3.0	0.5-1.5	1.0 Max	1.5 Max	-
1245	18.0-21.0	9.0-12.0	0.08 Max	1.5 Max	2.0 Max	2.0-3.0	-
1247	25.0-27.0	5.0-6.0	0.4 Max	-	-	2.0	Cu 3.0

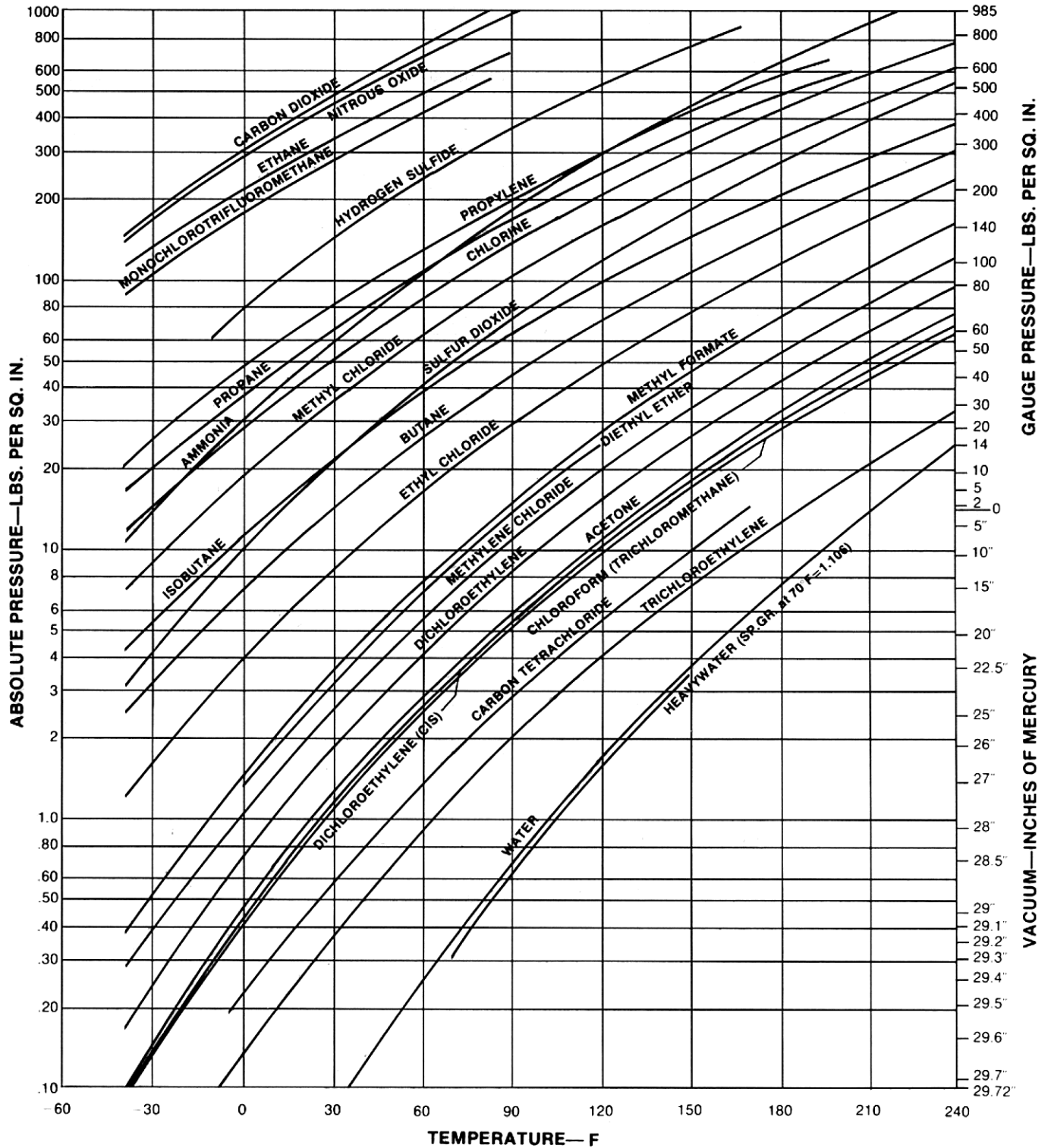
Fig. G

Slurry Pump Application Guidelines



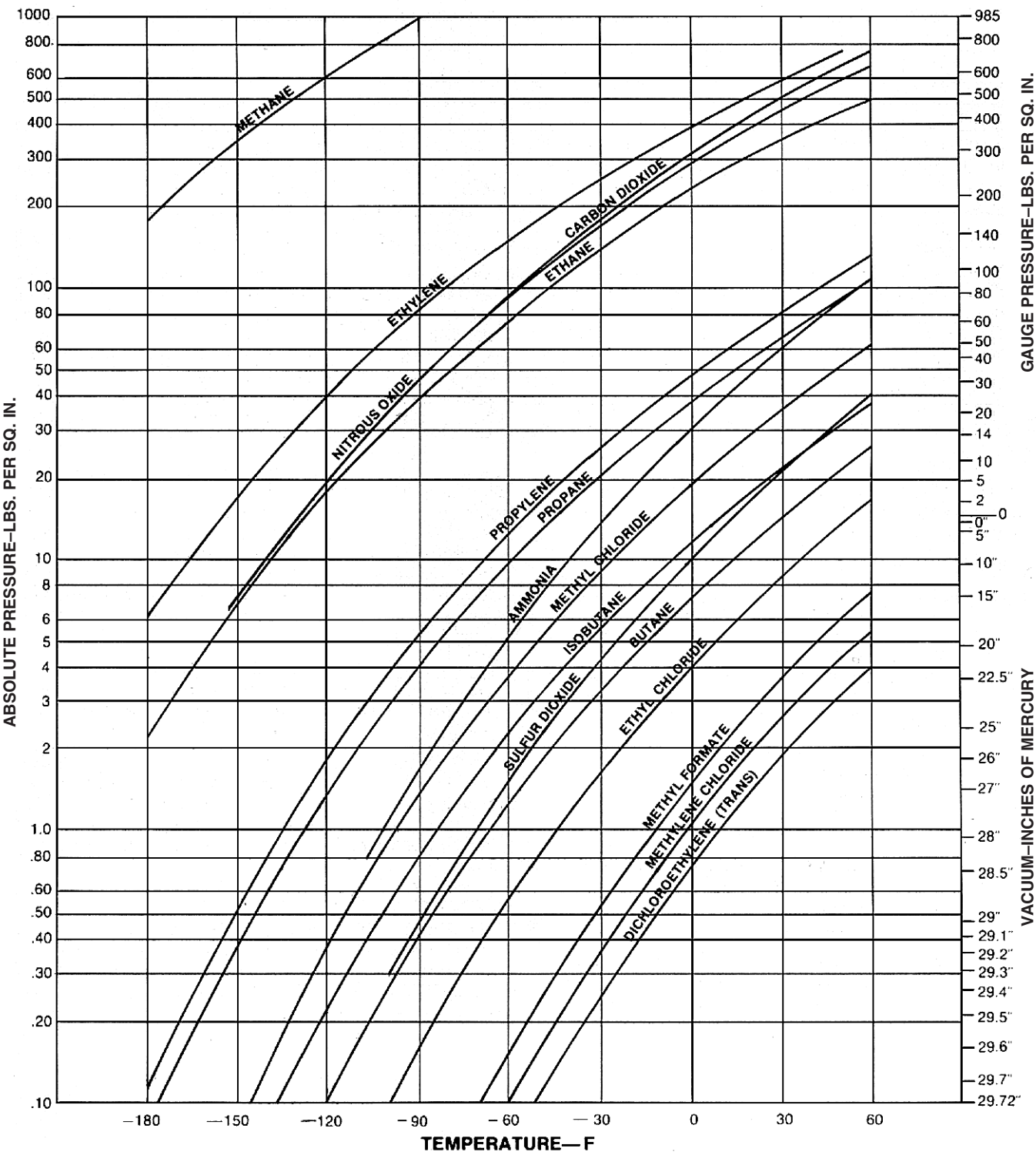
TECH-D-9A Vapor Pressure – Various Liquids

Chart 1 –60° to 240°F



TECH-D-9A Vapor Pressure – Various Liquids

Chart 2 –180° to 60°F





Section TECH-E

Paper Stock

TECH-E-1 Paper Stock, Discussion

Centrifugal pumps are used with complete success in handling paper stock and other fibrous suspensions. However, the nature of a stock suspension requires certain special considerations. All of the factors affecting pump operation discussed below must be carefully considered for a good installation.

SUCTION PIPING

The stock must be delivered freely to the impeller for the pump to operate. The suction pipe should be as short and direct as possible. The suction pipe and entrance from the stock chest should never be smaller than the pump suction connection, and should be level with no air pockets. Always keep the direction of flow in a straight line.

Inadequate suction design with undersize pipe and excessive fittings can prevent the pump from delivering rated capacity, or from operating at all on high consistency stocks.

SUCTION HEAD

Stock pumps will not operate when a vacuum is required to maintain flow into the pump. Thus, there must be a static suction head sufficient to overcome suction line friction losses.

PERCENT CONSISTENCY

The consistency of a pulp and water suspension is the percent by weight of pulp in the mixture. Oven Dry (O.D.) consistency is the amount of pulp left in a sample after drying in an oven at 212°F. Air Dry (A.D.) consistency is an arbitrary convention used by paper-makers, and is the amount of pulp left in a sample after drying in atmosphere. Air Dry stock contains 10% more moisture than Bone Dry stock, i.e. 6% O.D. is 6.67% A.D.

Traditional paper stock pumps will handle stock up to approximately 6% O.D. consistency. The absolute maximum limit is a function of many factors including stock fiber length, pulping process, degree of refining, available suction head, etc. In certain situations, consistencies as high as 8% O.D. can be successfully handled with a standard paper stock pump.

Recent testing on various types of stock has indicated that pump performance is the same as on water for stock consistencies up to 6% O.D. In other words, water curves can be used to select stock pumps, as the capacity, head and efficiency are the same as for water.

Medium consistency paper stock is a term generally used to describe stock between 7% and 15% O.D. consistency. Pumping of medium consistency paper stock with a centrifugal pump is possible, but requires a special design due to the fiber network strength and the inherently high air content.

AIR IN STOCK

Entrained air is detrimental to good operation of any centrifugal pump, and can result in reduced capacity, increased erosion and shaft breakage. Obviously every effort must be made to prevent the over-entrainment of air throughout the process.

EXCESSIVE DISCHARGE THROTTLING

While it is realized that excess capacity is normally required over the paper machine output in tons per day, "over-selection" of pumps on the basis of capacity and head usually results in the necessity of throttling the pump at the valve in the discharge line. Since the valve is normally located adjacent to the pump, the restriction of the valve and the high velocity within the valve will result in some dehydration and cause vibration due to slugs of stock. Vibration at the valve due to throttling is transmitted to the pump and may reduce the normal life of the pump-rotating element.

Centrifugal pumps operating at greatly reduced capacity have more severe loading internally due to hydraulic radial thrust. Hence pumps selected too greatly oversize in both capacity and head have the combination of the vibration due to throttling plus the greater internal radial load acting to reduce the life of the rotating element. As a general rule, stock pumps should not be operated for extended periods at less than one quarter of their capacity at maximum efficiency. When excessive throttling is required, one of the two methods below should be employed.

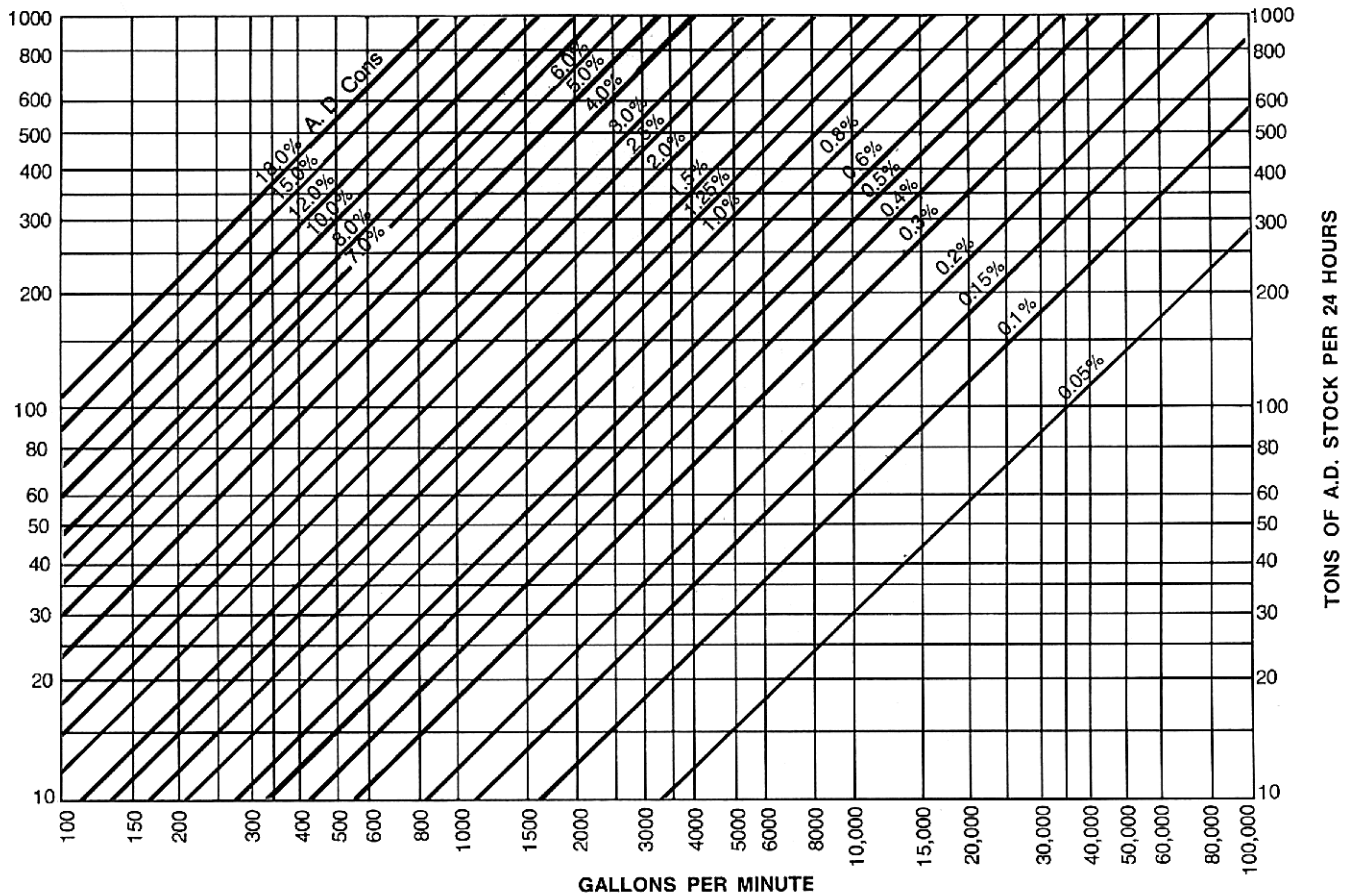
1. Review capacity requirements and check the static and friction head required for the capacity desired. Reduce the impeller diameter to meet the maximum operating conditions. This will also result in considerable power saving.
2. Install a by-pass line upstream from the discharge valve back to the suction chest below the minimum chest level, if possible, and at a point opposite the chest opening to the pump suction. This by-pass line should include a valve for flow regulation. This method is suggested where mill production includes variation in weight of sheet.

FILLERS AND ADDITIVES

The presence of fillers and chemical additives such as clay, size and caustics can materially increase the ability of paper stock to remain in suspension. However, overdosing with additives such as alum may cause gas formation on the stock fibers resulting in interruption of pumping.

TECH-E-2 Conversion Chart of Mill Output in Tons per 24 Hours

To U.S. Gallons per Minute of Paper Stock of Various Densities



EXAMPLE:

Find the capacity in gallons per minute of a pump handling 4% stock for a mill producing 200 tons per 24 hours.

Enter chart at 200 tons per day, read horizontally to 4% stock, then downward to find pump capacity of 840 GPM.

TECH-E-2.1 Definitions / Conversion Factors

A.D. = Air Dry stock (Contains 10% Water)

O.D. = Oven Dry stock (All Water Removed)
Also Called Bone Dry (B.D.)

A.D. = 1.11 x O.D.

O.D. = 0.90 x A.D.

A.D. = 1.11 O.D. T/D

O.D. = 0.90 x A.D. T/D

A.D. Consistency = 1.11 x O.D. Consistency

O.D. Consistency = 0.90 x A.D. Consistency

T/D or TPD or S. T/D = Short Tons Per Day

One Short Ton = 2000 lbs.

M. T/D = Metric Tons per Day

One Metric Ton = 2205 lbs.

A.D.S. T/D = Air Dry Short Tons/Day

A.D.M. T/D = Air Dry Metric Tons/Day

S. T/D = 1.1025 x M. T/D

$\frac{\text{Production in A. D. S. T/D} \times 15}{\% \text{ O.D. Cons.}} = \text{Flow in GPM}$

$\frac{\text{Production in A. D. S. T/D} \times 16.67}{\% \text{ A.D. Cons.}} = \text{Flow in GPM}$

TECH-E-3 Friction Loss of Pulp Suspensions in Pipe

I. INTRODUCTION

In any stock piping system, the pump provides flow and develops hydraulic pressure (head) to overcome the differential in head between two points. This total head differential consists of pressure head, static head, velocity head and total friction head produced by friction between the pulp suspension and the pipe, bends, and fittings. The total friction head is the most difficult to determine because of the complex, nonlinear nature of the friction loss curve. This curve can be affected by many factors.

The following analytical method for determining pipe friction loss is based on the recently published TAPPI Technical Information Sheet

(TIS) 408-4 (Reference 1), and is applicable to stock consistencies (oven-dried) from 2 to 6 percent. Normally, stock consistencies of less than 2% (oven-dried) are considered to have the same friction loss characteristic as water.

The friction loss of pulp suspensions in pipe, as presented here, is intended to supersede the various methods previously issued.

II. BACKGROUND

Figure 1 and Figure 2 show typical friction loss curves for two different consistencies ($C_2 > C_1$) of chemical pulp and mechanical pulp, respectively.

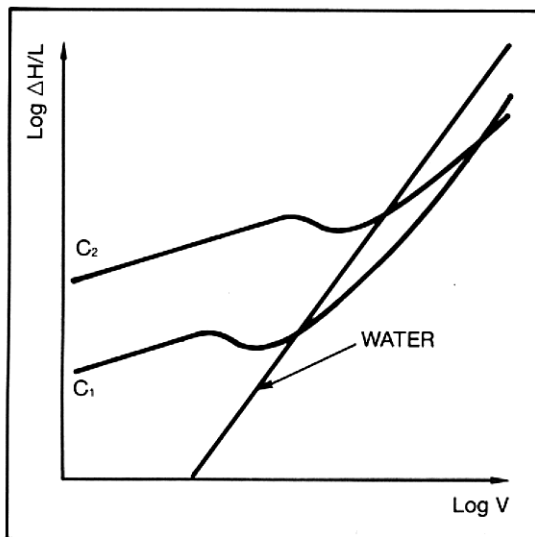


Figure 1 – Friction loss curves for chemical pulp ($C_2 > C_1$).

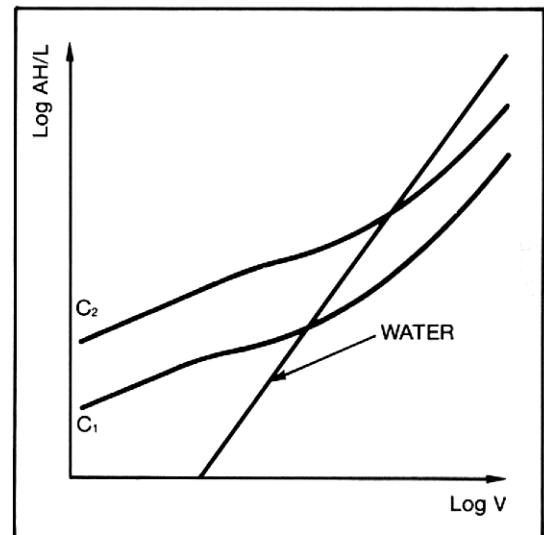


Figure 2 – Friction loss curves for mechanical pulp ($C_2 > C_1$).

The friction loss curve for chemical pulp can be conveniently divided into three regions, as illustrated by the shaded areas of Figure 3.

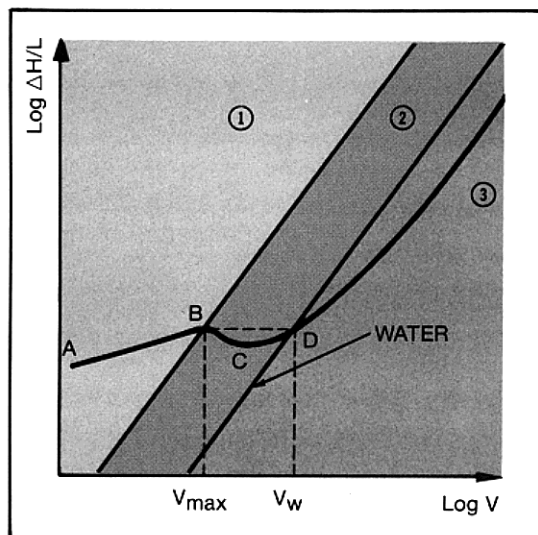


Figure 3 – Friction loss curves for chemical pulp, shaded to show individual regions.

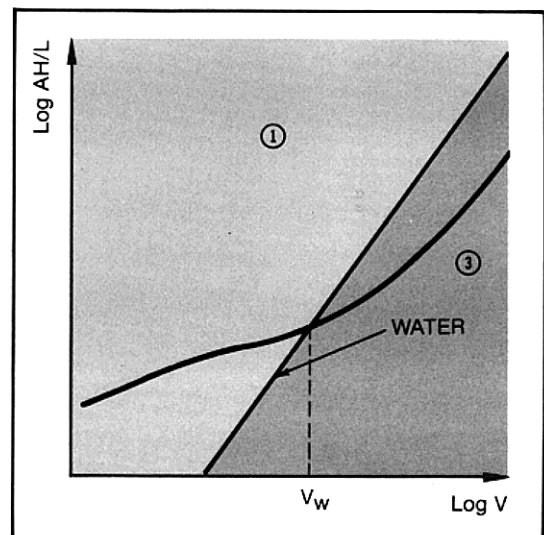


Figure 4 – Friction loss curves for mechanical pulp, shaded to show individual regions.

These regions may be described as follows:

Region 1 (Curve AB) is a linear region where friction loss for a given pulp is a function of consistency, velocity, and pipe diameter. The velocity at the upper limit of this linear region (Point B) is designated V_{max} .

Region 2 (Curve BCD) shows an initial decrease in friction loss (to Point C) after which the friction loss again increases. The intersection of the pulp friction loss curve and the water friction loss curve (Point D) is termed the onset of drag reduction. The velocity at this point is designated V_w .

Region 3 (Curve DE) shows the friction loss curve for pulp fiber suspensions below the water curve. This is due to a phenomenon called drag reduction. Reference 2 describes the mechanisms which occur in this region.

Regions 2 and 3 are separated by the friction loss curve for water, which is a straight line with a slope approximately equal to 2.

The friction loss curve for mechanical pulp, as illustrated in Figure 4, is divided into only two regions:

Regions 1 and 3. For this pulp type, the friction loss curve crosses the water curve at V_w and there is no true V_{max} .

III. DESIGN PARAMETERS

To determine the pipe friction loss component for a specified design basis (usually daily mass flow rate), the following parameters must be defined:

- Pulp Type** - Chemical or mechanical pulp, long or short fibered, never dried or dried and reslurried, etc. This is required to choose the proper coefficients which define the pulp friction curve.
- Consistency, C (oven-dried)** - Often a design constraint in an existing system. NOTE: If air-dried consistency is known, multiply by 0.9 to convert to oven-dried consistency.
- Internal pipe diameter, D** - Lowering D reduces initial capital investment, but increases pump operating costs. Once the pipe diameter is selected, it fixes the velocity for a prespecified mass flow rate.
- Bulk velocity, V** - Usually based on a prespecified daily mass flow rate. Note that both V and D are interdependent for a constant mass flow rate.
- Stock temperature, T** - Required to adjust for the effect of changes in viscosity of water (the suspending medium) on pipe friction loss.
- Freeness** - Used to indicate the degree of refining or to define the pulp for comparison purposes.
- Pipe material** - Important to specify design correlations and compare design values.

IV. PIPE FRICTION ESTIMATION PROCEDURE

The bulk velocity (V) will depend on the daily mass flow rate and the pipe diameter (D) selected. The final value of V can be optimized to give the lowest capital investment and operating cost with due consideration of future demands or possible system expansion.

The bulk velocity will fall into one of the regions previously discussed. Once it has been determined in which region the design velocity will occur, the appropriate correlations for determining pipe friction loss value(s) may be selected. The following describes the procedure to be used for estimating pipe friction loss in each of the regions.

Region 1 The upper limit of Region 1 in Figure 3 (Point B) is designated V_{max} . The value of V_{max} is determined using Equation [1] and data given in Table I or IA.

$$V_{max} = K' C^\sigma \quad (\text{ft/s}), \quad [1]$$

where K' = numerical coefficient (constant for a given pulp is attained from Table I or IA.

C = consistency (oven-dried, expressed as a percentage, not decimally), and

σ = exponent (constant for a given pulp), obtained from Table I or IA.

If the proposed design velocity (V) is less than V_{max} , the value of flow resistance ($\Delta H/L$) may be calculated using Equation [2] and data given in Table II or IIA, and the appendices.

$$H/L = F K V^\alpha C^\beta D^y \quad (\text{ft/100 ft}), \quad [2]$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D),

K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA,

V = bulk velocity (ft/s),

C = consistency (oven-dried, expressed as a percentage, not decimally),

D = pipe inside diameter (in), and

α, β, y = exponents (constant for a given pulp), obtained from Table II or IIA.

For mechanical pumps, there is no true V_{max} . The upper limit of the correlation equation (Equation [2]) is also given by Equation [1]. In this case, the upper velocity is actually V_w .

Region 2 The lower limit of Region 2 in Figure 3 (Point B) is V_{max} and the upper limit (Point D) is V_w . The velocity of the stock at the onset of drag reduction is determined using Equation [3]

$$V_w = 4.00 C^{1.40} \quad (\text{ft/s}), \quad [3]$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

If V is between V_{max} and V_w , Equation 2 may be used to determine $\Delta H/L$ at the maximum point (V_{max}). Because the system must cope with the worst flow condition, $\Delta H/L$ at the maximum point (V_{max}) can be used for all design velocities between V_{max} and V_w .

Region 3 A conservative estimate of friction loss is obtained by using the water curve. $(\Delta H/L)_w$ can be obtained from a Friction Factor vs. Reynolds Number plot (Reference 3, for example), or approximated from the following equation (based on the Blasius equation).

$$(\Delta H/L)_w = 0.58 \cdot V^{1.75} D^{-1.25} \text{ (ft/100 ft)}, \quad [4]$$

where V = bulk velocity (ft/s), and

D = pipe diameter (in).

Previously published methods for calculating pipe friction loss of pulp suspensions gave a very conservative estimate of head loss. The method just described gives a more accurate estimate of head loss due to friction, and has been used successfully in systems in North America and world-wide.

Please refer to Appendix A for equivalent equations for use with metric (SI) units. Tables I and IA are located in Appendix B; Tables II and IIA are located in Appendix C. Pertinent equations, in addition to those herein presented, are located in Appendix D. Example problems are located in Appendix E.

V. HEAD LOSSES IN BENDS AND FITTINGS

The friction head loss of pulp suspensions in bends and fittings may be determined from the basic equation for head loss, Equation [5].

$$H = K V_1^2 / 2g \text{ (ft)}, \quad [5]$$

where K = loss coefficient for a given fitting,

V_1 = inlet velocity (ft/s), and

g = acceleration due to gravity (32.2 ft/s²).

Values of K for the flow of water through various types of bends and fittings are tabulated in numerous reference sources (Reference 3, for example). The loss coefficient for valves may be obtained from the valve manufacturer.

The loss coefficient for pulp suspensions in a given bend or fitting generally exceeds the loss coefficient for water in the same bend or fitting. As an approximate rule, the loss coefficient (K) increases 20 percent for each 1 percent increase in oven-dried stock consistency. Please note that this is an approximation; actual values of K may differ, depending on the type of bend or fitting under consideration (4).

APPENDIX A

When metric (SI) units are utilized, the following replace the corresponding equations in the main text.

$$V_{\max} = K' C^\sigma \text{ (m/s)} \quad [1M]$$

where K = numerical coefficient (constant for a given pulp), obtained from Table I or IA,

C = consistency (oven-dried, expressed as a percentage, not decimally), and

σ = exponent (constant for a given pulp), obtained from Table I or IA.

$$\Delta H/L = F K V^\alpha C^\beta D^y \text{ (m/100m)}, \quad [2M]$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D),

K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA,

V = bulk velocity (m/s),

C = consistency (oven-dried, expressed as a percentage, not decimally),

D = pipe inside diameter (mm), and

α, β, y = exponents (constant for a given pulp), obtained from Table II or IIA.

$$V_W = 1.22 C^{1.40} \text{ (m/s)}, \quad [3M]$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

$$(\Delta H/L)_w = 264 V^{1.75} D^{-1.25} \text{ (m/100m)}, \quad [4M]$$

where V = bulk velocity (m/s), and

D = pipe inside diameter (mm).

$$H = K V_1^2 / 2g \text{ (m)}, \quad [5M]$$

where K = loss coefficient for a given fitting,

V_1 = inlet velocity (m/s), and

g = acceleration due to gravity (9.81 m/s²).

APPENDIX B

TABLE I

Data for use with Equation $\boxed{1}$ or Equation $\boxed{1M}$ to determine velocity limit, $V_{\max}$ (1).

Pulp Type	Pipe Material	K'	σ
Unbeaten aspen sulfite never dried	Stainless Steel	0.85 (0.26)	1.6
Long fibered kraft never dried CSF = 725 (6)	PVC	0.98 (0.3)	1.85
	Stainless Steel	0.89 (0.27)	1.5
Long fibered kraft never dried CSF = 650 (6)	PVC	0.85 (0.26)	1.9
Long fibered kraft never dried CSF = 550 (6)	PVC	0.75 (0.23)	1.65
Long fibered kraft never dried CSF = 260 (6)	PVC	0.75 (0.23)	1.8
Bleached kraft never dried and reslurried (6)	PVC	0.79 (0.24)	1.5
	Stainless Steel	0.59 (0.18)	1.45
Long fibered kraft dried and reslurried (6)	PVC	0.49 (0.15)	1.8
Kraft birch dried and reslurried (6)	PVC	0.69 (0.21)	1.3
Stone groundwood CSF = 114	PVC	4.0 (1.22)	1.40
Refiner groundwood CSF = 150	PVC	4.0 (1.22)	1.40
Newsprint broke CSF = 75	PVC	4.0 (1.22)	1.40
Refiner groundwood (hardboard)	PVC	4.0 (1.22)	1.40
Refiner groundwood (insulating board)	PVC	4.0 (1.22)	1.40
Hardwood NSSC CSF = 620	PVC	0.59 (0.18)	1.8

- NOTES:**
1. When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimetres (mm) and velocity (V) in metres per second (m/s).
 2. Original data obtained in stainless steel and PVC pipe. PVC is taken to be hydraulically smooth pipe.
 3. Stainless steel may be hydraulically smooth although some manufacturing processes may destroy the surface and hydraulic smoothness is lost.
 4. For cast iron and galvanized pipe, the K' values will be reduced. No systematic data are available for the effects of surface roughness.
 5. It pulps are not identical to those shown, some engineering judgement is required.
 6. Wood is New Zealand Kraft pulp.

TABLE IA

Data (5, 6) for use with Equation $\boxed{1}$ or Equation $\boxed{1M}$ to determine velocity limit, $V_{\max}$.

Pulp Type (5)	Pipe Material	K'	σ
Unbleached sulphite	Copper	0.98 (0.3)	1.2
Bleached sulphite	Copper	0.98 (0.3)	1.2
Kraft	Copper	0.98 (0.3)	1.2
Bleached straw	Copper	0.98 (0.3)	1.2
Unbleached straw	Copper	0.98 (0.3)	1.2

Estimates for other pulps based on published literature.

Pulp Type (5, 6)	Pipe Material	K'	σ
Cooked groundwood	Copper	0.75 (0.23)	1.8
Soda	Steel	4.0 (1.22)	1.4

NOTE: When metric (SI) units are utilized, use the value of K' given in parentheses, When the metric values are used, diameter (D) must be millimeters (mm) and velocity (V) in meters per second (m/s)

APPENDIX C

TABLE II
Data for use with Equation $\boxed{2}$ or Equation $\boxed{2M}$ to determine head loss, $\Delta H/L$ (1).

Pulp Type	K	α	β	y
Unbeaten aspen sulfite never dried	5.30 (235)	0.36	2.14	-1.04
Long fibered kraft never dried CSF = 725 (5)	11.80 (1301)	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 650 (5)	11.30 (1246)	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 550 (5)	12.10 (1334)	0.31	1.81	-1.34
Long fibered kraft never dried CSF = 260 (5)	17.00 (1874)	0.31	1.81	-1.34
Bleached kraft bleached and reslurred (5)	8.80 (970)	0.31	1.81	-1.34
Long fibered kraft dried and reslurred (5)	9.40 (1036)	0.31	1.81	-1.34
Kraft birch dried and reslurred (5)	5.20 (236)	0.27	1.78	-1.08
Stone groundwood CSF = 114	3.81 (82)	0.27	2.37	-0.85
Refiner groundwood CSF = 150	3.40 (143)	0.18	2.34	-1.09
Newspaper broke CSF = 75	5.19 (113)	0.36	1.91	-0.82
Refiner groundwood CSF (hardboard)	2.30 (196)	0.23	2.21	-1.29
Refiner groundwood CSF (insulating board)	1.40 (87)	0.32	2.19	-1.16
Hardwood NSSF CSF = 620	4.56 (369)	0.43	2.31	-1.20

- NOTES:**
1. When metric (SI) units are utilized, use the value of K given in parentheses. When the metric values are used, diameter (D) must be in millimetres (mm) and velocity must be in metres per second (m/s).
 2. Original data obtained in stainless steel and PVC pipe (7,8, 9).
 3. No safety factors are included in the above correlations.
 4. The friction loss depends considerably on the condition of the inside of the pipe surface (10).
 5. Wood is New Zealand Kraft pulp.

TABLE IA
Data (5, 6) for use with Equation $\boxed{2}$ or Equation $\boxed{2M}$ to determine head loss, $\Delta H/L$.

Pulp Type (5)	K	α	β	y
Unbleached sulfite	12.69 (1438)	0.36	1.89	-1.33
Bleached sulfite	11.40 (1291)	0.36	1.89	-1.33
Kraft	1140 (1291)	0.36	1.89	-1.33
Bleached straw	11.40 (1291)	0.36	1.89	-1.33
Unbleached straw	5.70 (646)	0.36	1.89	-1.33

Estimates for other pulps based on published literature.

Pulp Type (5, 6)	K	α	β	y
Cooked groundwood	6.20 (501)	0.43	2.13	-1.20
Soda	6.50 (288)	0.36	1.85	-1.04

NOTE: When metric (SI) units are utilized, use the value of K given in parentheses, When the metric values are used, diameter (D) must be millimeters (mm) and velocity (V) in meters per second (m/s)

APPENDIX D

The following gives supplemental information to that where I.P.D. mill capacity (metric tons per day), provided in the main text.

1. Capacity (flow), Q —

$$Q = \frac{16.65 (\text{T.P.D.})}{C} \quad (\text{U.S. GPM}), \quad (\text{i})$$

Where T.P.D. = mill capacity (short tons per day), and
C = consistency (oven-dried, expressed as a percentage, not decimally).

If SI units are used, the following would apply:

$$Q = \frac{1.157 (10^{-3}) (\text{T.P.D.})}{C} \quad (\text{m}^3/\text{s}), \quad (\text{IM})$$

Where T.P.D. = mill capacity (metric tons per day), and
C = consistency (oven-dried, expressed as a percentage, not decimally).

2. Bulk velocity, V —

$$V = \frac{0.321 Q}{A} \quad (\text{ft/s}), \text{ or} \quad (\text{ii})$$

$$V = \frac{0.4085 Q}{D^2} \quad (\text{ft/s}), \quad (\text{ii})$$

Where Q = capacity (U.S. GPM)
A = inside area of pipe (in²), and
D = inside diameter of pipe (in)

The following would apply if SI units are used:

$$V = \frac{1 (10^6) Q}{A} \quad (\text{m/s}), \text{ or} \quad (\text{iiM})$$

$$V = \frac{1.273 (10^6) Q}{D^2} \quad (\text{m/s}), \quad (\text{iiM})$$

Where Q = capacity (m³/s),

A = inside area of pipe (mm²), and

D = inside diameter of pipe (mm)

3. Multiplication Factor, F (included in Equation 2) -

$$F = F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5, \quad (\text{iv})$$

where F_1 = correction factor for temperature. Friction loss calculations are normally based on a reference pulp temperature of 95° F (35°C). The flow resistance may be increased or decreased by 1 percent for each 1.8°F (1°C) below or above 95°F (35°C), respectively. This may be expressed as follows:

$$F_1 = 1.528 - 0.00556 T, \quad (\text{v})$$

where T = pulp temperature (° F), or

$$F_1 = 1.35 - 0.01 T, \quad (\text{vM})$$

where T = pulp temperature (°C).

F_2 = correction factor for pipe roughness. This factor may vary due to manufacturing processes of the piping, surface roughness, age, etc. Typical values for PVC and stainless steel piping are listed below:

F_2 = 1.0 for PVC piping,

F_2 = 1.25 for stainless steel piping.

Please note that the above are typical values; experience and/or additional data may modify the above factors.

F_3 = correction factor for pulp type. Typical values are listed below:

F_3 = 1.0 for pulps that have never been dried and reslurried,

F_3 = 0.8 for pulps that have been dried and reslurried. Note: This factor has been incorporated in the numerical coefficient, K, for the pulps listed in Table II. When using Table II, F_3 should not be used.

F_4 = correction factor for beating. Data have shown that progressive beating causes, initially, a small decrease in friction loss, followed by a substantial increase. For a kraft pine pulp initially at 725 CSF and F_4 = 1.0, beating caused the freeness to decrease to 636 CSF and F_4 to decrease to 0.96. Progressive beating decreased the freeness to 300 CSF and increased F_4 to 1.37 (see K values in Table II). Some engineering judgement may be required.

F_5 = design safety factor. This is usually specified by company policy with consideration given to future requirements.

APPENDIX E

The following are three examples which illustrate the method for determination of pipe friction loss in each of the three regions shown in Figure 3.

Example 1.

Determine the friction loss (per 100 ft of pipe) for 1000 U.S. GPM of 4.5% oven-dried unbeaten aspen sulfite stock, never dried, in 8 inch schedule 40 stainless steel pipe (pipe inside diameter = 7.981 in). Assume the pulp temperature to be 95° F.

Solution:

a) The bulk velocity, V, is

$$V = \frac{0.4085 Q}{D^2}, \quad (\text{ii})$$

and Q = flow = 1000 U.S. GPM.

D = pipe inside diameter = 7.981 in.

$$V = \frac{0.4085 (1000)}{7.981^2} = 6.41 \text{ ft/s.}$$

b) It must be determined in which region (1, 2, or 3) this velocity falls. Therefore, the next step is to determine the velocity at the upper limit of the linear region, $V_{\max}$.

$$V_{\max} = K' C^\sigma, \quad [1]$$

and K' = numerical coefficient = 0.85 (from Appendix B, Table I),

C = consistency = 4.5%,

σ = exponent = 1.6 (from Appendix B, Table I).

$$V_{\max} = 0.85 (4.5^{1.6}) = 9.43 \text{ ft/s.}$$

c) Since $V_{\max}$ exceeds V, the friction loss, $\Delta H/L$, falls within the linear region, Region 1. The friction loss is given by the correlation:

$$\Delta H/L = F K V^\alpha C^\beta D^y \quad [2]$$

and F = correction factor = $F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5$,

F_1 = correction factor for pulp temperature. Since the pulp temperature is 95° F,

$$F_1 = 1.0,$$

F_2 = correction factor for pipe roughness. For stainless steel pipe,

$$F_2 = 1.25 \text{ (from Appendix D),}$$

F_3 = correction factor for pulp type. Numerical coefficients for this pulp are contained in Appendix C, Table II, and have already incorporated this factor.

F_4 = correction factor for beating. No additional beating has taken place, therefore

$$F_4 = 1.0 \text{ (from Appendix D),}$$

F_5 = design safety factor. This has been assumed to be unity.

$$F_5 = 1.0.$$

$$F = (1.0) (1.25) (1.0) (1.0) (1.0) = 1.25,$$

K = numerical coefficient = 5.30 (from Appendix C, Table II),

α , β , y = exponents = 0.36, 2.14, and -1.04, respectively (from Appendix C, Table II),

V, C, D have been evaluated previously.

$$\begin{aligned}\Delta H/L &= (1.25) (5.30) (6.41^{0.36}) (4.52^{1.14}) (7.981^{-1.04}) \\ &= (1.25) (5.30) (1.952) (25.0) (0.1153) \\ &= 37.28 \text{ ft head loss/100 ft of pipe.}\end{aligned}$$

This is a rather substantial head loss, but may be acceptable for short piping runs. In a large system, the economics of initial piping costs versus power costs should be weighed, however, before using piping which gives a friction loss of this magnitude.

Example 2.

Determine the friction loss (per 100 ft of pipe) of 2500 U.S. GPM of 3% oven-dried bleached kraft pine, dried and reslurried, in 12 inch schedule 10 stainless steel pipe (pipe inside diameter = 12.39 in). Stock temperature is 1250F.

Solution:

a) V, the bulk velocity, is

$$\begin{aligned}V &= \frac{0.4085 Q}{D^2}, & (ii) \\ &= \frac{0.4085 (2500)}{12.39^2} = 6.65 \text{ ft/s.}\end{aligned}$$

b) The velocity at the upper limit of the linear region, $V_{\max}$, is

$$\begin{aligned}V_{\max} &= K' C \sigma, & [1] \\ \text{and } K' &= 0.59 \text{ (from Appendix B, Table I),} \\ &= 1.45 \text{ (from Appendix B, Table I).}\end{aligned}$$

$$V_{\max} = 0.59 (3.0^{1.45}) = 2.90 \text{ ft/s.}$$

c) Region 1 (the linear region) has been eliminated, since the bulk velocity, V, exceeds $V_{\max}$.

The next step requires calculation of V_w .

$$\begin{aligned}V_w &= 4.00 C^{1.40} & [3] \\ &= 4.00 (3.0^{1.40}) = 18.62 \text{ ft/s.}\end{aligned}$$

d) V exceeds $V_{\max}$, but is less than V_w , indicating that it falls in Region 2. The friction loss in this region is calculated by substituting $V_{\max}$ into the equation for head loss, Equation [2].

$$\Delta H/L = F K (V_{\max})^\alpha C^\beta D^y,$$

$$\text{and } F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5; & (iv)$$

$$F_1 = 1.528 - 0.00556T, & (v)$$

and T = stock temperature = 125° F

$$F_1 = 1.528 - 0.00556 (125) = 0.833,$$

$$F_2 = 1.25 \text{ (from Appendix D),}$$

$$F_3 = F_4 = F_5 = 1.0,$$

$$F = 0.833 (1.25) (1.0) = 1.041,$$

$$K = 8.80 \text{ (from Appendix C, Table II),}$$

$$\alpha, \beta, y = 0.31, 1.81, \text{ and } -1.34, \text{ respectively (from Appendix C, Table II),}$$

$V_{\max}$, C, and D have been defined previously.

$$\begin{aligned}\Delta H/L &= 1.041 (8.80) (2.90^{0.31}) (3.0^{1.81}) (12.39^{-1.34}) \\ &= 1.041 (8.80) (1.391) (7.304) (0.03430) \\ &= 3.19 \text{ ft head loss/100 ft of pipe.}\end{aligned}$$

Example 3.

Determine the friction loss (per 100 ft of pipe) for 2% oven-dried bleached kraft pine, dried and reslurried, through 6 inch schedule 40 stainless steel pipe (inside diameter = 6.065 in). The pulp temperature is 90° F; the flow rate 1100 U.S. GPM.

Solution:

a) The bulk velocity is

$$\begin{aligned}V &= \frac{0.4085 Q}{D^2}, & (ii) \\ &= \frac{0.4085 (1100)}{6.065^2} = 12.22 \text{ ft/s.}\end{aligned}$$

b) It must be determined in which region (1, 2 or 3) this velocity falls.

To obtain an initial indication, determine $V_{\max}$.

$$V_{\max} = K' C \sigma, & [1]$$

and $K' = 0.59$ (from Appendix B, Table I),

$$\sigma = 1.45 \text{ (from Appendix B, Table I).}$$

$$V_{\max} = 0.59 (20^{1.40}) = 1.61 \text{ ft/s.}$$

c) Since V exceeds $V_{\max}$, Region 1 (the linear region) is eliminated. To determine whether V lies in Region 2 or 3, the velocity at the onset of drag reduction, V_w , must be calculated.

$$\begin{aligned}V_w &= 4.00 C^{1.40} & [3] \\ &= 4.00 (2.0^{1.40}) = 10.56 \text{ ft/s.}\end{aligned}$$

d) V exceeds V_w , indicating that it falls in Region 3.

The friction loss is calculated as that of water flowing at the same velocity.

$$\begin{aligned}(\Delta H/L)_w &= 0.579 V^{1.75} D^{-1.25}, & [4] \\ &= 0.579 (12.22^{1.75}) (6.065^{-1.25}) \\ &= 4.85 \text{ ft head loss/100 ft of pipe.}\end{aligned}$$

This will be a conservative estimate, as the actual friction loss curve for pulp suspensions under these conditions will be below the water curve.

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TECH-E-4 Pump Types Used in the Pulp & Paper Industry

Mill Area	Typical Services	Typical Pump Construction	Pump Type	Goulds Model
Woodyard	Log Flume Log/Chip Pile spray Chip Washer	Al/316SS Trim Al/316SS trim Al/316SS Trim	Mixed Flow Vertical Turbine Stock ANSI Double Suction	MF VIT 3175, 3180/85 3196 3410, 3415, 3420
Pump Mill	Shower Supply Dilution Supply Screen Supply Cleaner Supply Decker Supply Hi/Med. Density Storage Transfer Medium Cons. Storage Chip Chute Circulation White Liquor Circulation Condensate Wash Liquor Circulation	Al/31SS Al/316SS Al/316SS 316SS 316SS/317SS 316SS/317SS Various 316SS/317SS CD4MCu CD4MCu Al/316SS 316SS	Stock ANSI Double Suction Medium Consistency Hi Temp/Press Stock	3175, 3180/85 3196 3410, 3415, 3420 3500 3181/86
Bleach Plant	Brown Stock Storage Bleach Tower Storage Bleach Chemical Mixing High Density Storage Chemical Feed Washer Supply Washer Shower Water Dilution Water Medium Consistency O ₂ Reactor ClO ₂ Generator Circulation	316SS 316SS 317SS, 254 SMO, Titanium 316SS/317SS 316SS 316SS 316SS 316SS 316SS 316SS Titanium	Stock ANSI Medium Consistency Axial Flow Non-metallic	3175, 3180/85 3196 3500 AF NM 3196
Stock Prep	Refiner Supply Deflaker Supply Machine Chest Supply	316SS 316SS 316SS	Stock ANSI	3175, 3180/85 3196
Paper Machine (Wet End)	Fan Pumps Couch Pit Saveall Sweetner Shower	Al/316SS Trim, All 316SS Al/316SS Trim, All 316SS Al/316SS Trim, All 316SS Al/316SS Trim, All 316SS A/316SS Trim, All 316SS	Double Suction Stock Low Flow High Pressure Two-Stage	3415, 3420 3175, 3180/85 LF3196 3310H 3316
Paper Machine (Dry End)	Dryer Drainage Condensate Trim Squirt Broke Chest	Al/316SS Trim, Al/316SS Trim Al/316SS Trim Al/316SS Trim	ANSI Low Flow Stock	3196 LF 3196 3175, 3180/85
Coater	Coating Slurries Kaolin Clay (Fillers)	316SS/CD4MCu 316SS/CD4MCu	ANSI Medium Duty Slurry	3196 JC
Kraft Recovery	Weak Black Liquor Evaporator Circulation Concentrated Black Liquor Condensate Injection Black Liquor Transfer Pumps Smelt Spout Cooling Water Collection Weak Wash Scrubber Green Liquor (Storage Transfer) Lime Mud Dregs	316SS 316SS 316SS 316SS 316SS CD4MCu Al/316SS Trim Al/316SS Trim 316SS/CD4MCu/28% Chrome 316SS/CD4MCu/28% Chrome 316SS/CD4MCu/28% Chrome 316SS/CD4MCu/28% Chrome	ANSI Stock Medium Duty Slurry High Temp/Pressure Stock High Pressure Multi Stage	3196 3175, 3180/85 JC 3181/86 3316
Utility (Power House)	Feedwater Condensate Deaerator Booster	CS/Chrome Trim/All Chrome 316SS 316SS	Multi-Stage ANSI High Pressure Vertical Can	3310H, 3600 3196 3700, VIC
Miscellaneous	Mill Water Supply Sump Pumps	Al/316SS Trim Al/316SS Trim	Double Suction Vertical Turbine Self-Priming Vertical Sumps Vertical Sump; Recessed Submersible	3410, 3415, 3420 VIT 3796 3171 VHS HSU
Recycle	Hole/Slot Screen Supply Rejects Floating Cell Medium Consistency Storage Hydro Pulper Dilution Water	316SS/CD4MCu 316SS/CD4MCu 316SS 316SS/317SS 316SS/CD4MCu Al/316SS Trim	Stock Recessed ANSI Medium Consistency	3175, 3180/85 CV 3196,HS 3196 3500



Section TECH-F

Mechanical Data

TECH-F-1 Standard Weights and Dimensions of Mechanical Joint Cast Iron Pipe, Centrifugally Cast

Extracted from USA Standard Cast Iron Pipe Flanges and Flanged Fittings (USAS B16. 1-1967), with the permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, New York 10017.

Nom. Size & (Outside Diam), In.	Thickness, In.	Wall Weight Per Foot*	Average Thickness Class
3 (3.96)	0.32	11.9	22
	0.35	12.9	23
	0.38	13.8	24
4 (4.80)	0.35	16.1	22
	0.38	17.3	23
	0.41	18.4	24
	0.44	19.6	25
6 (6.90)	0.38	25.4	22
	0.41	27.2	23
	0.44	29.0	24
	0.48	31.3	25
8 (9.05)	0.52	33.6	26
	0.41	36.2	22
	0.44	38.6	23
	0.48	41.8	24
	0.52	45.0	25
10 (11.10)	0.56	48.1	26
	0.60	51.2	27
	0.44	48.0	22
	0.48	52.0	23
	0.52	55.9	24
12 (13.20)	0.56	59.9	25
	0.60	63.8	26
	0.65	68.6	27
	0.48	62.3	22
	0.52	67.1	23
	0.56	59.9	25
14 (15.30)	0.60	76.6	25
	0.65	82.5	26
	0.70	88.3	27
	0.76	95.2	28
	0.48	73.6	21
	0.51	77.8	22
	0.55	83.4	23
	0.59	89.0	24
	0.64	95.9	25
	0.69	102.7	26
	0.75	110.9	27
	0.81	118.9	28

Nom. Size & (Outside Diam), In.	Thickness, In.	Wall Weight Per Foot*	Average Thickness Class
16 (17.40)	0.50	87.6	21
	0.54	94.0	22
	0.58	100.3	23
	0.63	108.3	24
	0.68	116.2	25
	0.73	124.0	26
	0.79	133.3	27
	0.85	142.7	28
18 (19.50)	0.54	106.0	21
	0.58	113.2	22
	0.63	122.2	23
	0.68	131.0	24
	0.73	140.0	25
	0.79	150.6	26
	0.85	161.0	27
20 (21.60)	0.92	173.2	28
	0.57	124.2	21
	0.62	134.2	22
	0.67	144.2	23
	0.72	154.1	24
	0.78	165.9	25
	0.84	177.6	26
24 (25.80)	0.91	191.2	27
	0.98	214.8	28
	0.63	164.2	21
	0.68	176.2	22
	0.73	188.2	23
	0.79	202.6	24
	0.85	216.8	25
	0.92	233.2	26
	0.99	249.7	27
	1.07	268.2	28

*Based on 20 Ft. Laying Length of Mech. Joint Pipe including Bell.

TECH-F-2 125 Lb. & 250 Lb. Cast Iron Pipe Flanges and Flanged Fittings

Nominal Pipe Size	Diam. of Flange	Thick-ness of Flange (Min.)	Diam. of Bolt Circle	Num-ber of Bolts	Diam. of Bolts	Diam. of Drilled Bolt Holes	Length of Bolts
1	4 ¹ / ₄	7 ¹ / ₁₆	3 ¹ / ₈	4	1 ¹ / ₂	5 ⁵ / ₈	1 ³ / ₄
1 ¹ / ₄	4 ⁵ / ₈	1 ¹ / ₂	3 ¹ / ₂	4	1 ¹ / ₂	5 ⁵ / ₈	2
1 ¹ / ₂	5	9 ¹ / ₁₆	3 ³ / ₈	4	1 ¹ / ₂	5 ⁵ / ₈	2
2	6	5 ⁵ / ₈	4 ³ / ₄	4	5 ⁵ / ₈	3 ³ / ₄	2 ¹ / ₄
2 ¹ / ₂	7	1 ¹ / ₁₆	5 ¹ / ₂	4	5 ⁵ / ₈	3 ³ / ₄	2 ¹ / ₂
3	7 ¹ / ₂	3 ³ / ₄	6	4	5 ⁵ / ₈	3 ³ / ₄	2 ¹ / ₂
3 ¹ / ₂	8 ¹ / ₂	13 ¹ / ₁₆	7	8	5 ⁵ / ₈	3 ³ / ₄	2 ³ / ₄
4	9	15 ¹ / ₁₆	7 ¹ / ₂	8	5 ⁵ / ₈	3 ³ / ₄	3
5	10	15 ¹ / ₁₆	8 ¹ / ₂	8	3 ³ / ₄	7 ⁷ / ₈	3
6	11	1	9 ¹ / ₂	8	3 ³ / ₄	7 ⁷ / ₈	3 ¹ / ₄
8	13 ¹ / ₂	1 ¹ / ₈	11 ³ / ₄	8	3 ³ / ₄	7 ⁷ / ₈	3 ¹ / ₂
10	16	1 ³ / ₁₆	14 ¹ / ₄	12	7 ⁷ / ₈	1	3 ³ / ₄
12	19	1 ¹ / ₄	17	12	7 ⁷ / ₈	1	3 ³ / ₄
14	21	1 ³ / ₈	18 ³ / ₄	12	1	1 ¹ / ₈	4 ¹ / ₄
16	23 ¹ / ₂	1 ⁷ / ₁₆	21 ¹ / ₂	16	1	1 ¹ / ₈	4 ¹ / ₂
18	25	1 ⁹ / ₁₆	22 ³ / ₄	16	1 ¹ / ₈	1 ¹ / ₄	4 ³ / ₄
20	27 ¹ / ₂	1 ¹¹ / ₁₆	25	20	1 ¹ / ₈	1 ¹ / ₄	5
24	32	1 ⁷ / ₈	29 ¹ / ₂	20	1 ¹ / ₄	1 ³ / ₈	5 ¹ / ₂
30	38 ³ / ₄	2 ¹ / ₈	36	28	1 ¹ / ₄	1 ³ / ₈	6 ¹ / ₄
36	46	2 ³ / ₈	42 ³ / ₄	32	1 ¹ / ₂	1 ³ / ₈	7
42	53	2 ⁵ / ₈	49 ¹ / ₂	36	1 ¹ / ₂	1 ³ / ₈	7 ¹ / ₂
48	59 ¹ / ₂	2 ³ / ₄	56	44	1 ¹ / ₂	1 ³ / ₈	7 ³ / ₄

Chart 2 American Standard Class 125 Cast Iron Pipe Flanges (ASA B16.1)

Nominal Pipe Size	Diam. of Flange	Thick-ness of Flange ³ (Min.)	Diam. of Bolt Circle	Diam. of Bolt Holes ¹	Num-ber of Bolts ¹	Size of Bolt	Length of Bolts ²
1	4 ⁷ / ₈	1 ¹ / ₁₆	3 ¹ / ₂	3 ³ / ₄	4	5 ⁵ / ₈	2 ¹ / ₂
1 ¹ / ₄	5 ¹ / ₄	3 ³ / ₄	3 ³ / ₈	3 ³ / ₄	4	5 ⁵ / ₈	2 ¹ / ₂
1 ¹ / ₂	6 ¹ / ₈	13 ¹ / ₁₆	4 ¹ / ₂	7 ⁷ / ₈	4	3 ³ / ₄	2 ³ / ₄
2	6 ¹ / ₂	7 ⁷ / ₈	5	3 ³ / ₄	8	5 ⁵ / ₈	2 ³ / ₄
2 ¹ / ₂	7 ¹ / ₂	1	5 ⁷ / ₈	7 ⁷ / ₈	8	3 ³ / ₄	3 ¹ / ₄
3	8 ¹ / ₄	1 ¹ / ₈	6 ⁵ / ₈	7 ⁷ / ₈	8	3 ³ / ₄	3 ¹ / ₂
3 ¹ / ₂	9	1 ³ / ₁₆	7 ¹ / ₄	7 ⁷ / ₈	8	3 ³ / ₄	3 ¹ / ₂
4	10	1 ¹ / ₄	7 ⁷ / ₈	7 ⁷ / ₈	8	3 ³ / ₄	3 ³ / ₄
5	11	1 ³ / ₈	9 ¹ / ₄	7 ⁷ / ₈	8	3 ³ / ₄	4
6	12 ¹ / ₂	1 ⁷ / ₁₆	10 ⁵ / ₈	7 ⁷ / ₈	12	3 ³ / ₄	4
8	15	1 ⁵ / ₈	13	1	12	7 ⁷ / ₈	4 ¹ / ₂
10	17 ¹ / ₂	1 ⁷ / ₈	15 ¹ / ₄	1 ¹ / ₈	16	1	5 ¹ / ₄
12	20 ¹ / ₂	2	17 ³ / ₄	1 ¹ / ₄	16	1 ¹ / ₈	5 ¹ / ₂
14	23	2 ¹ / ₈	20 ¹ / ₄	1 ¹ / ₄	20	1 ¹ / ₈	6
16	25 ¹ / ₂	2 ¹ / ₄	22 ¹ / ₂	1 ³ / ₈	20	1 ¹ / ₄	6 ¹ / ₄
18	28	2 ³ / ₈	24 ³ / ₄	1 ³ / ₈	24	1 ¹ / ₄	6 ¹ / ₂
20	30 ¹ / ₂	2 ¹ / ₂	27	1 ³ / ₈	24	1 ¹ / ₄	6 ³ / ₄
24	36	2 ³ / ₄	32	1 ³ / ₈	24	1 ¹ / ₂	7 ³ / ₄
*30	43	3	39 ¹ / ₄	2	28	1 ¹ / ₂	8 ¹ / ₂
*36	50	3 ³ / ₈	46	2 ¹ / ₄	32	2	9 ¹ / ₂
*42	57	3 ¹ / ₁₆	52 ³ / ₂	2 ¹ / ₄	36	2	10 ¹ / ₄
*48	65	4	60 ³ / ₄	2 ¹ / ₄	40	2	10 ³ / ₄

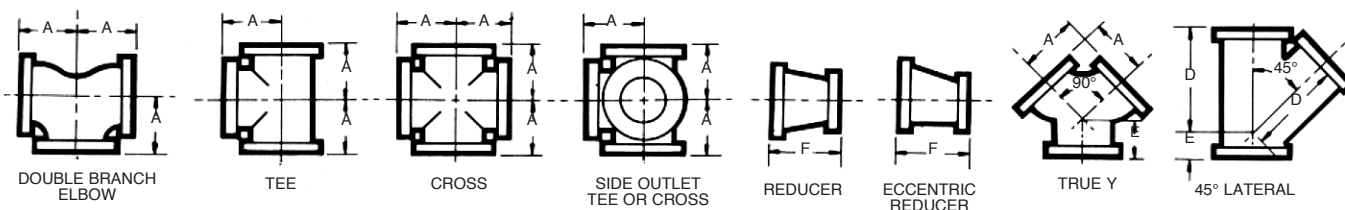
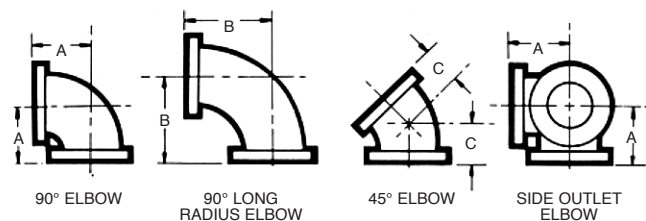
Chart 3 American Standard Class 250 Cast Iron Pipe Flanges (ASA B16b)

Nominal Pipe Size	Center to Face					Face to Face F	Body Wall Thick-ness ¹
	A	B	C	D	E		
1	3 ¹ / ₂	5	1 ³ / ₄	5 ³ / ₄	1 ³ / ₄		5 ¹ / ₁₆
1 ¹ / ₄	3 ³ / ₄	5 ¹ / ₂	2	6 ¹ / ₄	1 ³ / ₄		5 ¹ / ₁₆
1 ¹ / ₂	4	6	2 ¹ / ₄	7	2		5 ¹ / ₁₆
2	4 ¹ / ₂	6 ¹ / ₂	2 ¹ / ₂	8	2 ¹ / ₂	5	5 ¹ / ₁₆
2 ¹ / ₂	5	7	3	9 ¹ / ₂	2 ¹ / ₂	5 ¹ / ₂	5 ¹ / ₁₆
3	5 ¹ / ₂	7 ³ / ₄	3	10	3	6	3 ³ / ₈
3 ¹ / ₂	6	8 ¹ / ₂	3 ¹ / ₂	11 ¹ / ₂	3	6 ¹ / ₂	7 ¹ / ₁₆
4	6 ¹ / ₂	9	4	12	3	7	1 ¹ / ₂
5	7 ¹ / ₂	10 ¹ / ₄	4 ¹ / ₂	13 ¹ / ₂	3 ¹ / ₂	8	1 ¹ / ₂
6	8	11 ¹ / ₂	5	14 ¹ / ₂	3 ¹ / ₂	9	9 ¹ / ₁₆
8	9	14	5 ¹ / ₂	17 ¹ / ₂	4 ¹ / ₂	11	5 ⁵ / ₈
10	11	16 ¹ / ₂	6 ¹ / ₂	20 ¹ / ₂	5	14	3 ³ / ₄
12	12	19	7 ¹ / ₂	24 ¹ / ₂	5 ¹ / ₂	14	13 ¹ / ₁₆
14	14	21 ¹ / ₂	7 ¹ / ₂	27	6	16	7 ⁷ / ₈
16	15	24	8	30	6 ¹ / ₂	18	1
18	16 ¹ / ₂	26 ¹ / ₂	8 ¹ / ₂	32	7	19	1 ¹ / ₁₆
20	18	29	9 ¹ / ₂	35	8	20	1 ¹ / ₈
24	22	34	11	40 ¹ / ₂	9	24	1 ¹ / ₄
30	25	41 ¹ / ₂	15	49	10	30	1 ⁷ / ₁₆
36	28*	49	18			36	1 ⁵ / ₈
42	31*	56 ¹ / ₂	21			42	1 ¹³ / ₁₆
48	34*	64	24			48	2

Chart 4 American Standard Class 125 Cast Iron Flanged Fittings (ASA B16.1)

Nominal Pipe Size	Inside Diam. of Fitting (Min.)	Wall Thick-ness of Body*	Diam. of Raised Face	Center to Face					Face to Face F
				A	B	C	D	E	
2	2	7 ¹ / ₁₆	4 ³ / ₁₆	5	6 ¹ / ₂	3	9	2 ¹ / ₂	5
2 ¹ / ₂	2 ¹ / ₂	1 ¹ / ₂	4 ¹⁵ / ₁₆	5 ¹ / ₂	7	3 ¹ / ₂	10 ¹ / ₂	2 ¹ / ₂	5 ¹ / ₂
3	3	9 ¹ / ₁₆	5 ¹¹ / ₁₆	6	7 ³ / ₄	3 ¹ / ₂	11	3	6
3 ¹ / ₂	3 ¹ / ₂	9 ¹ / ₁₆	6 ⁵ / ₁₆	6 ¹ / ₂	8 ¹ / ₂	4	12 ¹ / ₂	3	6 ¹ / ₂
4	4	5 ⁵ / ₈	6 ¹⁵ / ₁₆	7	9	4 ¹ / ₂	13 ¹ / ₂	3	7
5	5	1 ¹ / ₁₆	8 ⁹ / ₁₆	8	10 ¹ / ₄	5	15	3 ¹ / ₂	8
6	6	3 ³ / ₄	9 ¹ / ₁₆	8 ¹ / ₂	11 ¹ / ₂	5 ¹ / ₂	17 ¹ / ₂	4	9
8	8	13 ¹ / ₁₆	11 ¹⁵ / ₁₆	10	14	6	20 ¹ / ₂	5	11
10	10	15 ¹ / ₁₆	14 ⁷ / ₁₆	11 ¹ / ₂	16 ¹ / ₂	7	24	5 ¹ / ₂	12
12	12	1	16 ⁷ / ₁₆	13	19	8	27 ¹ / ₂	6	14
14	13 ³ / ₄	1 ¹ / ₈	18 ¹⁵ / ₁₆	15	21 ¹ / ₂	8 ¹ / ₂	31	6 ¹ / ₂	16
16	15 ¹ / ₄	1 ¹ / ₄	21 ¹ / ₁₆	16 ¹ / ₂	24	9 ¹ / ₂	34 ¹ / ₂	7 ¹ / ₂	18
18	17	1 ³ / ₈	23 ⁵ / ₁₆	18	26 ¹ / ₂	10	37 ¹ / ₂	8	19
20	19	1 ¹ / ₂	25 ⁹ / ₁₆	19 ¹ / ₂	29	10 ¹ / ₂	40 ¹ / ₂	8 ¹ / ₂	20
24	23	1 ⁵ / ₈	30 ¹ / ₄	22 ¹ / ₂	34	12	47 ¹ / ₂	10	24

Chart 5 American Standard Class 250 Cast Iron Flanged Fittings (ASA B16b)



TECH-F-3 Steel Pipe, Dimensions and Weights

Size: Nom. & (Outside Diam.), In.*	Wall Thickness, In.	Weight per Foot, Plain Ends, Lb.	Schedule No.
1/8 (0.405)	0.068 0.095	0.24 0.31	40 S 80 XS
1/4 (0.540)	0.088 0.119	0.42 0.54	40 S 80 XS
3/8 (0.675)	0.091 0.126	0.57 0.74	40 S 80 XS
1/2 (0.840)	0.109 0.147 0.188 0.294	0.85 1.09 1.31 1.71	40 S 80 XS 160 XXS
3/4 (1.050)	0.113 0.154 0.219 0.308	1.13 1.47 1.94 2.44	40 S 80 XS 160 XXS
1 (1.315)	0.133 0.179 0.250 0.308	1.68 2.17 2.84 2.44	40 S 80 XS 160 XXS
1 1/4 (1.660)	0.140 0.191 0.250 0.382	2.27 3.00 3.76 5.21	40 S 80 XS 160 XXS
1 1/2 (1.900)	0.145 0.200 0.281 0.400	2.72 3.63 4.86 6.41	40 S 80 XS 160 XXS
2 (2.375)	0.154 0.218 0.344 0.436	3.65 5.02 7.46 9.03	40 S 80 XS 160 XXS
2 1/2 (2.875)	0.203 0.276 0.375 0.552 0.216	5.79 7.66 10.01 13.70 7.58	40 S 80 XS 160 XXS 40 S
3 (3.500)	0.300 0.438 0.600	10.25 14.31 18.58	80 XS 160 XXS
3 1/2 (4.000)	0.226 0.318	9.11 12.51	40 S 80 XS
4 (4.500)	0.237 0.337 0.438 0.531 0.674	10.79 14.98 18.98 22.52 27.54	40 S 80 XS 120 160 XXS
5 (5.563)	0.258 0.375 0.500 0.625 0.750	14.62 20.78 27.04 32.96 38.55	40 S 80 XS 120 160 XXS
6 (6.625)	0.280 0.432 0.562 0.719 0.864	18.97 28.57 36.42 45.34 53.16	40 S 80 XS 120 160 XXS
8 (8.625)	0.250 0.277 0.322 0.406 0.500 0.594 0.719 0.812 0.875 0.906	22.36 24.70 28.55 35.66 43.39 50.93 45.34 67.79 72.42 74.71	20 30 40 S 60 80 XS 100 160 140 XXS 160
10 (10.750)	0.250 0.307 0.365 0.500 0.594 0.719 0.844 1.000 1.125	28.04 34.24 40.48 54.74 64.40 77.00 89.27 104.13 115.65	20 30 40 S 60 XS 80 100 120 140 160

Size: Nom. & (Outside Diam.), In.*	Wall Thickness, In.	Weight per Foot, Plain Ends, Lb.	Schedule No.
12 (12.750)	0.250 0.330 0.375 0.406 0.500 0.562 0.688 0.844 1.000 1.125	33.38 43.77 49.56 53.56 65.42 73.22 88.57 107.29 125.49 139.68	20 30 S 40 XS 60 80 100 120 140
14 (14.000)	1.312 0.250 0.312 0.375 0.438 0.500 0.594 0.750 0.938 1.094 1.250 1.406	160.33 36.71 45.68 54.57 63.37 72.09 85.01 106.13 130.79 150.76 170.22 189.15	160 10 20 30 40 XS 60 80 100 120 140 160
16 (16.000)	0.250 0.312 0.375 0.500 0.656 0.844 1.031 1.219 1.438 1.594	42.05 52.36 62.58 82.77 107.54 136.58 164.86 192.40 223.57 245.22	10 20 30 40 XS 60 80 100 120 140 160
18 (18.000)	0.250 0.312 0.375 0.438 0.500 0.562 0.750 0.938 1.156 1.375 1.562 1.781	47.39 59.03 70.59 82.06 93.45 104.76 138.17 170.84 208.00 244.14 274.30 308.55	10 20 S 30 XS 40 60 80 100 120 140 160
20 (20.000)	0.250 0.375 0.500 0.594 0.812 1.031 1.281 1.500 1.750 1.969	47.39 78.60 93.45 123.06 166.50 208.92 256.15 296.37 341.10 379.14	10 20 S XS 40 60 80 100 120 140 160
22 (22.000)	0.250 0.375 0.500 0.875 1.125 1.375 1.625 1.875 2.125	58.07 86.61 114.81 197.42 250.82 302.88 353.61 403.01 451.07	10 20 S 30 XS 60 80 100 120 140 160
24 (24.000)	0.250 0.375 0.500 0.562 0.688 0.969 1.219 1.531 1.812 2.062 2.344	63.41 94.62 125.49 140.80 171.17 238.29 296.53 367.45 429.50 483.24 542.09	10 20 S XS 30 40 60 80 100 120 140 160

TECH-F-4 150 Lb. & 300 Lb. Steel Pipe Flanges and Fittings

Extracted from USA Standard Cast Iron Pipe Flanges and Flanged Fittings (USAS, B16. 5-1968), with the permission of the publisher, The American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York NY 10017.

Nominal Pipe Size	Diam. of Flange O	Thick-ness of Flange (Min.)*	Diam. of Bolt Circle	Diam. of Bolt Holes	Number of Bolts	Diam. of Bolts	Length of (with 1/16" Raised Face
1/2	3 1/2	7/16	2 3/8	5/8	4	1/2	1 3/4
3/4	3 7/8	1/2	2 1/4	5/8	4	1/2	2
1	4 1/4	9/16	3 1/8	5/8	4	1/2	2
1 1/4	4 5/8	5/8	3 1/2	5/8	4	1/2	2 1/4
1 1/2	5	1 1/16	3 7/8	5/8	4	1/2	2 1/4
2	6	3/4	4 3/4	3/4	4	5/8	2 3/4
2 1/2	7	7/8	5 1/2	3/4	4	5/8	3
3	7 1/2	15/16	6	3/4	4	5/8	3
3 1/2	8 1/2	15/16	7	3/4	8	5/8	3
4	9	15/16	7 1/2	3/4	8	5/8	3
5	10	15/16	8 1/2	7/8	8	3/4	3 1/4
6	11	1	9 1/2	7/8	8	3/4	3 1/4
8	13 1/2	1 1/8	11 3/4	7/8	8	3/4	3 1/2
10	16	1 3/16	14 1/4	1	12	7/8	3 3/4
12	19	1 1/4	17	1	12	7/8	4
14	21	1 3/8	18 3/4	1 1/8	12	1	4 1/4
16	23 1/2	1 7/16	21 1/4	1 1/8	16	1	4 1/2
18	25	1 9/16	22 3/4	1 1/4	16	1 1/8	4 3/4
20	27 1/2	1 11/16	25	1 1/4	20	1 1/8	5 1/4
24	32	1 7/8	29 1/2	3/8	20	1 1/4	5 3/4

Chart 6 150 Lb. Steel Pipe Flanges

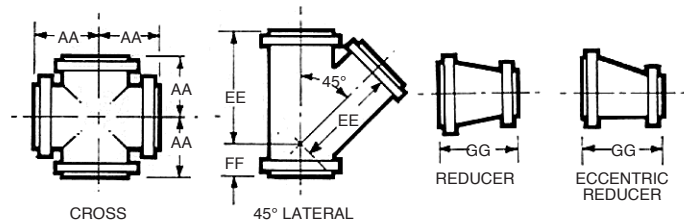
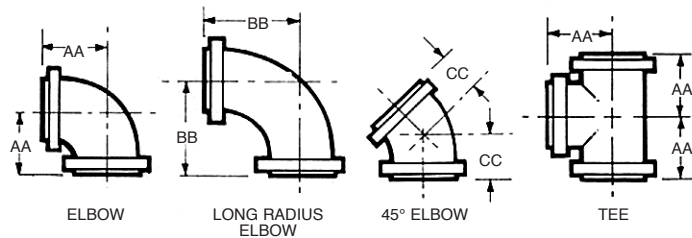
Nominal Pipe Size	Flange Diam.	Flange Thick-ness (Min.)*	Bolt Circle Diam.	Diam. of Bolt Holes	No. of Bolts	Size of Bolts
1	4 7/8	1 1/16	3 1/2	3/4	4	5/8
1 1/4	5 1/4	3/4	3 7/8	3/4	4	5/8
1 1/2	6 1/8	13/16	4 1/2	7/8	4	3/4
2	6 1/2	7/8	5	-	8	5/8
2 1/2	7 1/2	1	5 5/8	7/8	8	3/4
3	8 1/4	1 1/8	6 3/8	7/8	8	3/4
3 1/2	9	1 3/16	7 1/4	7/8	8	3/4
4	10	1 1/4	7 1/8	7/8	8	3/4
5	11	1 3/8	9 1/4	7/8	8	3/4
6	12 1/2	1 7/16	10 5/8	7/8	12	3/4
8	15	1 5/8	13	1	12	7/8
10	17 1/2	1 7/8	15 1/4	1 1/8	16	1
12	20 1/2	2	17 3/4	1 1/4	16	1 1/8
14	23	2 1/8	20 1/4	1 1/4	20	1 1/8
16	25 1/2	2 1/4	22 1/2	1 3/8	20	1 1/4
18	28	2 3/8	24 3/4	1 3/8	24	1 1/4
20	30 1/2	2 1/2	27	1 3/8	24	1 1/4
24	36	2 3/4	32	1 5/8	24	1 1/2

Chart 7 300 Lb. Steel Pipe Flanges

* A raised face of 1/16 inch is included in (a) minimum thickness of flanges, and (b) "center to contact surface" dimension of fitting. Where facings other than 1/16 inch raised face are used, the "center to contact surface" dimensions shall remain unchanged.

Nominal Pipe Size	AA	BB	CC	EE	FF	GG
1	3 1/2	5	1 3/4	5 3/4	1 3/4	4 1/2
1 1/4	3 3/4	5 1/2	2	6 1/4	1 3/4	4 1/2
1 1/2	4	6	2 1/4	7	2	4 1/2
2	4 1/2	6 1/2	2 1/2	8	2 1/2	5
2 1/2	5	7	3	9 1/2	2 1/2	5 1/2
3	5 1/2	7 3/4	3	10	3	6
3 1/2	6	8 1/2	3 1/2	11 1/2	3	6 1/2
4	6 1/2	9	4	12	3	7
5	7 1/2	10 1/4	4 1/2	13 1/2	3 1/2	8
6	8	11 1/2	5	14 1/2	3 1/2	9
8	9	14	5 1/2	17 1/2	4 1/2	11
10	11	16 1/2	6 1/2	20 1/2	5	12
12	12	19	7 1/2	24 1/2	5 1/2	14
14	14	21 1/2	7 1/2	27	6	16
16	15	24	8	30	6 1/2	18
18	16 1/2	26 1/2	8 1/2	32	7	19
20	18	29	9 1/2	35	8	20
24	22	34	11	40 1/2	9	24

Chart 8 150 Lb. Steel Flanged Fittings



Nominal Pipe Size	AA	BB	CC	EE	FF	GG
1	4	5	2 1/4	6 1/2	2	4 1/2
1 1/4	4 1/4	5 1/2	2 1/2	7 1/4	2 1/4	4 1/2
1 1/2	4 1/2	6	2 3/4	8 1/2	2 1/2	4 1/2
2	5	6 1/2	3	9	2 1/2	5
2 1/2	5 1/2	7	3 1/2	10 1/2	2 1/2	5 1/2
3	6	7 3/4	3 1/2	11	3	6
3 1/2	6 1/2	8 1/2	4	12 1/2	3	6 1/2
4	7	9	4 1/2	13 1/2	3	7
5	8	10 1/4	5	15	3 1/2	8
6	8 1/2	11 1/2	5 1/2	17 1/2	4	9
8	10	14	6	20 1/2	5	11
10	11 1/2	16 1/2	7	24	5 1/2	12
12	13	19	8	27 1/2	6	14
14	15	21 1/2	8 1/2	31	6 1/2	16
16	16 1/2	24	9 1/2	34 3/4	7 1/2	18
18	18	26 1/2	10	37 1/2	8	19
20	19 1/2	29	10 1/2	40 1/2	8 1/2	20
24	22 1/2	34	12	47 1/2	10	24

Chart 9 300 Lb. Steel Flanged Fittings

TECH-F-5 150 Lb. ANSI / Metric Flange Comparison

Flange Nom.	Outside Diameter			Bolt Circle			Thickness (Min.)			Bolt Hole			Bolts Quantity			Bolt Size			Raised Face Diameter		
	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K	ANSI 150 lb.	ISO 10 Bar	JIS 10 K
1.00	4.25	4.53	4.92	3.12	3.35	3.54	0.56	0.63	0.55	0.62	0.55	0.75	4	-	-	0.5	-	-	2.00	2.68	2.64
25	108	115	125	79	85	90	14	16	14	16	14	19	-	4	4	-	M12	M16	51	68	67
1.50	5.00	5.91	5.51	3.88	4.33	4.13	0.69	0.71	0.63	0.62	0.71	0.75	4	-	-	0.5	-	-	2.88	3.46	3.19
40	127	150	140	98	110	105	17	18	16	16	18	19	-	4	4	-	M16	M16	73	88	81
2.00	6.00	6.50	6.10	4.75	4.92	4.72	0.75	0.79	0.63	0.75	0.71	0.75	4	-	-	0.62	-	-	3.62	4.02	3.78
50	52	165	155	121	125	120	19	20	16	19	18	19	-	4	4	-	M16	M16	92	102	96
2.50	7.00	7.28	6.89	5.50	5.71	5.51	0.88	0.79	0.71	0.75	0.71	0.75	4	-	-	0.62	-	-	4.12	4.80	4.57
65	178	185	175	140	145	140	22	20	18	19	18	19	-	8	4	-	M16	M16	105	122	116
3.00	7.50	7.87	7.28	6.00	6.30	5.91	0.94	0.79	0.71	0.75	0.71	0.75	4	-	-	0.62	-	-	5.00	5.24	4.96
80	191	200	185	152	160	150	24	20	18	19	18	19	-	8	8	-	M16	M16	127	133	126
3.50	8.50	0.00	7.68	7.00	0.00	6.30	0.94	0.00	0.71	0.75	0.00	0.75	8	-	-	0.62	-	-	5.50	0.00	5.35
90	216	0	195	178	0	160	24	0	18	19	0	19	-	-	8	-	-	M16	140	0	136
4.00	9.00	8.66	8.27	7.50	7.09	6.89	0.94	0.87	0.71	0.75	0.71	0.75	8	-	-	0.62	-	-	6.19	6.22	5.94
100	229	220	210	191	180	175	24	22	18	19	18	19	-	8	8	-	M16	M16	157	158	151
6.00	11.00	11.22	11.02	9.50	9.45	9.45	1.00	0.94	0.87	0.88	0.87	0.91	8	-	-	0.75	-	-	8.50	8.35	8.35
150	279	285	280	241	240	240	25	24	22	22	22	23	-	8	8	-	M20	M20	216	212	212
8.00	13.50	13.39	12.99	11.75	11.61	11.42	1.12	0.94	0.87	0.88	0.87	0.91	8	-	-	0.75	-	-	10.62	10.55	10.31
200	343	340	330	298	295	290	28	24	22	22	22	23	-	8	12	-	M20	M20	270	268	262
10.00	16.00	15.55	15.75	14.25	13.78	13.98	1.19	1.02	0.94	1.00	0.87	0.98	12	-	-	0.88	-	-	12.75	12.60	12.76
250	406	395	400	362	350	355	30	26	24	25	22	25	-	12	12	-	M20	M22	324	320	324
12.00	19.00	17.52	17.52	17.00	15.75	15.75	1.25	1.10	0.94	1.00	0.87	0.98	12	-	-	0.88	-	-	15.00	14.57	14.49
300	483	445	445	432	400	400	32	28	24	25	22	25	-	12	16	-	M20	M22	381	370	368
14.00	21.00	19.88	19.29	18.75	18.11	17.52	1.38	1.18	1.02	1.12	0.87	0.98	12	-	-	1.00	-	-	16.25	16.93	16.26
350	533	505	490	476	460	445	35	30	26	28	22	25	-	16	16	-	M20	M22	413	430	413
16.00	23.50	22.24	22.05	21.25	20.28	20.08	1.44	1.26	1.10	1.12	1.02	1.06	16	-	-	1.00	-	-	18.50	18.98	18.70
400	597	565	560	540	515	510	37	32	28	28	26	27	-	16	16	-	M24	M24	470	482	475
18.00	25.00	24.21	24.41	22.75	22.24	22.24	1.56	1.38	1.18	1.25	1.02	1.06	16	-	-	1.12	-	-	21.00	20.94	20.87
450	635	615	620	578	565	565	40	35	30	32	26	27	-	20	20	-	M24	M24	533	532	530
20.00	27.50	26.38	26.57	25.00	24.41	24.41	1.69	1.50	1.18	1.25	1.02	1.06	20	-	-	1.12	-	-	23.00	23.03	23.03
500	699	670	675	635	620	620	43	38	30	32	26	27	-	20	20	-	M24	M24	584	585	585
24.00	32.00	30.71	31.30	29.50	28.54	28.74	1.88	1.65	1.26	1.38	1.16	1.30	20	-	-	1.25	-	-	27.25	26.97	27.17
600	813	780	795	749	725	730	48	42	32	35	29.5	33	-	20	24	-	M27	M30	692	685.0	690
30.00	38.75	0.00	38.19	36.00	0.00	35.43	2.12	0.00	1.42	1.38	0.00	1.30	28	-	-	1.25	-	-	33.75	0.00	33.66
750	984	0	970	914	0	900	54	0	36	35	0	33	-	0	24	-	-	M30	857	0	855
36.00	46.00	43.90	44.09	42.75	41.34	41.34	2.38	1.34	1.50	1.62	1.28	1.30	32	-	-	1.50	-	-	40.25	39.57	39.57
900	1168	1115	1120	1086	1050	1050	60	34	38	41	32.5	33	-	28	28	-	M30	M30	1022	1005.0	1005
42.00	53.00	48.43	48.62	49.50	45.67	45.67	2.62	1.34	1.57	1.62	1.40	1.54	36	-	-	1.50	-	-	47.00	43.70	43.70
1000	1230	1230	1235	1257	1160	1160	67	34	40	41	35.5	39	-	28	28	-	M33	M36	1194	1110.0	1110
48.00	59.50	57.28	57.68	56.00	54.33	54.33	2.75	1.50	1.73	1.62	1.54	1.54	44	-	-	1.50	-	-	53.50	52.36	52.17
1200	1230	1455	1465	1422	1380	1380	70	38	44	41	39	39	-	32	32	-	M36	M36	1359	1330	1325

TECH-F-6 300 Lb. ANSI / Metric Flange Comparison

Flange Nom. I.D.	Outside Diameter			Bolt Circle			Thickness (Min.)			Bolt Hole			Bolts Quantity			Bolt Size			Raised Face Diameter		
	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K	ANSI 300 lb.	ISO 16 Bar	JIS 16 K
1.00	4.88	4.53	4.92	3.50	3.35	3.54	0.69	0.63	0.55	0.75	0.55	0.75	4	-	-	0.62	-	-	2.00	2.68	2.64
25	124	115	125	90	85	90	17	16	14	19	14	19	-	4	4	-	M12	M16	51	68	67
1.50	6.12	5.91	5.51	4.50	4.33	4.13	0.81	0.71	0.63	0.88	0.71	0.75	4	-	-	0.75	-	-	2.88	3.46	3.19
40	156	150	140	114	110	105	21	18	16	22	18	19	-	4	4	-	M16	M16	73	88	81
2.00	6.50	6.50	6.10	5.00	4.92	4.72	0.88	0.79	0.63	0.75	0.71	0.75	8	-	-	0.62	-	-	3.62	4.02	3.78
50	165	165	155	127.0	125	120	22	20	16	19	18	19	-	4	8	-	M16	M16	92	102	96
2.50	7.50	7.28	6.89	5.88	5.71	5.51	1.00	0.79	0.71	0.88	0.71	0.75	8	-	-	0.75	-	-	4.12	4.80	4.57
65	191	185	175	149	145	140	25	20	18	22	18	19	-	8	8	-	M16	M16	105	122	116
3.00	8.25	7.87	7.87	6.62	6.30	6.30	1.12	0.79	0.79	0.88	0.71	0.91	8	-	-	0.75	-	-	5.00	5.24	5.20
80	210	200	200	169	160	160	29	20	20	22	18	23	-	8	8	-	M16	M20	127	133	132
3.50	9.00	0.00	8.27	7.25	0.00	6.69	1.19	0.00	0.79	0.88	0.00	0.91	8	-	-	0.75	-	-	5.50	0.00	5.71
90	229	-	210	184	-	170	30	-	20	22	-	23	-	-	8	-	-	M20	140	0	145
4.00	10.00	8.66	8.86	7.88	7.09	7.28	1.25	0.87	0.87	0.88	0.71	0.91	8	-	-	0.75	-	-	6.19	6.22	6.30
100	254	220	225	200	180	185	32	22	22	22	18	23	-	8	8	-	M16	M20	157	158	160
6.00	12.50	11.22	12.01	10.62	9.54	10.24	1.44	0.94	0.94	0.88	0.87	0.98	12	-	-	0.75	-	-	8.50	8.35	9.06
150	381	285	305	270	240	260	37	24	24	22	22	25	-	8	12	-	M20	M22	216	212	230
8.00	15.00	13.39	13.78	13.00	11.61	12.01	1.62	1.02	1.02	1.00	0.87	0.98	12	-	-	0.88	-	-	10.62	10.55	10.83
200	381	340	350	330	295	305	41	26	26	25	22	25	-	12	12	-	M20	M22	270	268	275
10.00	17.50	15.94	16.93	15.25	13.98	14.96	1.88	1.10	1.10	1.12	1.02	1.06	16	-	-	1.00	-	-	12.75	12.60	13.58
250	445	405	430	387	355	380	48	28	28	28	26	27	-	12	12	-	M24	M24	324	320	345
12.00	20.50	18.11	18.90	17.75	16.14	16.93	2.00	1.26	1.18	1.25	1.02	1.06	16	-	-	1.12	-	-	15.00	14.57	15.55
300	521	460	480	451	410	430	51	32	30	32	26	27	-	12	16	-	M24	M24	381	370	395
14.00	23.00	20.47	21.26	20.25	18.50	18.90	2.12	1.38	1.34	1.25	1.02	1.30	20	-	-	1.12	-	-	16.25	16.93	17.32
350	584	520	540	514	470	480	54	35	34	32	26	33	-	16	16	-	M24	M30	413	430	440
16.00	25.50	22.83	23.82	22.50	20.67	21.26	2.25	1.50	1.50	1.38	1.16	1.30	20	-	-	1.25	-	-	18.50	18.98	19.49
400	648	580	605	572	525	540	57	38	38	35	29.5	33	-	16	16	-	M27	M30	470	482	495
18.00	28.00	25.20	26.57	24.75	23.03	23.82	2.83	1.65	1.57	1.38	1.16	1.30	24	-	-	1.25	-	-	21.00	20.94	22.05
450	711	640	675	629	585	605	60	42	40	35	29.5	33	-	20	20	-	M27	M30	533	532	560
20.00	30.50	28.15	28.74	27.00	25.59	25.98	2.50	1.81	1.65	1.38	1.28	1.30	24	-	-	1.25	-	-	23.00	23.03	24.21
500	775	715	730	686	650	660	64	46	42	35	32.5	33	-	20	20	-	M30	M30	584	585	615
24.00	36.00	33.07	33.27	32.00	30.31	30.31	2.75	2.05	1.81	1.62	1.40	1.54	24	-	-	1.50	-	-	27.25	26.97	28.35
600	914	840	845	813	770	770	70	52	46	41	35.5	39	-	20	24	-	M33	M36	692	685.0	720
30.00	43.00	0.00	40.16	39.25	0.00	36.81	3.00	0.00	2.05	1.88	0.00	1.65	28	-	-	1.75	-	-	33.75	0.00	34.65
750	1092	0	1020	997	0	935	76	0	52	48	0	42	-	0	24	-	0	M39	857	0	880
36.00	50.00	44.29	46.65	46.00	41.34	42.91	3.38	2.99	2.28	2.12	1.54	1.89	32	-	-	2.00	-	-	40.25	39.57	40.55
900	1270	1125	1185	1168	1050	1090	86	76	58	54	39	48	-	28	28	-	M36	M45	1022	1005.0	1030
42.00	57.00	49.41	51.97	52.75	46.06	47.64	3.69	3.31	2.44	2.12	1.65	2.20	36	-	-	2.00	-	-	47.00	43.70	44.88
1000	1448	1255	1320	1340	1170	1210	94	84	62	54	42	56	-	28	28	-	M39	M52	1194	1110.0	1140
48.00	65.00	58.46	60.24	60.75	54.72	55.91	4.00	3.86	2.76	2.12	1.89	2.20	40	-	-	2.00	-	-	58.44	52.36	53.15
1200	1651	1485	1530	1543	1390	1420	102	98	70	54	48	56	-	32	32	-	M45	M52	1484	1330	1350

TECH-F-7 Weights and Dimensions of Steel & Wrought Iron Pipe Recommended for Use as Permanent Well Casings

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Size In.	Steel Pipe, Black or Galvanized				
	Diameter - In.		Thickness In.	Weight Per Foot - Lb ¹	
	External	Internal		Plain Ends (Calculated)	With Threads and Couplings (Nominal) ²
6	6.625	6.065	0.280	18.97	19.18
8	8.625	8.249	0.188	16.90	17.80
8	8.625	8.071	0.277	24.70	25.55
8	8.625	7.981	0.322	28.55	29.35
10	10.750	10.192	0.279	31.20	32.75
10	10.750	10.136	0.307	34.24	35.75
10	10.750	10.020	0.365#	40.48	41.85
12	12.750	12.090	0.330	43.77	45.45
12	12.750	12.000	0.375#	49.56	51.15
14	14.000	13.500	0.250	36.71	
14	14.000	13.250	0.375#	54.57	57.00
16	16.000	15.376	0.312	52.36	
16	16.000	15.250	0.375#	62.58	65.30
18	18.000	17.376	0.312	59.03	
18	18.000	17.250	0.375#	70.59	73.00
20	20.000	19.376	0.312	65.71	
20	20.000	19.250	0.375#	78.60	81.00
22	22.000	21.376	0.312	72.38	
22	22.000	21.250	0.375	86.61	
22	22.000	21.000	0.500	114.81	
24	24.000	23.376	0.312	79.06	
24	24.000	23.250	0.375	94.62	
24	24.000	23.000	0.500#	125.49	
26	26.000	25.376	0.312	85.73	
26	26.000	25.000	0.500#	136.17	
28	28.000	27.376	0.312	92.41	
28	28.000	27.000	0.500#	146.85	
30	30.000	29.376	0.312	99.08	
30	30.000	29.000	0.500#	157.53	
32	32.000	31.376	0.312	105.76	
32	32.000	31.000	0.500#	168.21	
34	34.000	33.376	0.312	112.43	
34	34.000	33.000	0.500#	178.89	
36	36.000	35.376	0.312	119.11	
36	36.000	35.000	0.500#	189.57	

#Thickness indicated is believed to be best practice. If soil and water conditions are unusually favorable, lighter pipe may be used if permitted in the purchaser's specifications.

¹Manufacturing weight tolerance is 10 per cent over and 3.5 per cent under nominal weight for pipe 6-20 in. in size and +/- per cent of nominal weight for larger sizes.

²Nominal weights of pipe with threads and couplings (based on lengths of 20 ft. including coupling) are shown for purposes of specification. Thread data are contained in the various standards covering sizes which can be purchased with threads.

Size In.	Wrought-Iron Pipe, Black or Galvanized				
	Diameter - In.		Thickness In.	Weight Per Foot - Lb ¹	
	External	Internal		Plain Ends (Calculated)	With Threads and Couplings (Nominal) ²
6	6.625	6.053	0.286	18.97	19.45
8	8.625	7.967	0.329	28.55	29.35
10	10.750	10.005	0.372	40.48	41.85
12	12.750	11.985	0.383	49.56	51.15
14	14.000	13.234	0.383	54.56	57.00
16	16.000	15.324	0.383	62.58	65.30
18	18.000	17.165	0.417	76.84	81.20
20	20.000	19.125	0.438	89.63	94.38
20	20.000	19.000	0.500*	102.10	106.62
22	22.000	21.125	0.438	98.77	
22	22.000	21.000	0.500*	112.57	
24	24.000	23.125	0.438	107.96	
24	24.000	23.000	0.500*	123.04	
26	26.000	25.125	0.438	117.12	
26	26.000	25.000	0.500*	133.51	
28	28.000	27.125	0.438	126.27	
28	28.000	27.000	0.500*	143.99	
30	30.000	29.125	0.438	135.42	
30	30.000	29.000	0.500*	154.46	

¹Manufacturing weight tolerance is 10 per cent over and 3.5 per cent under nominal weight for pipe ~20 in. in size and +10 per cent of nominal weight for larger sizes.

²Based on length of 20 ft. including coupling. Threaded pipe has 8 threads per inch.

*Thickness indicated is believed to be best practice. If soil and water conditions are unusually favorable tighter pipe may be used if permitted in the purchaser's specifications.

Note: Welded joints advocated for pipe larger than 20 in. in diameter; also for smaller diameter pipe, where applicable, to obtain clearance and maintain uniform grout thickness.

TECH-F-8 Capacities of Tanks of Various Dimensions

Diam.	Gals.	Area Sq. Ft.	Diam.	Gals.	Area Sq. Ft.	Diam.	Gals.	Area Sq. Ft.	Diam.	Gals.	Area Sq. Ft.
1'	5.87	.785	4' 2"	102.00	13.635	10'	587.52	78.54	21' 6"	2715.80	363.05
1' 1"	6.89	.922	4' 3"	106.12	14.186	10' 3"	617.26	82.52	21' 9"	2779.30	371.54
1' 2"	8.00	1.069	4' 4"	110.32	14.748	10' 6"	640.74	86.59	22'	2843.60	380.13
1' 3"	9.18	1.277	4' 5"	114.61	15.321	10' 9"	678.95	90.76	22' 3"	2908.60	388.82
1' 4"	10.44	1.396	4' 6"	118.97	15.90	11'	710.90	95.03	22' 6"	2974.30	397.61
1' 5"	11.79	1.576	4' 7"	123.42	16.50	11' 3"	743.58	99.40	22' 9"	3040.80	406.49
1' 6"	13.22	1.767	4' 8"	127.95	17.10	11' 6"	776.99	103.87	23'	3108.00	415.48
1' 7"	14.73	1.969	4' 9"	132.56	17.72	11' 9"	811.14	108.43	23' 3"	3175.90	424.56
1' 8"	16.32	2.182	4' 10"	137.25	18.35	12'	846.03	113.10	23' 6"	3244.60	433.74
1' 9"	17.99	2.405	4' 11"	142.02	18.99	12' 3"	881.65	117.86	23' 9"	3314.00	443.01
1' 10"	19.75	2.640	5'	146.91	19.64	12' 6"	918.00	122.72	24'	3384.10	452.39
1' 11"	21.58	2.885	5' 1"	151.81	20.30	12' 9"	955.09	127.68	24' 3"	3455.00	461.86
2'	23.50	3.142	5' 2"	156.83	20.97	13'	992.91	132.73	24' 6"	3526.60	471.44
2' 1"	25.50	3.409	5' 3"	161.94	21.65	13' 3"	1031.50	137.89	24' 9"	3598.90	481.11
2' 2"	27.58	3.687	5' 4"	167.11	22.34	13' 6"	1070.80	142.14	25'	3672.00	490.87
2' 3"	29.74	3.976	5' 5"	172.38	23.04	13' 9"	1110.80	148.49	25' 3"	3745.80	500.74
2' 4"	31.99	4.276	5' 6"	177.71	23.76	14'	1151.50	153.94	25' 6"	3820.30	510.71
2' 5"	34.31	4.587	5' 7"	183.14	24.48	14' 3"	1193.00	159.48	25' 9"	3895.60	527.77
2' 6"	36.72	4.909	5' 8"	188.66	25.22	14' 6"	1235.30	165.13	26'	3971.60	530.93
2' 7"	39.21	5.241	5' 9"	194.25	25.97	14' 9"	1278.20	170.87	26' 3"	4048.40	541.19
2' 8"	41.78	5.585	5' 10"	199.92	26.73	15'	1321.90	176.71	26' 6"	4125.90	551.55
2' 9"	44.43	5.940	5' 11"	205.67	27.49	15' 3"	1366.40	182.65	26' 9"	4204.10	562.00
2' 10"	47.16	6.305	6"	211.51	28.27	15' 6"	1411.50	188.69	27'	4283.00	572.66
2' 11"	49.98	6.681	6' 3"	229.50	30.68	15' 9"	1457.40	194.83	27' 3"	4362.70	583.21
3'	52.88	7.069	6' 6"	248.23	35.18	16'	1504.10	201.06	27' 6"	4443.10	593.96
3' 1"	55.86	7.467	6' 9"	267.69	35.78	16' 3"	1551.40	207.39	27' 9"	4524.30	604.81
3' 2"	58.92	7.876	7'	287.88	38.48	16' 6"	1599.50	213.82	28'	4606.20	615.75
3' 3"	62.06	8.296	7' 3"	308.81	41.28	16' 9"	1648.40	220.35	28' 3"	4688.80	626.80
3' 4"	65.28	8.727	7' 6"	330.48	44.18	19'	2120.90	283.53	28' 6"	4772.10	637.94
3' 5"	68.58	9.168	7' 9"	352.88	47.17	19' 3"	2177.10	291.04	28' 9"	4856.20	649.18
3' 6"	71.97	9.621	8'	376.01	50.27	19' 6"	2234.00	298.65	29'	4941.00	660.52
3' 7"	75.44	10.085	8' 3"	399.80	53.46	19' 9"	2291.70	306.35	29' 3"	5026.60	671.96
3' 8"	78.99	10.559	8' 6"	424.48	56.75	20'	2350.10	314.16	29' 6"	5112.90	683.49
3' 9"	82.62	11.045	8' 9"	449.82	60.13	20' 3"	2409.20	322.06	29' 9"	5199.90	695.13
3' 10"	86.33	11.541	9"	475.89	63.62	20' 6"	2469.10	330.06	30'	5287.70	706.86
3' 11"	90.13	12.048	9' 3"	502.70	67.20	20' 9"	2529.60	338.16	30' 3"	5376.20	718.69
4'	94.00	12.566	9' 6"	530.24	70.88	21'	2591.00	346.36	30' 6"	5465.40	730.62
4' 1"	97.96	13.095	9' 9"	558.51	74.66	21' 3"	2653.00	346.36	30' 9"	5555.40	742.64

To find the capacity of tanks greater than shown above, find a tank of one-half the size desired, and multiply its capacity by four, or find one one-third the size desired and multiply its capacity by 9.

Chart 10 Capacity of Round Tanks (per foot of depth)

Dimensions in Feet	Contents in Gallons for Depth in Feet of:							
	1'	4'	5'	6'	8'	10'	11'	12'
4 X 4	119.68	479.	598.	718.	957.	1197.	1316.	1436.
5 X 5	187.00	748.	935.	1202.	1516.	1870.	2057.	2244.
6 X 6	269.28	1077.	1346.	1616.	2154.	2693.	2968.	3231.
7 X 7	366.52	1466.	1833.	2199.	2922.	3665.	4032.	4398.
8 X 8	478.72	1915.	2394.	2872.	3830.	4787.	5266.	5745.
9 X 9	605.88	2424.	3029.	3635.	4847.	6059.	6665.	7272.
10 X 10	748.08	2992.	3740.	4488.	5984.	7480.	8228.	8976.
11 X 11	905.08	3620.	4525.	5430.	7241.	9051.	9956.	10861.
12 X 12	1077.12	4308.	5386.	6463.	8617.	10771.	11848.	12925.

To find the capacity of a depth not given, multiply the capacity for one foot by the required depth in feet.

Chart 11 Capacity of Square Tanks

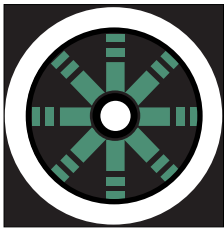
Capacities of Tanks of Various Dimensions

Diameter	Gallons Per Foot of Length When Tank is Filled								
	1/10	1/5	3/10	2/5	1/2	3/5	7/10	4/5	9/10
1 ft.	.3	.8	1.4	2.1	2.9	3.6	4.3	4.9	5.5
2 ft.	1.2	3.3	5.9	8.8	11.7	14.7	17.5	20.6	22.2
3 ft.	2.7	7.5	13.6	19.8	26.4	33.0	39.4	45.2	50.1
4 ft.	4.9	13.4	23.8	35.0	47.0	59.0	70.2	80.5	89.0
5 ft.	7.6	20.0	37.0	55.0	73.0	92.0	110.0	126.0	139.0
6 ft.	11.0	30.0	53.0	78.0	106.0	133.0	158.0	182.0	201.0
7 ft.	15.0	41.0	73.0	107.0	144.0	181.0	215.0	247.0	272.0
8 ft.	19.0	52.0	96.0	140.0	188.0	235.0	281.0	322.0	356.0
9 ft.	25.0	67.0	112.0	178.0	238.0	298.0	352.0	408.0	450.0
10 ft.	30.0	83.0	149.0	219.0	294.0	368.0	440.0	504.0	556.0
11 ft.	37.0	101.0	179.0	265.0	356.0	445.0	531.0	610.0	672.0
12 ft.	44.0	120.0	214.0	315.0	423.0	530.0	632.0	741.0	800.0
13 ft.	51.0	141.0	250.0	370.0	496.0	621.0	740.0	850.0	940.0
14 ft.	60.0	164.0	291.0	430.0	576.0	722.0	862.0	989.0	1084.0
15 ft.	68.0	188.0	334.0	494.0	661.0	829.0	988.0	1134.0	1253.0

Chart 12 Cylindrical Tanks Set Horizontally and Partially Filled

Diam. In.	Length of Cylinder																			Diam. In.
	1"	1'	5'	6'	7'	8'	9'	10'	11'	12'	13'	14'	15'	16'	17'	18'	20'	22'	24'	
1		0.04	0.20	0.24	0.28	0.32	0.36	0.40	0.44	0.48	0.52	0.56	0.60	0.64	0.68	0.72	0.80	0.88	0.96	1
2	0.01	0.16	0.80	0.96	1.12	1.28	1.44	1.60	1.76	1.92	2.08	2.24	2.40	2.56	2.72	2.88	3.20	3.52	3.84	2
3	0.03	0.37	1.84	2.20	2.56	2.92	3.30	3.68	4.04	4.40	4.76	5.12	5.48	5.84	6.22	6.60	7.36	8.08	8.80	3
4	0.05	0.65	3.26	3.92	4.58	5.24	5.88	6.52	7.18	7.84	8.50	9.16	9.82	10.5	11.1	11.8	13.0	14.4	15.7	4
5	0.08	1.02	5.10	6.12	7.14	8.16	9.18	10.2	11.2	12.2	13.3	14.3	15.3	16.3	17.3	18.4	20.4	22.4	24.4	5
6	0.12	1.47	7.34	8.80	10.3	11.8	13.2	14.7	16.1	17.6	19.1	20.6	22.0	23.6	25.0	26.4	29.4	32.2	35.2	6
7	0.17	2.00	10.0	12.0	14.0	16.0	18.0	20.0	22.0	24.0	26.0	28.0	30.0	32.0	34.0	36.0	40.0	44.0	48.0	7
8	0.22	2.61	13.0	15.6	18.2	20.8	23.4	26.0	28.6	31.2	33.8	36.4	39.0	41.6	44.2	46.8	52.0	57.2	62.4	8
9	0.28	3.31	16.5	19.8	23.1	26.4	29.8	33.0	36.4	39.6	43.0	46.2	49.6	52.8	56.2	60.0	66.0	72.4	79.2	9
10	0.34	4.08	20.4	24.4	28.4	32.6	36.8	40.8	44.8	48.8	52.8	56.8	61.0	65.2	69.4	73.6	81.6	89.6	97.6	10
11	0.41	4.94	24.6	29.6	34.6	39.4	44.4	49.2	54.2	59.2	64.2	69.2	74.0	78.8	83.8	88.8	98.4	104.	118.	11
12	0.49	5.88	29.4	35.2	41.0	46.8	52.8	58.8	64.6	70.4	76.2	82.0	87.8	93.6	99.6	106	118.	129.	141.1	12
13	0.57	6.90	34.6	41.6	48.6	55.2	62.2	69.2	76.2	83.2	90.2	97.2	104.	110.	117.	124.	138.	152.	166.	13
14	0.67	8.00	40.0	48.0	56.0	64.0	72.0	80.0	88.0	96.0	104.	112.	120.	128.	136.	144.	160.	176.	192.	14
15	0.77	9.18	46.0	55.2	64.4	73.6	82.8	92.0	101.	110.	120.	129.	138.	147.	156.	166.	184.	202.	220.	15
16	0.87	10.4	52.0	62.4	72.8	83.2	93.6	104.	114	125.	135.	146.	156.	166.	177.	187.	208.	229.	250.	16
17	0.98	11.8	59.0	70.8	81.6	94.4	106.	118.	130.	142.	153.	163.	177.	189.	201.	212.	236.	260.	283.	17
18	1.10	13.2	66.0	79.2	92.4	106.	119.	132.	145.	158.	172.	185.	198.	211.	224.	240.	264.	290.	317.	18
19	1.23	14.7	73.6	88.4	103.	118.	132.	147.	162.	177.	192.	206.	221.	235.	250.	265.	294.	324.	354.	19
20	1.36	16.3	81.6	98.0	114.	130	147.	163.	180.	196.	212.	229	245.	261.	277.	294.	326.	359.	392.	20
21	1.50	18.0	90.0	108	126	144.	162.	180.	198.	216.	238.	252.	270.	288.	306.	324.	360.	396.	432.	21
22	1.65	19.8	99.0	119.	139.	158.	178.	198.	218.	238.	257.	277.	297.	317.	337.	356.	396.	436.	476.	22
23	1.80	21.6	108.	130.	151.	173.	194.	216.	238.	259.	281.	302.	324.	346.	367.	389.	432.	476.	518.	23
24	1.96	23.5	118.	141.	165.	188.	212.	235.	259.	282.	306.	330.	353.	376.	400.	424	470.	518.	564.	24
25	2.12	25.5	128.	153.	179.	204	230.	255.	281.	306.	332.	358.	383.	408.	434.	460.	510.	562.	612.	25
26	2.30	27.6	138.	166.	193.	221.	248.	276.	304.	331.	359.	386.	414.	442.	470.	496.	552.	608.	662.	26
27	2.48	29.7	148.	178.	208.	238.	267.	297.	326.	356.	386.	416.	426.	476.	504.	534.	594.	652.	712.	27
28	2.67	32.0	160	192.	224.	256.	288.	320.	352.	384.	416.	448.	480.	512.	544.	576.	640.	704.	768.	28
29	2.86	34.3	171.	206.	240.	274.	309.	343.	377.	412.	446.	480.	514.	548.	584.	618.	686.	754.	824.	29
30	3.06	36.7	183	220.	257.	294.	330.	367.	404.	440.	476.	514.	550.	588.	624.	660.	734.	808.	880.	30
32	3.48	41.8	209	251.	293.	334.	376.	418.	460.	502.	544.	586.	628.	668.	710.	752.	836.	920.	1004.	32
34	3.93	47.2	236.	283.	330.	378.	424.	472.	520.	566.	614.	660.	708.	756.	802.	848.	944.	1040.	1132.	34
36	4.41	52.9	264.	317.	370.	422.	476.	528.	582.	634.	688.	740.	792.	844.	898.	952.	1056.	1164.	1268.	36

Chart 13 Capacities, in U.S. Gallons of Cylinders of Various Diameters and Lengths



Section TECH-G

Motor Data

TECH-G-1 Motor Enclosures

The selection of a motor enclosure depends upon the ambient and surrounding conditions. The two general classifications of motor enclosures are open and totally enclosed. An open motor has ventilating openings which permit passage of external air over and around the motor windings. A totally enclosed motor is constructed to prevent the free exchange of air between the inside and outside of the frame, but not sufficiently enclosed to be termed air-tight.

These two categories are further broken down by enclosure design, type of insulation, and/or cooling method. The most common of these types are listed below.

Open Dripproof - An open motor in which all ventilating openings are so constructed that drops of liquid or solid particles falling on the motor at any angle from 0 to 15 degrees from vertical cannot enter the machine. This is the most common type and is designed for use in nonhazardous, relatively clean, industrial areas.

Encapsulated - A dripproof motor with the stator windings completely surrounded by a protective coating. An encapsulated motor offers more resistance to moisture and/or corrosive environments than an ODP motor.

Totally Enclosed, Fan-Cooled - A enclosed motor equipped for external cooling by means of a fan integral with the motor, but external to the enclosed parts. TEFC motors are designed for use in extremely wet, dirty, or dusty areas.

Explosion-Proof, Dust-Ignition-Proof - An enclosed motor whose enclosure is designed to withstand an explosion of a specified dust, gas, or vapor which may occur within the motor and to prevent the ignition of this dust, gas, or vapor surrounding the motor. A motor manufacturer should be consulted regarding the various classes and groups of explosion-proof motors available and the application of each.

Motor insulation is classified according to the total allowable temperature. This is made up of a maximum ambient temperature plus a maximum temperature rise plus allowances for hot spots and service factors. Class B insulation is the standard and allows for a total temperature of 130°C. The maximum ambient is 40°C, and the temperature rise is 70°C, for ODP motors and 75°C for TEFC motors.

Design L, 60 cycles, class B insulation system, open type, 1.15 service factor.

hp	speed, rpm		
	3600	1800	1200
3/4	-	-	145T
1	-	143T	182T
1 1/2	143T	145T	184T
2	145T	182T	-
3	182T	184T	-
5	184T	213T	-
7 1/2	213T	215T	-

TECH-G-2 NEMA Frame Assignments

SINGLE-PHASE MOTORS Horizontal and Vertical
open _____

POLYPHASE SQUIRREL-CAGE MOTORS Horizontal and Vertical
open type _____

Designs A and B - class B insulation system, open type 1.15 service factor, 60 cycles.

hp	speed, rpm			
	3600	1800	1200	900
1/2	-	-	-	143T
3/4	-	-	143T	145T
1	-	143T	145T	182T
1 1/2	143T	145T	182T	184T
2	145T	145T	184T	213T
3	145T	182T	213T	215T
5	182T	184T	215T	254T
7 1/2	184T	213T	254T	256T
10	213T	215T	256T	284T
15	215T	254T	284T	286T
20	254T	256T	286T	324T
25	256T	284T	324T	326T
30	284TS	286T	326T	364T
40	286TS	324T	364T	365T
50	324TS	326T	365T	404T
60	326TS	364TS	404T	405T
75	364TS	365TS	405T	444T
100	365TS	404TS	444T	445T
125	404TS	405TS	445T	-
150	405TS	444TS	-	-
200	444TS	454TS	-	-
250	445TS*	-	-	-

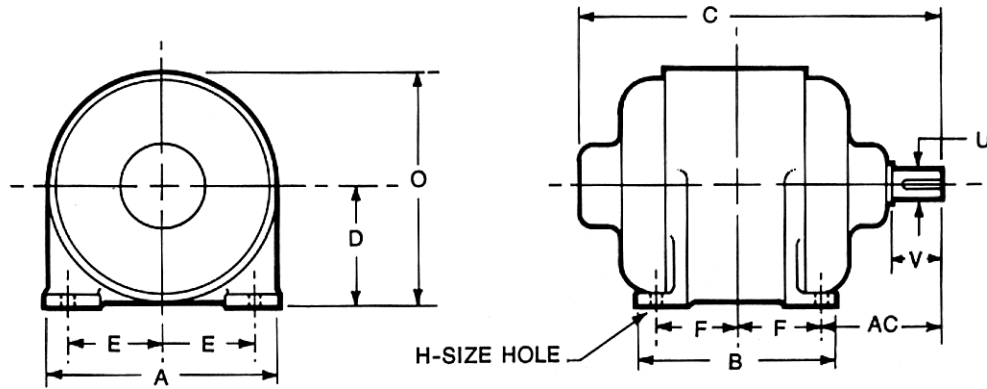
*The 250 hp rating at the 3600 rpm speed has a 1.0 service factor

fan cooled _____

Designs A and B - class B insulation system totally-enclosed fan-cooled type, 1.00 service factor, 60-cycles.

hp	speed, rpm			
	3600	1800	1200	900
1/2	-	-	-	143T
3/4	-	-	143T	145T
1	-	143T	145T	182T
1 1/2	143T	145T	182T	184T
2	145T	145T	184T	213T
3	182T	182T	213T	215T
5	184T	184T	215T	254T
7 1/2	213T	213T	254T	256T
10	215T	215T	256T	284T
15	254T	254T	284T	286T
20	256T	256T	286T	324T
25	284TS	284T	324T	326T
30	286TS	286T	326T	364T
40	324TS	324T	364T	365T
50	326TS	326T	365T	404T
60	364TS	364TS	404T	405T
75	365TS	365TS	405T	444T
100	405TS	405TS	444T	445T
125	444TS	444TS	445T	-
150	445TS	445TS	-	-

TECH-G-3 NEMA Frame Dimensions



Motor Frame	H.P. (Open)			H.P. (Enclosed)				A Max.	B Max.	C (Approx.)		D	E	F	H	O (Approx.)		U	Keyway	V Min.	AC	Bolts		Wt. (Approx.)		
	900	1200	1800	3600	900	1200	1800			3600	Open					Encl.	Open					Encl.	Dia.	Lg.	Open	Encl.
143T	1/2	3/4	1	1 1/2	1/2	3/4	1	1 1/2	7	6	12	12 1/2	3 1/2	2 3/4	2	1 1/32	6 7/8	7	7/8	3/16 x 3/32	2	4 1/2	1/4		40	45
145T	3/4	1	1 1/2 - 2	2-3	3/4	1	1 1/2 - 2	2	7	6	12 1/2	13 1/2	3 3/4	2 3/4	2 3/4	1 1/32	6 7/8	7	7/8	3/16 x 3/32	2	4 1/2	1/4		45	50
182T	1	1 1/2	3	5	1	1 1/2	3	3	9	6 1/2	13	14 1/2	4 1/2	3 3/4	2 1/4	1 13/32	9 1/8	9 1/4	1 1/8	1/4 x 1/8	2 1/2	5 1/2	5/16	1	65	79
184T	1 1/2	2	5	7 1/2	1 1/2	2	5	5	9	7 1/2	14	15 1/2	4 1/2	3 3/4	2 3/4	1 13/32	9 1/8	9 1/4	1 1/8	1/4 x 1/8	2 1/2	5 1/2	5/16	1	80	95
213T	2	3	7 1/2	10	2	3	7 1/2	7 1/2	10 1/2	7 1/2	16	18	5 1/4	4 1/4	2 3/4	1 13/32	10 3/4	10 7/8	1 3/8	5/16 x 3/32	3 1/8	6 7/8	5/16	1	120	140
215T	3	5	10	15	3	5	10	10	10 1/2	9	17 1/2	19 1/2	5 1/4	4 1/4	3 3/4	1 13/32	10 3/4	10 7/8	1 3/8	5/16 x 3/32	3 3/8	6 7/8	5/16	1	140	160
254T	5	7 1/2	15	20	5	7 1/2	15	15	12 1/2	10 3/4	20 1/2	22 1/2	6 1/4	5	4 1/8	1 17/32	12 5/8	12 3/4	1 5/8	3/8 x 3/16	3 3/4	8 1/4	3/8	1 1/4	200	235
256T	7 1/2	10	20	25	7 1/2	10	20	20	12 1/2	12 1/2	22 1/2	24	6 1/4	5	5	1 17/32	12 5/8	12 3/4	1 5/8	3/8 X 3/16	3 3/4	8 1/4	3/8	1 1/4	235	270
284T	10	15	25		10	15	25		14	12 1/2	23 1/2	25 1/2	7	5 1/2	4 3/4	1 17/32	14	14 3/8	1 7/8	1/2 x 1/4	4 3/8	9 3/8	3/8	1 1/2	295	370
284TS				30				25	14	12 1/2	22	24 1/2	7	5 1/2	4 3/4	1 17/32	14	14 3/8	1 5/8	3/8 x 3/16	3	8	3/8	1 1/2	255	340
286T	15	20	30		15	20	30		14	14	25	27	7	5 1/2	5 1/2	1 17/32	14	14 3/8	1 7/8	1/2 x 1/4	4 3/8	9 3/8	3/8	1 1/2	340	405
286TS				40				30	14	14	23 1/2	26	7	5 1/2	5 1/2	1 17/32	14	14 3/8	1 5/8	3/8 x 3/16	3	8	3/8	1 1/2	295	395
324T	20	25	40		20	25	40		16	14	26	28 1/2	8	6 1/4	5 1/4	2 1/32	16	16 5/8	2 1/8	1/2 x 1/4	5	10 1/2	1 1/2	1 3/4	440	520
324TS				50				40	16	14	24 1/2	27	8	6 1/4	5 1/4	2 1/32	16	16 5/8	1 7/8	1/2 x 1/4	3 1/2	9	1 1/2	1 3/4	445	500
326T	25	30	50		25	30	50		16	15 1/2	27 1/2	30	8	6 1/4	6	2 1/32	16	16 5/8	2 1/8	1/2 x 1/4	5	10 1/2	1 1/2	1 3/4	435	580
326TS				60				50	16	15 1/2	26	28 1/2	8	6 1/4	6	2 1/32	16	16 5/8	1 7/8	1/2 x 1/4	3 1/2	9	1 1/2	1 3/4	480	560
364T	30	40			30	40			18	15 1/4	29	33	9	7	5 5/8	2 1/32	18	18 1/2	2 3/8	5/8 x 5/16	5 5/8	11 3/4	1 1/2	1 3/4	605	755
364TS			60	75			60	60	18	15 1/4	27	31	9	7	5 5/8	2 1/32	18	18 1/2	1 7/8	1/2 x 1/4	3 1/2	9 5/8	1 1/2	1 3/4	670	740
365T	40	50			40	50			18	16 1/4	30	34	9	7	6 1/8	2 1/32	18	18 1/2	2 3/8	5/8 x 5/16	5 5/8	11 3/4	1 1/2	1 3/4	665	835
365TS			75	100			75	75	18	16 1/4	28	32	9	7	6 1/8	2 1/32	18	18 1/2	1 7/8	1/2 x 1/4	3 1/2	9 5/8	1 1/2	1 3/4	730	820
404T	50	60			50	60			20	16 1/4	32 1/2	37	10	8	6 1/8	1 13/16	20	20 5/8	2 1/8	3/4 x 3/8	7	13 1/8	5/8	2 1/4	830	1050
404TS			100	125					20	16 1/4	29 1/2	34	10	8	6 1/8	1 13/16	20	20 5/8	2 1/8	1/2 x 1/4	4	10 7/8	5/8	2 1/4	870	1050
405T	60	75			60	75			20	17 3/4	34	38 1/2	10	8	6 7/8	1 13/16	20	20 5/8	2 1/8	3/4 x 3/8	7	13 1/8	5/8	2 1/4	930	1160
405TS			125	150			100	100	20	17 3/4	31	35 1/2	10	8	6 7/8	1 13/16	20	20 5/8	2 1/8	1/2 x 1/4	4	10 7/8	5/8	2 1/4	950	1150
444T	75	100			75	100			22	18 1/2	38	42 1/2	11	9	7 1/4	1 13/16	22 3/8	23 1/8	3 3/8	7/8 x 7/16	8 1/4	16	5/8	2 1/4	1165	1440
444TS			150	200			125	125	22	18 1/2	34	38 1/2	11	9	7 1/4	1 13/16	22 3/8	23 1/8	3 3/8	5/8 x 5/16	4 1/2	12 1/4	5/8	2 1/4	1050	1440
445T	100	125			100	125			22	20 1/2	40	44 1/2	11	9	8 1/4	1 13/16	22 3/8	23 1/8	3 3/8	7/8 x 7/16	8 1/4	16	5/8	2 1/4	1370	1650
445TS			200	250			150	150	22	20 1/2	36	41	11	9	8 1/4	1 13/16	22 3/8	23 1/8	3 3/8	5/8 x 5/16	4 1/2	12 1/4	5/8	2 1/4	1250	1615
447T									22	23 1/4	43 1/2	48	11	9	10	1 13/16	22 3/8	23 1/8	3 3/8	7/8 x 7/16	8 1/4	16	5/8	2 1/4	1800	2260
447TS									22	23 1/4	40 1/2	46 1/2	11	9	10	1 13/16	22 3/8	23 1/8	2 3/8	5/8 x 5/16	4 1/2	12 1/4	5/8	2 1/4	1800	2260
56									6 1/2	3 7/8	10 1/2		3 1/2	2 7/16	1 1/2	1 11/32	6 7/8		5/8	3/16 x 3/32	1 7/8	4 3/8	1/4	1		
182	1/2	3/4	1	1 1/2	1/2	3/4	1	1 1/2	9	6 1/2	12 1/2	14 1/2	4 1/2	3 3/4	2 1/4	1 13/32	9	9	7/8	3/16 x 3/32	2	5	5/16	1	60	70
184	3/4	1-1 1/2	1 1/2-2	2-3	3/4	1-1 1/2	1 1/2-2	2-3	9	7 1/2	13 1/2	15 1/2	4 1/2	3 3/4	2 3/4	1 13/32	9	9	7/8	3/16 X 3/32	2	5	5/16	1	70	80
213	1-1 1/2	2	3	5	1-1 1/2	2	3	5	10 1/2	7 1/2	15 1/2	17 1/2	5 1/4	4 1/4	2 3/4	1 13/32	10 1/2	10 5/8	1 1/8	1/2 x 1/8	2 3/4	6 1/2	5/16	1	105	125
215	2	3	5	7 1/2	2	3	5	7 1/2	10 1/2	9	17	19	5 1/4	4 1/4	3 1/2	1 13/32	10 1/2	10 5/8	1 1/8	1/2 x 1/8	2 3/4	6 1/2	5/16	1	115	140
254U	3	5	7 1/2	10	3	5	7 1/2	10	12 1/2	10 3/4	20 1/2	22	6 1/4	5	4 1/8	1 17/32	12 5/8	13 1/8	1 3/8	5/16 x 3/32	3 1/2	8	3/8	1 1/4	180	210
256U	5	7 1/2	10	15	5	7 1/2	10	15	12 1/2	12 1/2	22 1/2	24	6 1/4	5	5	1 17/32	12 5/8	13 1/8	1 3/8	5/16 x 3/32	3 1/2	8	3/8	1 1/4	210	245
284U	7 1/2	10	15	20	7 1/2	10	15		14	12 1/2	24	25	7	5 1/2	4 3/4	1 17/32	14	14 3/8	1 5/8	3/8 x 3/16	4 5/8	9 5/8	3/8	1 1/2	280	330
286U	10		20	25	10		20	20	14	14	25 1/2	26 1/2	7	5 1/2	5 1/2	1 17/32	14	14 3/8	1 5/8	3/8 X 3/16	4 5/8	9 5/8	3/8	1 1/2	325	365
324U		15	25			15	25	25	16	14	26 1/2	28	8	6 1/4	5 1/4	2 1/32	16	16 3/4	1 7/8	1/2 x 1/4	5 5/8	10 7/8	1 1/2	1 3/4	380	480
324S				30					16	14	24 1/2	25 1/2	8	6 1/4	5 1/4	2 1/32	16	16 3/4	1 7/8	3/8 X 3/16	3	8 1/2	1 1/2	1 3/4	380	480
326U	15	20	30		15	20	30		16	15 1/2	28	29 1/2	8	6 1/4	6	2 1/32	16	16 3/4	1 7/8	1/2 x 1/4	5 5/8	10 7/8	1 1/2	1 3/4	430	560
326S				40				30	16	15 1/2	26	27	8	6 1/4	6	2 1/32	16	16 3/4	1 5/8	3/8 X 3/16	3	8 1/2	1 1/2	1 3/4	430	560
364U	20	25	40		20	25	40		18	15 1/4	29 1/2	34	9	7	5 5/8	2 1/32	18 1/4	18 3/4	2 1/8	1/2 x 1/4	6 1/8	12 1/4	1 1/2	1 3/4	525	720
364US				50				40	18	15 1/4	27	31	9	7	5 5/8	2 1/32	18 1/4	18 3/4	1 7/8	1/2 x 1/4	3 1/2	9 5/8	1 1/2	1 3/4	670	710
365U	25	30			25	30			18	16 1/4	30 1/2	35	9	7	6 1/8	2 1/32	18 1/4	18 3/4	2 1/8	1/2 x 1/4	6 1/8	12 1/4	1 1/2	1 3/4	580	785
365US			50	60			50	50	18	16 1/4	28	32	9	7	6 1/8	2 1/32	18 1/4	18 3/4	1 7/8	1/2 x						

TECH-G-4 Synchronous and Approximate Full Load Speed of Standard A.C. Induction Motors

NUMBER of POLES	60 CYCLE RPM		50 CYCLE RPM	
	SYNC.	F.L.	SYNC.	F.L.
2	3600	3500	3000	2900
4	1800	1770	1500	1450
6	1200	1170	1000	960
8	900	870	750	720
10	720	690	600	575
12	600	575	500	480
14	515	490	429	410
16	450	430	375	360
18	400	380	333	319
20	360	340	300	285
22	327	310	273	260
24	300	285	240	230
26	277	265	231	222
28	257	245	214	205
30	240	230	200	192

TECH-G-5 Full Load Amperes at Motor Terminals* Average Values for All Speeds and Frequencies

MOTOR HP	SINGLE-PHASE A-C		THREE PHASE A-C INDUCTION TYPE SQUIRREL CAGE & WOUND ROTOR			DIRECT CURRENT	
	115 VOLTS	230 VOLTS**	230 VOLTS**	460 VOLTS	575 VOLTS	120 VOLTS	240 VOLTS
1/2	9.8	4.9	2.0	1.0	.8	5.2	2.6
3/4	13.8	6.9	2.8	1.4	1.1	7.4	3.7
1	16	8	3.6	1.8	1.4	9.4	4.7
1 1/2	20	10	5.2	2.6	2.1	13.2	6.6
2	24	12	6.8	3.4	2.7	17	8.5
3	34	17	9.6	4.8	3.9	25	12.2
5	56	28	15.2	7.6	6.1	40	20
7 1/2	80	40	22	11	9	58	29
10	100	50	28	14	11	76	29
15			42	21	17	112	55
20			54	27	22	148	72
25			68	34	27	184	89
30			80	40	32	220	106
40			104	52	41	292	140
50			130	65	52	360	173
60			154	77	62	430	206
75			192	96	77	536	255
100			240	120	96		350
125			296	148	118		440
150			350	175	140		530
200			456	228	182		710
250			558	279	223		

* These values for full load current are for running at speeds usual for belted motors and motors with normal torque characteristics. Motors built for especially low speeds or high torques may require more running current, in which case the nameplate current rating should be used.

** For full-load currents of 208 and 200 volt motors, increase the corresponding 230 volt motor full-load current by 10 and 15 per cent respectively.

TECH-G-6 Motor Terms

AMPERE: a unit of intensity of electric current being produced in a conductor by the applied voltage.

FREQUENCY: the number of complete cycles per second of alternating current, e.g., 60 Hertz.

HORSEPOWER: the rate at which work is done. It is the result of the work done (stated in foot-pounds) divided by the time involved.

INERTIA: the property of physical matter to remain at rest unless acted on by some external force. Inertia usually concerns the driven load.

MOTOR EFFICIENCY: a measure of how effectively the motor turns electrical energy into mechanical energy. Motor efficiency is never 100% and is normally in the neighborhood of 85%.

POWER FACTOR: the ratio of the true power to the volt-amperes in an alternating current circuit or apparatus.

SERVICE FACTOR: a safety factor in some motors which allows the motor, when necessary, to deliver greater than rated horsepower.

SYNCHRONOUS SPEED & SLIP: the speed of an a-c motor at which the motor would operate if the rotor turned at the exact speed of the rotating magnetic field. However, in a-c induction motors, the rotor actually turns slightly slower. This difference is defined as slip and is expressed in percent of synchronous speed. Most induction motors have a slip of 1-3%.

TORQUE: that force which tends to produce torsion or rotation. In motors, it is considered to be the amount of force produced to turn the load, it is measured in lb.-ft.

VOLTAGE: a unit of electro-motive force. It is a force which, when applied to a conductor, will produce a current in the conductor.

APPROXIMATE RULES OF THUMB		MECHANICAL FORMULAS
At 1800 rpm, a motor develops 3 lb.- ft per hp.	At 230 volts, a single- phase motor draws 2.5 amp per hp.	Torque in lb-ft = $\frac{HP \times 5250}{RPM}$
At 1200 rpm, a motor develops 4.5 lb-ft per hp.	At 230 volts, a single- phase motor draws 5 amp per hp.	Hp= $\frac{\text{Torque} \times RPM}{5250}$
At 575 volts, a 3-phase motor draws 1 amp per hp.	At 115 volts, a single- phase motor draws 10 amp per hp.	RPM = $\frac{120 \times \text{Frequency}}{\text{No. of poles}}$
At 460 volts, a 3-phase motor draws 1.25 amp per hp.		

Average Efficiencies and Power Factors of Electric Motors							
kW	Efficiency %			Power Factor			Full Load Amps on 3ph 415V
	Full Load	$\frac{3}{4}$ Load	$\frac{1}{2}$ Load	Full Load	$\frac{3}{4}$ Load	$\frac{1}{2}$ Load	
0.75	74	73	69	0.72	0.65	0.53	2.0
1.5	79	78.5	76	0.83	0.78	0.69	3.2
3	82.5	82	80.5	0.85	0.80	0.73	6.0
5.5	84.5	84.5	83.5	0.87	0.82	0.75	10.5
7.5	85.5	85.5	84.5	0.87	0.83	0.76	14
11	87	87	85.5	0.88	0.84	0.77	20
18.5	88.5	88.5	87	0.89	0.85	0.79	33
30	90	89.5	88	0.89	0.86	0.80	52
45	91	90.5	89	0.89	0.86	0.80	77
75	92	91.5	90	0.90	0.87	0.81	126

Required Value	Direct Current	Single Phases	Two-Phase 4-Wire	Three Phase
HP Output	$\frac{I \times E \times \text{Eff}}{746}$	$\frac{I \times E \times \text{Eff} \times \text{PF}}{746}$	$\frac{I \times E \times 2 \times \text{Eff} \times \text{Pf}}{746}$	$\frac{I \times E \times 1.73 \times \text{Eff} \times \text{PF}}{746}$

TECH-G-7 Electrical Conversion Formulae

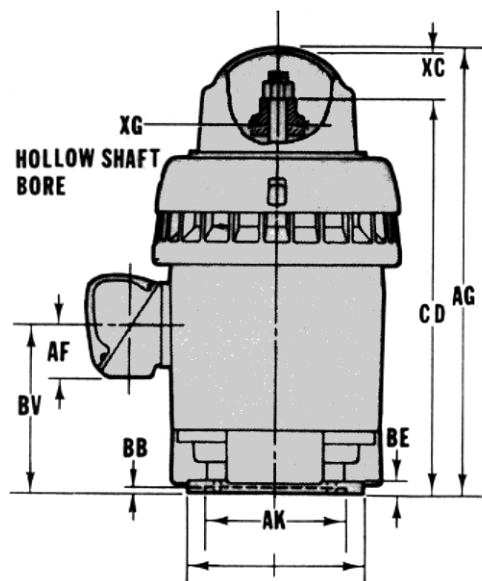
TO FIND	DIRECT CURRENT	ALTERNATING CURRENT	
		Single Phase	Three Phase
Amperes when horsepower (input) is known	$\frac{HP \times 746}{E \times \text{Eff}}$	$\frac{HP \times 746}{E \times \text{Eff} \times \text{P.F.}}$	$\frac{HP \times 746}{1.73 \times E \times \text{Eff} \times \text{P.F.}}$
Amperes when kilowatts is known	$\frac{KW \times 1000}{E}$	$\frac{KW \times 1000}{E \times \text{P.F.}}$	$\frac{KW \times 1000}{1.73 \times E \times \text{P.F.}}$
Amperes when kva is known		$\frac{kva \times 1000}{E}$	$\frac{kva \times 1000}{1.73 \times E}$
Kilowatts	$\frac{I \times E}{1000}$	$\frac{I \times E \times \text{P.F.}}{1000}$	$\frac{1.73 \times I \times E \times \text{P.F.}}{1000}$
Kva		$\frac{I \times E}{1000}$	$\frac{1.73 \times I \times E}{1000}$
P.F.		$\frac{KW}{Kva}$	$\frac{KW}{Kva}$
Horespower (output)	$\frac{I \times E \times \text{Eff}}{746}$	$\frac{I \times E \times \text{Eff} \times \text{P.F.}}{746}$	$\frac{1.73 \times I \times E \times \text{Eff} \times \text{P.F.}}{746}$

I = Amperes
E = Volts
HP= Horsepower

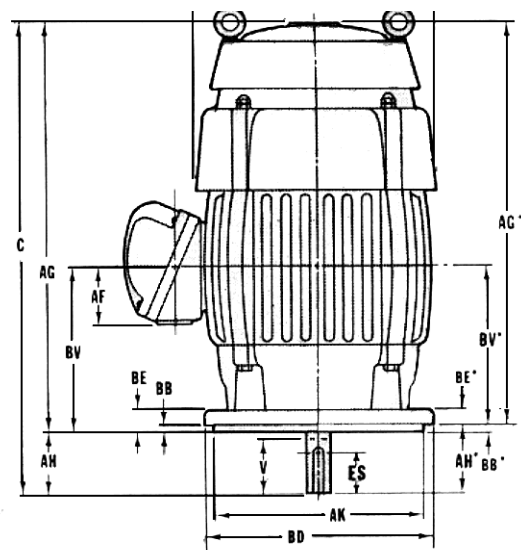
Eff= Efficiency (decimal)
P.F = Power Factor

Kva = Kilovolt- amperes
kW = Kilowatts

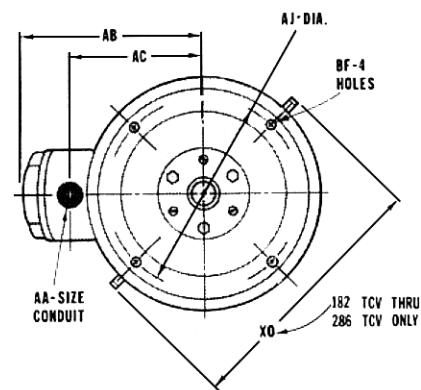
TECH-G-8 Vertical Motors



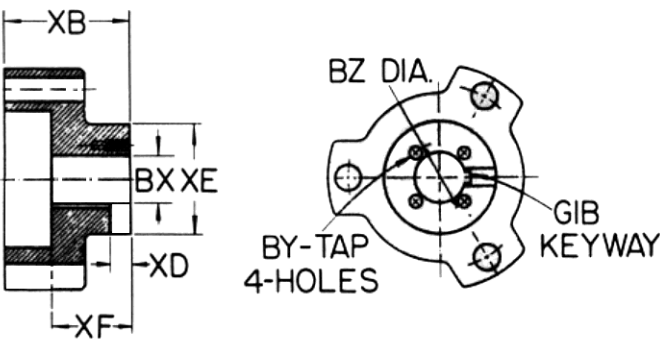
VHS VERTICAL HOLLOWSHAFT
 Pump shaft thru motor and coupled below motor with impeller adjustment made at top of motor.



VHS VERTICAL SOLID SHAFT
 Pump shaft coupled to shaft extension below motor. Impeller adjustment at coupling



NOTE: The following dimensions may vary upon vendor selection and design: XC, CD, AG, AF, BV, C.

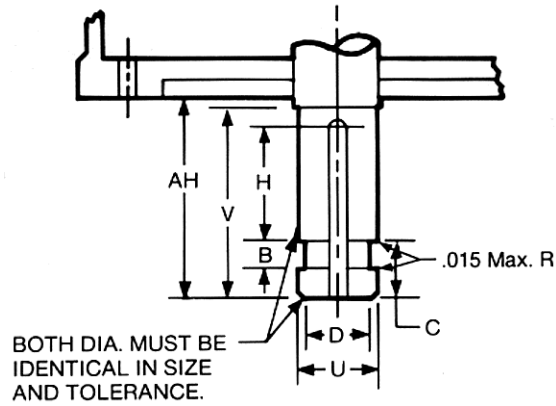


VERTICAL HOLLOWSHAFT
 NEMA dimensions for common top drive coupling sizes.

DIMENSIONS				
Top Shaft Dia.	BX Bore	BZ Dia. BC	SQ Key Size	BY Tap Size
3/4	0.751	1 3/8	3/16	10-32
1	1.001	3/8	1/4	10-32
1 3/16	1.188	1 3/4	1/4	1/4 - 20
1 1/2	1.501	2 1/8	3/8	1/4 - 20
1 15/16	1.938	2 1/2	1/2	1/4 - 20
2 3/16	2.188	3 1/4	1/2	3/8 - 16

NEMA SOLID SHAFT

NEMA DIMENSIONS FOR COMMON SOLID SHAFT EXTENSION SIZES.



Motor Shaft Dia. U	DIMENSIONS						Nominal Keyway	Pump Shaft Diameters
	AH	V	H	B	C	D		
$\frac{7}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$	$\frac{3}{16} \times \frac{3}{22}$	$\frac{7}{8}$
$1\frac{1}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$	1	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{5}{16}$	$\frac{1}{4} \times \frac{1}{8}$	$\frac{7}{8}, 1$
$1\frac{5}{8}$	$4\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{8} \times \frac{3}{16}$	$\frac{7}{8}, 1, 1\frac{3}{16}, 1\frac{1}{2}$
$2\frac{1}{8}$	$4\frac{1}{2}$	$4\frac{1}{4}$	$2\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{4}$	$1\frac{3}{4}$	$\frac{1}{2} \times \frac{1}{4}$	$1, 1\frac{3}{16}, 1\frac{1}{2}, 1\frac{15}{16}$
$2\frac{5}{8}$	5	5	$3\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{4}$	$2\frac{1}{4}$	$\frac{5}{8} \times \frac{5}{16}$	$2\frac{3}{16}$
$2\frac{7}{8}$	7	$6\frac{1}{2}$	5	$\frac{1}{2}$	1	$2\frac{3}{8}$	$\frac{3}{4} \times \frac{3}{8}$	$2\frac{3}{16}, 2\frac{11}{16}$
$3\frac{1}{8}$	7	7	$4\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{2}$	$2\frac{5}{8}$	$\frac{3}{4} \times \frac{3}{8}$	$2\frac{3}{16}, 2\frac{11}{16}, 2\frac{15}{16}$

HEADSHAFT COUPLINGS

WITH VERTICAL HOLLOWSHAFT MOTOR:

Impeller adjustment made on adjusting nut above motor (under motor canopy and bolted to top drive coupling).

1. Sleeve type (lineshaft) coupling.
2. Rigid flanged coupling (Type AR).
3. No coupling-straight shaft
(not recommended due to difficult Installation/disassembly of head and motor).

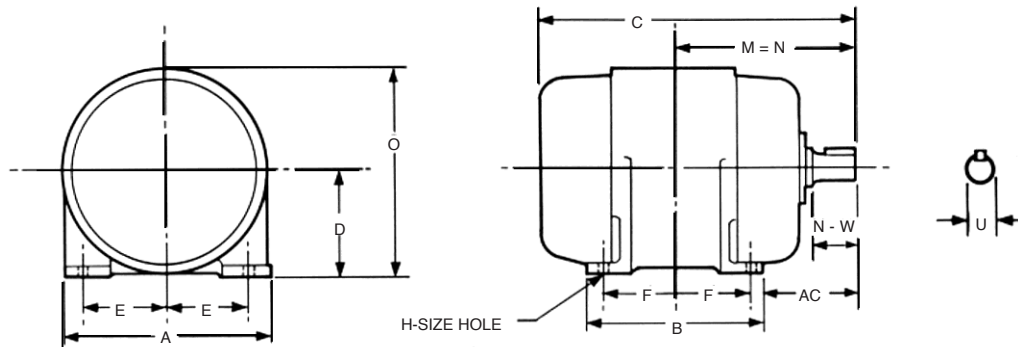
WITH VERTICAL SOLID SHAFT MOTOR:

Impeller adjustment made on adjusting plate of coupling without removal of motor canopy. (VSS motors also provide a lesser tolerance of shaft run-out which coincides with mechanical seal recommendations).

1. Adjustable coupling (Type A).
2. Adjustable spacer coupling
(Type AS-recommended for applications with mechanical seals. The mechanical seal can be removed without disengaging motor).

TECH-G-9 I.E.C. Motor Frames

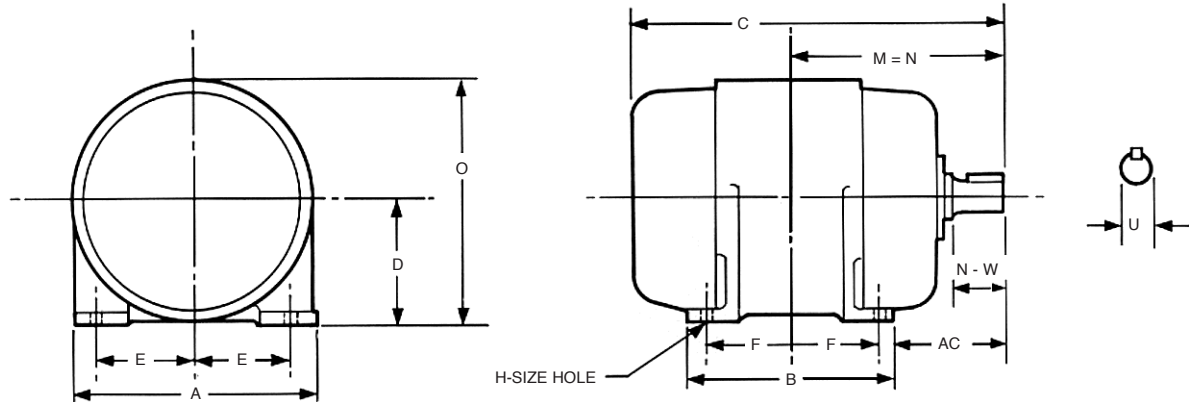
IPP44 TOTALLY ENCLOSED & FLAMEPROOF
(Similar to NEMA TEFC & Explosion Proof)



DIMENSIONS																
I.E.C. Frames	Poles	Units	A Max.	B Max.	C Approx.	D	E	F	H	M & N	O Approx.	U Nominal Tolerance		N & W	AC	Weight Approx.
D80-19	All	mm	157	130	245	80	63	50	10	140	185	19	j6	40	90	10 kg
E80-19		Inches	6 1/8	5 1/8	10	3.15	2 1/2	2	3/8	5 1/2	7 1/4			1 1/2	3 1/2	20 Lbs
D90S24	"	mm	180	130	300	90	70	50	10	156	210	24	j6	50	106	20 kg.
E900S24		Inches	7	5 1/8	10	3.54	2 3/4	2	3/8	6 3/16	8 1/4			2	4 3/16	45 kg.
D90L24	"	mm	180	155	320	90	70	63	10	169	210	24	j6	50	106	22 kg.
E90L24		Inches	7	6 1/8	12 1/2	3.54	2 3/4	2 1/2	3/8	6 11/16	8 1/4			2	4 3/16	50 Lbs.
D100L28	"	mm	205	180	380	100	80	70	12	193	230	28	j6	60	123	30 kg.
E100L28		Inches	8	7	15	3.94	3 1/8	2 3/4	1 5/32	7 5/8	9			2 3/8	4 7/8	65 Lbs.
D112M28	"	mm	240	185	380	112	95	70	12	200	250	28	j6	60	130	44 kg.
E112M28		Inches	9 1/2	7 1/4	15	4.41	3 3/4	2 3/4	1 5/32	7 7/8	10			2 3/8	5 1/8	100 Lbs
D132S38	"	mm	266	185	440	132	108	70	12	239	290	38	k6	80	169	65 kg.
E132S38		Inches	10 1/2	7 1/4	17 1/2	5.20	4 1/4	2 3/4	1 5/32	9 3/8	11 1/2			3 1/8	6 5/8	145 Lbs
D132M38	"	mm	266	225	480	132	108	89	12	258	290	38	k6	80	169	90 kg.
E132M38		Inches	10 1/2	8 3/4	19	5.20	4 1/4	3 1/2	1 5/32	10 1/8	11 1/2			3 1/8	6 5/8	100 Lbs.
D160M42	"	mm	318	267	580	160	127	105	15	323	360	42	k6	110	218	120 kg.
E160M42		Inches	12 1/2	10 1/2	23	6.30	5	4 1/8	1 9/32	12 1/4	14			4 3/8	8 7/8	265 Lbs.
D160L42	"	mm	318	311	620	160	127	127	15	345	360	42	k6	110	218	150 kg.
E160L42		Inches	12 1/2	12 1/4	24 1/2	6.30	5	5	1 9/32	13 3/8	14			4 3/8	8 7/8	330 Lbs
D180M48	"	mm	356	300	650	180	140	121	15	352	400	48	k6	110	231	175 kg.
E180M48		Inches	14	11 3/4	25 1/2	7.09	5 1/2	4 3/4	1 9/32	13 7/8	15 3/4			4 3/8	9 1/8	385 Lbs.
D180L48	"	mm	356	340	685	180	140	140	15	371	400	48	k6	110	231	190 kg.
E180L48		Inches	14	13 3/8	27	7.09	5 1/2	5 1/2	1 9/32	14 5/8	15 3/4			4 3/8	9 1/8	420 Lbs.
D200L55	"	mm	400	368	760	200	159	153	19	396	440	55	m6	110	243	255 kg.
E200L55		Inches	15 3/4	14 1/2	30	7.87	6 1/4	6	3/4	15 1/2	17 1/2			4 3/8	9 1/2	560 Lbs.
D225S55	2	mm	457	370	810	225	178	143	19	402	490	55	m6	110	259	290 kg.
E225S55		Inches	18	14 1/2	32	8.86	7	5 5/8	3/4	15 5/8	19 1/4			4 3/8	10 1/4	640 Lbs
D225M60	4 to 8	mm	457	395	835	225	178	156	19	445	490	60	m6	140	289	350 kg
E225M60		Inches	18	15 1/2	33	8.86	7	6 1/8	3/4	17 1/2	19 1/4			5 1/2	11 3/8	770 Lbs.
D250M60	2	mm	508	426	925	250	203	175	24	483	550	60	m6	140	308	440 kg.
E250M60		Inches	20	16 3/4	36 1/2	9.84	8	6 7/8	1 5/16	19	21 5/8			5 1/2	12 1/8	970 Lbs.
D250M65	4 to 8	mm	508	426	925	250	203	175	24	483	550	65	m6	140	308	440 kg.
E250M65		Inches	20	16 3/4	36 1/2	9.84	8	6 7/8	1 5/16	19	21 5/8			5 1/2	12 1/8	970 Lbs.
D280S65	2	mm	570	470	1000	280	229	184	24	514	630	65	m6	140	330	615 kg
E280S65		Inches	22 1/2	18 1/2	39 1/2	11.02	9	7 1/4	1 5/16	20 1/4	24 3/4			5 1/2	13	1355 Lbs.
D280S75	4 to 8	mm	570	470	1000	280	229	184	24	514	630	75	m6	140	330	615 kg.
E280S75		Inches	22 1/2	18 1/2	39 1/2	11.02	9	7 1/4	1 5/16	20 1/4	24 3/4			5 1/2	13	1355 Lbs.
D280M65	2	mm	570	520	1060	280	229	210	24	540	630	65	m6	140	330	675 kg.
E280M65		Inches	22 1/2	20 1/2	42	11.02	9	8 1/4	1 5/16	21 1/4	24 3/4			5 1/2	13	1500 Lbs.
D280M75	4 to 8	mm	570	520	1060	280	229	210	24	540	630	75	m6	140	330	675 kg.
E280M75		Inches	22 1/2	20 1/2	42	11.02	9	8 1/4	1 5/16	21 1/4	24 3/4			5 1/2	13	1500 Lbs.
D315S65	2	mm	635	520	1140	315	254	203	28	559	725	65	m6	140	356	800 kg.
E315S65		Inches	25	20 1/2	45	12.41	10	8	1 3/32	22	28 1/2			5 1/2	14	1760 Lbs.
D315S80	4 to 8	mm	635	520	1140	315	254	203	28	589	725	80	m6	170	386	800 kg.
E315S80		Inches	25	20 1/2	45	12.41	10	8	1 3/32	23 3/4	28 1/2			6 11/16	15 1/4	1760 Lbs
D315S80	2	mm	635	570	1190	315	254	229	28	585	725	65	m6	140	356	900 kg.
E315S80		Inches	25	22 1/2	47	12.41	10	9	1 3/32	23	28 1/2			5 1/2	14	1985 Lbs.
D315M80	4 to 8	mm	635	570	1190	315	254	229	28	615	725	80	m6	170	386	900 kg.
E315M80		Inches	25	22 1/2	47	12.41	10	9	1 3/32	24 1/4	28 1/2			6 11/16	15 1/4	1985 Lbs.

I.E.C. Motor Frames (cont'd)

IP23 ENCLOSED VENTILATED
(Similar to NEMA Open Drip Proof)



DIMENSIONS																
I.E.C. Frames	Poles	Units	A Max.	B Max.	C Approx.	D	E	F	H	M & N	O Approx.	U Nominal Tolerance		N & W	AC	Weight Approx.
C160M48	All	mm	318	267	700	160	127	105	15	323	330	48	k6	110	218	120 kg
		inches	12½	10½	27½	6.30	5	4⅞	19/32	12¾	13	1.8898		4⅞	8⅞	265 Lbs.
C160L48	All	mm	318	311	750	160	127	127	15	345	330	48	k6	110	218	150 kg
		inches	12½	12¼	29½	6.30	5	5	19/32	13⅝	13	1.8898		4⅞	8⅞	330 Lbs.
C180M55	All	mm	356	300	770	180	140	121	15	352	370	55	m6	110	231	200 kg
		inches	14	11¾	30¼	7.09	5½	4¾	19/32	13⅞	14½	2.1654		4⅞	9⅞	440 Lbs.
C180L55	All	mm	356	340	810	180	140	140	15	371	370	55	m6	110	231	210 kg
		inches	14	13⅜	31⅞	7.09	5½	5½	19/32	14⅝	14½	2.1654		4⅞	9⅞	465 Lbs.
C200M60	All	mm	400	326	870	200	159	133	19	406	410	60	m6	140	273	270 kg
		inches	15¾	12⅞	34¼	7.87	6¼	5¼	¾	16	16	2.3622		5½	10¾	595 Lbs.
C200L60	All	mm	400	368	900	200	159	152	19	425	410	60	m6	140	273	285 kg
		inches	15¾	14½	35½	7.87	6¼	6	¾	16¾	16	2.3622		5½	10¾	630 Lbs.
C225M60	2	mm	457	395	970	225	178	156	19	445	490	60	m6	140	289	350 kg
		inches	18	15½	38	8.86	7	6⅞	¾	17½	19¼	2.3622		5½	11⅝	770 Lbs.
C225M65	4 to 8	mm	457	395	970	225	178	156	19	445	490	65	m6	140	289	350 kg
		inches	18	15½	38	8.86	7	6⅞	¾	17½	19¼	2.5591		5½	11⅝	770 Lbs.
C250S65	2	mm	508	388	1100	250	203	154	24	464	550	65	m6	140	308	450 kg
		inches	20	15¼	43¼	9.84	8	6⅞	15/16	18¼	21⅝	2.5591		5½	12⅞	990 Lbs.
C250S75	4 to 8	mm	508	388	1100	250	203	154	24	464	550	75	m6	140	308	450 kg
		inches	20	15¼	43¼	9.84	8	6⅞	15/16	18¼	21⅝	2.9528		5½	12⅞	990 Lbs.
C250M65	2	mm	508	426	1140	250	203	175	24	483	550	65	m6	140	308	500 kg
		inches	20	16¾	44⅞	9.84	8	6⅞	15/16	19	21⅝	2.5591		5½	12⅞	1100 Lbs.
C250M75	4 to 8	mm	508	426	1140	250	203	175	24	483	550	75	m6	140	308	500 kg
		inches	20	16¾	44⅞	9.84	8	6⅞	15/16	19	21⅝	2.9528		5½	12⅞	1100 Lbs.
C280S65	2	mm	570	470	1265	280	229	184	24	514	630	65	m6	140	330	650 kg
		inches	22½	18½	49¾	11.02	9	7¼	15/16	20¼	24¾	2.5591		5½	13	1435 Lbs.
C280S80	4 to 8	mm	570	470	1265	280	229	184	24	544	630	80	m6	170	360	650 kg
		inches	22½	18½	49¾	11.02	9	7¼	15/16	21⅞	24¾	3.1496		6⅞	14⅞	1435 Lbs.
C280M65	2	mm	570	520	1315	280	229	210	24	540	630	65	m6	140	330	700 kg
		inches	22½	20½	51¾	11.02	9	8¼	15/16	21¼	24¾	2.5591		5½	13	1545 Lbs.
C280M80	4 to 8	mm	570	520	1315	280	229	210	24	570	630	80	m6	170	360	700 kg
		inches	22½	20½	51¾	11.02	9	8¼	15/16	22⅞	24¾	3.1496		6⅞	14⅞	1545 Lbs.
C315S70	2	mm	635	520	1475	315	254	203	28	559	725	70	m6	140	356	850 kg
		inches	25	20½	58	12.40	10	8	1⅜	22	28½	2.7559		5½	14	1875 Lbs.
C315S90	4 to 8	mm	635	520	1475	315	254	203	28	589	725	90	m6	170	386	850 kg
		inches	25	20½	58	12.40	10	8	1⅜	23¼	28½	3.5433		6⅞	15¼	1875 Lbs.
C315M7C	2	mm	635	570	1525	315	254	229	28	585	725	70	m6	140	356	950 kg
		inches	25	22½	60	12.40	10	9	1⅜	23	28½	2.7559		5½	14	2100 Lbs.
C315M90	4 to 8	mm	635	570	1525	315	254	229	28	615	725	90	m6	170	386	950 kg
		inches	25	22½	60	12.40	10	9	1⅜	24¼	28½	3.5433		6⅞	15¼	2100 Lbs.

TECH-G-10 TEFC IP55 Metric IEC Motors (Conversion NEMA to Metric)

HP	kW	RPM	FRAME	NEMA Equivalent Frame
1	.75	3000	80	56
1	.75	1500	80	56
1	.75	1000	90S	143T
1.5	1.1	3000	80	56
1.5	1.1	1500	90S	143T
1.5	1.1	1000	90L	145T
2	1.5	3000	90S	143T
2	1.5	1500	90L	145T
2	1.5	1000	100L	182T
3	2.2	3000	90L	145T
3	2.2	1500	100L	182T
3	2.2	1000	112M	184T
4	3.0	3000	100L	182T
4	3.0	1500	100L	182T
4	3.0	1000	132S	213T
5.5	4.0	3000	112M	184T
5.5	4.0	1500	112M	184T
5.5	4.0	1000	132M	215T
7.5	5.5	3000	132S	213T
7.5	5.5	1500	132S	213T
7.5	5.5	1000	132M	215T
10	7.5	3000	132S	213T
10	7.5	1500	132M	215T
10	7.5	1000	160M	254T
15	11	3000	160M	254T
15	11	1500	160M	254T
15	11	1000	160L	256T
20	15	3000	160M	254T
20	15	1500	160L	256T
20	15	1000	180L	284T
25	18.5	3000	160L	256T
25	18.5	1500	180M	284T
25	18.5	1000	200L	326T
30	22	3000	180M	284T
30	22	1500	180L	286T
30	22	1000	200L	326T
40	30	3000	200L	326T
40	30	1500	200L	326T
40	30	1000	225M	365T
50	37	3000	200L	326T
50	37	1500	225S	364T
50	37	1000	250S	404T
60	45	3000	225M	354T
60	45	1500	225M	365T
60	45	1000	250M	405T
75	55	3000	250S	404T
75	55	1500	250S	404T
75	55	1000	280S	444T
100	75	3000	250M	405T
100	75	1500	250M	405T
100	75	1000	280M	445T
125	90	3000	280S	444T
125	90	1500	280S	444T
125	90	1000	315S	504Z
150	110	3000	280M	445T
150	110	1500	280M	445T
150	110	1000	315M	505Z



Section TECH-H

Conversion Factors

TECH-H-1 Temperature Conversion Chart {Centigrade (Celsius)-Fahrenheit}

C	F	C	F	C	F	C	F	C	F	C	F
-40	-40.0	+5	+41.0	+40	+104.0	+175	+347	+350	+662	+750	+1382
-38	-36.4	6	42.8	41	105.8	180	356	355	671	800	1472
-36	-32.8	7	44.6	42	107.6	185	365	360	680	850	1562
-34	-29.2	8	46.4	43	109.4	190	374	365	689	900	1652
-32	-25.6	9	48.2	44	111.2	195	383	370	698	950	1742
-30	-22.0	10	50.0	45	113.0	200	392	375	707	1000	1832
-28	-18.4	11	51.8	46	114.8	205	401	380	716	1050	1922
-26	-14.8	12	53.6	47	116.6	210	410	385	725	1100	2012
-24	-11.2	13	55.4	48	118.4	215	419	390	734	1150	2102
-22	-7.6	14	57.2	49	120.2	220	428	395	743	1200	2192
-20	-4.0	15	59.0	50	122.0	225	437	400	752	1250	2282
-19	-2.2	16	60.8	55	131.0	230	446	405	761	1300	2372
-18	-0.4	17	62.6	60	140.0	235	455	410	770	1350	2462
-17	+1.4	18	64.4	65	149.0	240	464	415	779	1400	2552
-16	3.2	19	66.2	70	158.0	245	473	420	788	1450	2642
-15	5.0	20	68.0	75	167.0	250	482	425	797	1500	2732
-14	6.8	21	69.8	80	176.0	255	491	430	806	1550	2822
-13	8.6	22	71.6	85	185.0	260	500	435	815	1600	2912
-12	10.4	23	73.4	90	194.0	265	509	440	824	1650	3002
-11	12.2	24	75.2	95	203.0	270	518	445	833	1700	3092
-10	14.0	25	77.0	100	212.0	275	527	450	842	1750	3182
-9	15.8	26	78.8	105	221.0	280	536	455	851	1800	3272
-8	17.6	27	80.6	110	230.0	285	545	460	860	1850	3362
-7	19.4	28	82.4	115	239.0	290	554	465	869	1900	3452
-6	21.2	29	84.2	120	248.0	295	563	470	878	1950	3542
-5	23.0	30	86.0	125	257.0	300	572	475	887	2000	3632
-4	24.8	31	87.8	130	266.0	305	581	480	896	2050	3722
-3	26.6	32	89.6	135	275.0	310	590	485	905	2100	3812
-2	28.4	33	91.4	140	284.0	315	599	490	914	2150	3902
-1	30.2	34	93.2	145	293.0	320	608	495	923	2200	3992
0	32.0	35	95.0	150	302.0	325	617	500	932	2250	4082
+1	33.8	36	96.8	155	311.0	330	626	550	1022	2300	4172
2	35.6	37	98.6	160	320.0	335	635	600	1112	2350	4262
3	47.4	38	100.4	165	329.0	340	644	650	1202	2400	4352
4	39.2	39	102.2	170	338.0	345	653	700	1292	2450	4442

$$\text{Degrees Celsius} = \frac{(\text{Degrees Fahrenheit} - 32) \times 5}{9}$$

$$\text{Degrees Fahrenheit} = \frac{(\text{Degrees Celsius} \times 9) + 32}{5}$$

$$\text{Degrees Kelvin (K)} = \text{Degrees Celsius} + 273.15$$

$$\text{Degrees Rankine (R)} = \text{Degrees Fahrenheit} + 459.69$$

(0 degrees K or R = absolute zero)

TECH-H-2 A.P.I. and Baumé Gravity Tables and Weight Factors

A.P.I Gravity	Baume Gravity	Specific Gravity	Lbs. Per U.S. Gal.	U.S. Gals. per Lb.
0	10.247	1.0760	8.962	0.1116
1	9.223	1.0679	8.895	0.1124
2	8.198	1.0599	8.828	0.1133
3	7.173	1.0520	8.762	0.1141
4	6.148	1.0443	8.698	0.1150
5	5.124	1.0366	8.634	0.1158
6	4.099	1.0291	8.571	0.1167
7	3.074	1.0217	8.509	0.1175
8	2.049	1.0143	8.448	0.1184
9	1.025	1.0071	8.388	0.1192
10	10.00	1.0000	8.328	0.1201
11	10.99	0.9930	8.270	0.1209
12	11.98	0.9861	8.212	0.1218
13	12.97	0.9792	8.155	0.1226
14	13.96	0.9725	8.099	0.1235
15	14.95	0.9659	8.044	0.1243
16	15.94	0.9593	7.989	0.1252
17	16.93	0.9529	7.935	0.1260
18	17.92	0.9465	7.882	0.1269
19	18.90	0.9402	7.830	0.1277
20	19.89	0.9340	7.778	0.1286
21	20.88	0.9279	7.727	0.1294
22	21.87	0.9218	7.676	0.1303
23	22.86	0.9159	7.627	0.1311
24	23.85	0.9100	7.578	0.1320
25	24.84	0.9024	7.529	0.1328
26	25.83	0.8984	7.481	0.1337
27	26.82	0.8927	7.434	0.1345
28	27.81	0.8871	7.387	0.1354
29	28.80	0.8816	7.341	0.1362
30	29.79	0.8762	7.296	0.1371
31	30.78	0.8708	7.251	0.1379
32	31.77	0.8654	7.206	0.1388
33	32.76	0.8602	7.163	0.1396
34	33.75	0.8550	7.119	0.1405
35	34.73	0.8498	7.076	0.1413
36	35.72	0.8448	7.034	0.1422
37	36.71	0.8398	6.993	0.1430
38	37.70	0.8348	6.951	0.1439
39	38.69	0.8299	6.910	0.1447
40	39.68	0.8251	6.870	0.1456
41	40.67	0.8203	6.830	0.1464
42	41.66	0.8155	6.790	0.1473
43	42.65	0.8109	6.752	0.1481
44	43.64	0.8063	6.713	0.1490
45	44.63	0.8017	6.675	0.1498
46	45.62	0.7972	6.637	0.1507
47	50.61	0.7927	6.600	0.1515
48	50.60	0.7883	6.563	0.1524
49	50.59	0.7839	6.526	0.1532
50	50.58	0.7796	6.490	0.1541

A.P.I Gravity	Baume Gravity	Specific Gravity	Lbs. Per U.S. Gal.	U.S. Gals. per Lb.
51	50.57	0.7753	6.455	0.1549
52	51.55	0.7711	6.420	0.1558
53	52.54	0.7669	6.385	0.1566
54	53.53	0.7628	6.350	0.1575
55	54.52	0.7587	6.316	0.1583
56	55.51	0.7547	6.283	0.1592
57	56.50	0.7507	6.249	0.1600
58	57.49	0.7467	6.216	0.1609
59	58.48	0.7428	6.184	0.1617
60	59.47	0.7389	6.151	0.1626
61	60.46	0.7351	6.119	0.1634
62	61.45	0.7313	6.087	0.1643
63	62.44	0.7275	6.056	0.1651
64	63.43	0.7238	6.025	0.1660
65	64.42	0.7201	5.994	0.1668
66	65.41	0.7165	5.964	0.1677
67	66.40	0.7128	5.934	0.1685
68	67.39	0.7093	5.904	0.1694
69	68.37	0.7057	5.874	0.1702
70	69.36	0.7022	5.845	0.1711
71	70.35	0.6988	5.817	0.1719
72	71.34	0.6953	5.788	0.1728
73	72.33	0.6919	5.759	0.1736
74	73.32	0.6886	5.731	0.1745
75	74.31	0.6852	5.703	0.1753
76	75.30	0.6819	5.676	0.1762
77	76.29	0.6787	5.649	0.1770
78	77.28	0.6754	5.622	0.1779
79	78.27	0.6722	5.595	0.1787
80	79.26	0.6690	5.568	0.1796
81	80.25	0.6659	5.542	0.1804
82	81.24	0.6628	5.516	0.1813
83	82.23	0.6597	5.491	0.1821
84	83.22	0.6566	5.465	0.1830
85	84.20	0.6536	5.440	0.1838
86	85.19	0.6506	5.415	0.1847
87	86.18	0.6476	5.390	0.1855
88	87.17	0.6446	5.365	0.1864
89	88.16	0.6417	5.341	0.1872
90	89.15	0.6388	5.316	0.1881
91	90.14	0.6360	5.293	0.1889
92	91.13	0.6331	5.269	0.1898
93	92.12	0.6303	5.246	0.1906
94	93.11	0.6275	5.222	0.1915
95	94.10	0.6247	5.199	0.1924
96	95.09	0.6220	5.176	0.1932
97	96.08	0.6193	5.154	0.1940
98	97.07	0.6166	5.131	0.1949
99	98.06	0.6139	5.109	0.1957
100	99.05	0.6112	5.086	0.1966

The relation of Degrees Baume or A.P.I. to Specific Gravity is expressed by the following formulas:

For liquids lighter than water:

$$\text{Degrees Baume} = \frac{140}{G} - 130, \quad G = \frac{140}{130 + \text{Degrees Baume}}$$

$$\text{Degrees A.P.I.} = \frac{141.5}{G} - 131.5, \quad G = \frac{141.5}{131.5 + \text{Degrees A.P.I.}}$$

For liquids heavier than water:

$$\text{Degrees Baume} = 145 - \frac{145}{G}, \quad G = \frac{145}{145 - \text{Degrees Baume}}$$

G = Specific Gravity = ratio of the weight of a given volume of oil at 60° Fahrenheit to the weight of the same volume of water at 60° Fahrenheit.

The above tables are based on the weight of 1 gallon (U.S.) of oil with a volume of 231 cubic inches at 60 degrees Fahrenheit in air at 760 m.m. pressure and 50% humidity. Assumed weight of 1 gallon of water at 60° Fahrenheit in air is 8.32828 pounds.

To determine the resulting gravity by missing oils of different gravities:

$$D = \frac{md_1 - nd_2}{m + n}$$

D = Density or Specific Gravity of mixture

m = Proportion of oil of d₁ density

n = Proportion of oil of d₂ density

d₁ = Specific Gravity of m oil

d₂ = Specific Gravity of n oil

TECH-H-3 Approximate Conversion Table for Hardness Numbers Obtained by Different Methods*

Brinell Number 10 mm. Ball 3000 Kg. Load	Rockwell Number		Shore Scieroscope Number	Vickers Pyramid Number
	C-Scale	B-Scale		
682	61.7		84	737
653	60		81	697
633	59		79	674
614	58		78	654
596	57		77	636
578	56		75	615
560	55		73	596
543	54		72	578
527	53		71	561
500	52		69	544
475	50		67	513
451	48		64	484
432	46		62	458
409	44		58	434
390	42		56	412
371	40		54	392
353	38		51	372
336	36		49	354
319	34		47	336
301	32		44	318
286	30		42	302
271	28		41	286
258	26		38	272
247	24		37	260
237	22	99	35	248
226	20	98	34	238
212	16	95	32	222
194	12	92	29	204
179	8	89	27	188
158	2	83	24	166
141		77	21	141
125		70	18	125
110		62		110
99		55		99
89		47		89

*Compiled from various manufacturers' Tables.

TECH-H-4 Conversion Factors

English measures - unless otherwise designated, are those used in the United States, and the units of weight and mass are avoirdupois units.

Gallon - designates the U.S. gallon. To convert into the Imperial gallon, multiply the U.S. gallon by 0.83267. Likewise, the word ton designates a short ton, 2,000 pounds.

Properties of water- it freezes at 32°F., and is at its maximum density at 39.2° F. In the multipliers using the properties of water, calculations are based on water at 39.2° F. in a vacuum, weighing 62.427 pounds per cubic foot, or 8.345 pounds per U.S. gallon.

Multiply	By	To Obtain
Acres	43,560	Square feet
Acres	4047	Square meters
Acres	1.562 x 10 ⁻³	Square miles
Acres	4840	Square yards
Acre-feet	43,560	Cubic feet
Acre-feet	325,851	Gallons
Acre-feet	1233.48	Cubic Meters
Atmospheres	1.0332	Atmospheres (metric)
Atmospheres	1.01325	Bars
Atmospheres	76.0	Cms. Of mercury
Atmospheres	29.92	Inches of mercury

Multiply	By	To Obtain
Atmospheres	33.90	Feet of water
Atmospheres	10,332	kgs/sq. ft
Atmospheres	14.70	Lbs./ sq. inch
Atmospheres	1.058	Tons/sq. ft.
Atmospheres (metric)	0.9678	Atmospheres
Atmospheres (metric)	980,665.	Bars
Bars	.98692	Atmospheres
Bars	33.456	Feet H ₂ O @39°F.
Bars	29.530	In. Hg @ 32° F.
Bars	1.0197	kg/cm ²
Bars	2088.6	Pounds/ ft. ²

Multiply	By	To Obtain
Bars	14.504	Pounds/in. ²
Barrels- oil	42	Gallons- oil
Barrels- beer	31	Gallons- beer
Barrels- whiskey	45	Gallons- whiskey
Barrels/day- oil	0.02917	Gallons/min.- oil
Bags or sacks-cement	94	Pounds/cement
Board feet	144 sq. in. x 1 in.	Cubic inches
British Thermal Units	0.2520	Kilogram- calories
British Thermal Units	777.6	Foot- lbs.
British Thermal Units	3.927 x 10 ⁴	Horsepower- hrs.
British Thermal Units	107.5	Kilogram- meters
British Thermal Units	2.928 x 10 ⁴	Kilowatt- hrs.
B.T.U./min.	12.96	Foot-lbs./sec.
B.T.U./min.	0.02356	Horsepower
B.T.U./min.	0.01757	Kilowatts
B.T.U./min.	17.57	Watts
Centares (Centiares)	1	Square meters
Centigrams	0.01	Grams
Centiliters	0.01	Liters
Centimeters	0.3937	Inches
Centimeters	0.01	Meters
Centimeters	10	Millimeters
Centimeters of mercury	0.01316	Atmosphere
Centimeters of mercury	0.013332	Bars
Centimeters of mercury	0.013595	kg/cm ²
Centimeters of mercury	0.4461	Feet of water
Centimeters of mercury	136.0	kgs/sq. meter
Centimeters of mercury	27.85	Lbs./sq. ft.
Centimeters of mercury	0.1934	Lbs./sq. inch
Centimeters/sec.	1.969	Feet/min.
Centimeters/sec.	0.03281	Feet/sec.
Centimeters/sec.	0.036	Kilometers/hr.
Centimeters/sec.	0.6	Meters/min.
Centimeters/sec.	0.02237	Miles/hr.
Centimeters/sec.	3.728 x 10 ⁻⁴	Miles/min.
Cms./sec./sec.	0.03281	Feet/sec./sec.
Centipoises	0.001	Pascal-second
Centipoises	0.01	Poises
Centistokes	0.01	Sq. cm/sec.
Centistokes	0.01	Stokes
Cubic centimeters	3.531 x 10 ⁻⁵	Cubic feet
Cubic centimeters	6.102 x 10 ⁻²	Cubic inches
Cubic centimeters	10 ⁻⁶	Cubic meters
Cubic centimeters	1.308 x 10 ⁻⁶	Cubic yards
Cubic centimeters	2.642 x 10 ⁻⁴	Gallons
Cubic centimeters	9.999 x 10 ⁻⁴	Liters
Cubic centimeters	2.113 x 10 ⁻³	Pints (liq.)
Cubic centimeters	1.057 x 10 ⁻³	Quarts (liq.)
Cubic cm/sec.	0.0158502	Gallons/minute
Cubic cm/sec.	0.001	Liters/sec.
Cubic feet	0.1781	Barrels (42 US Gal.)
Cubic feet	2.832 x 10 ⁻⁴	Cubic cms.
Cubic feet	1728	Cubic inches
Cubic feet	0.02832	Cubic meters
Cubic feet	0.03704	Cubic yards
Cubic feet	7.48052	Gallons
Cubic feet	28.32	Liters
Cubic feet	59.84	Pints (liq.)
Cubic feet	29.92	Quarts (liq.)
Cubic feet/min.	472.0	Cubic cms./sec.

Multiply	By	To Obtain
Cubic feet/min.	0.1247	Gallons/sec.
Cubic feet/min.	0.4719	Liters/sec.
Cubic feet/min.	62.43	Pounds of water/min.
Cubic feet/sec.	0.646317	Millions gals./day
Cubic feet/sec.	448.831	Gallons/min.
Cubic inches	16.39	Cubic centimeters
Cubic inches	5.787 x 10 ⁻⁴	Cubic feet
Cubic inches	1.639 x 10 ⁻⁵	Cubic meters
Cubic inches	2.143 x 10 ⁻⁵	Cubic yards
Cubic inches	4.329 x 10 ⁻³	Gallons
Cubic inches	1.639 x 10 ⁻²	Liters
Cubic inches	0.03463	Pints (liq.)
Cubic inches	0.01732	Quarts (liq.)
Cubic meters	10 ⁶	Cubic centimeters
Cubic meters	35.31	Cubic feet
Cubic meters	61023	Cubic inches
Cubic meters	1.308	Cubic yards
Cubic meters	264.2	Gallons
Cubic meters	999.97	Liters
Cubic meters	2113	Pints (liq.)
Cubic meters	1057	Quarts (liq.)
Cubic meters/hr.	4.40	Gallons/min.
Cubic yards	4.8089	Barrels (42 U.S. Gal.)
Cubic yards	764,554.86	Cubic centimeters
Cubic yards	27	Cubic feet
Cubic yards	46, 656	Cubic inches
Cubic yards	0.7646	Cubic meters
Cubic yards	202.0	Gallons
Cubic yards	764.5	Liters
Cubic yards	1616	Pints (liq.)
Cubic yards	807.9	Quarts (liq.)
Cubic yards/min.	0.45	Cubic feet/sec.
Cubic yards/min.	202.0	Gallons/min.
Cubic yards/min.	3.366	Gallons/sec.
Cubic yards/min.	12.74	Liters/sec.
Decigrams	0.1	Grams
Deciliters	0.1	Liters
Decimeters	0.1	Meters
Degrees (angle)	60	Minutes
Degrees (angle)	0.01745	Radians
Degrees (angle)	3600	Seconds
Degrees/sec.	0.01745	Radians/sec.
Degrees/sec.	0.1667	Revolutions/min.
Degrees/sec.	0.002778	Revolutions/sec.
Dekagrams	10	Grams
Dekaliters	10	Liters
Dekameters	10	Meters
Drams	27.34375	Grains
Drams	0.0625	Ounces
Drams	1.771845	Grams
Fathoms	6	Feet
Feet	30.48	Centimeters
Feet	0.166667	Fathoms
Feet	3.0480 x 10 ⁻⁴	Kilometers
Feet	304.80	Millimeters
Feet	12	Inches
Feet	0.3048	Meters
Feet	1/3	Yards
Feet of water	0.0295	Atmospheres
Feet of water	0.8826	Inches of mercury

Multiply	By	To Obtain
Feet of water	304.8	kgs./sq. meter
Feet of water	62.43	Lbs./sq. ft.
Feet of water	0.4335	Lbs./sq. inch
Feet/min.	0.5080	Centimeters/sec.
Feet/min.	0.01667	Feet/sec.
Feet/min.	0.01829	Kilometers/hr.
Feet/min.	0.3048	Meters/min.
Feet/min.	0.01136	Miles/hr.
Feet/sec.	30.48	Centimeters/sec.
Feet/sec.	1.09726	Kilometers/hr.
Feet/sec.	0.5924	Knots
Feet/sec.	18.29	Meters/min.
Feet/sec.	0.6818	Miles/hr.
Feet/sec.	0.01136	Miles/min.
Feet/sec./sec.	30.48	Cms./sec./sec.
Feet/sec./sec.	0.3048	Meters/sec./sec.
Feet/sec./sec.	0.0310810	g's (gravity)
Foot- pounds	1.286×10^{-3}	British Thermal Units
Foot- pounds	5.050×10^{-7}	Horsepower-hrs.
Foot- pounds	3.240×10^{-4}	Kilogram- calories
Foot- pounds	0.1383	Kilogram- meters
Foot- pounds	3.766×10^{-7}	Kilowatt- hours
Foot- pounds/min.	2.140×10^{-5}	B.T.U./sec.
Foot- pounds/min.	0.01667	Foot-pounds/sec.
Foot- pounds/min.	3.030×10^{-5}	Horsepower
Foot- pounds/min.	5.393×10^{-3}	Gm.-calories/sec.
Foot- pounds/min.	2.280×10^{-5}	Kilowatts
Foot- pounds/sec.	7.704×10^{-2}	B.T.U./min.
Foot- pounds/sec.	1.818×10^{-3}	Horsepower
Foot- pounds/sec.	1.941×10^{-2}	kg.-calories/min.
Foot- pounds/sec.	1.356×10^{-3}	Kilowatts
G's (Accel. due to grav.)	32.174	Feet/sec. ²
G's (Accel. due to grav.)	35.3034	Km/hr.-sec.
G's (Accel. due to grav.)	9.80665	Meters/sec. ²
G's (Accel. due to grav.)	21.9371	Miles/hr.-sec.
Gallons	3785	Cubic centimeters
Gallons	0.1337	Cubic feet
Gallons	231	Cubic inches
Gallons	3.785×10^{-3}	Cubic meters
Gallons	4.951×10^{-3}	Cubic yards
Gallons	3.785	Liters
Gallons	8	Pints (liq.)
Gallons	4	Quarts (liq.)
Gallons-Imperial	1.20095	US Gallons
Gallons- US	0.83267	Imperial Gallons
Gallons water	8.345	Pounds of water
Gallons per day	9.284×10^{-5}	Cubic ft./min.
Gallons per day	1.5472×10^{-6}	Cubic ft./sec.
Gallons per day	2.6289×10^{-6}	Cubic meters/min.
Gallons per day	0.09284	Liters/min.
Gallons per hour	0.1337	Cubic ft./hr.
Gallons per hour	0.002228	Cubic ft./min.
Gallons per hour	3.71×10^{-5}	Cubic ft./sec.
Gallons per hour	6.309×10^{-5}	Cubic meters/min.
Gallons per hour	.016667	Gallons/min.
Gallons per hour	2.7778×10^{-4}	Gallons/sec.
Gallons per hour	0.06309	Liters/min.
Gallons/min.	34.286	Barrels (42 US Gal.)/day

Multiply	By	To Obtain
Gallons/min.	1.4286	Barrels (42 US Gal.)/hr.
Gallons/min.	0.02381	Barrels (42 USGal.)/min.
Gallons/min.	1440	Gallons/day
Gallons/min.	2.228×10^{-3}	Cubic feet/sec.
Gallons/min.	0.06308	Liters/sec.
Gallons/min.	8.0208	Cu. ft./hr.
Gallons/sec.	60	Gallons/min.
Gallons/sec.	227.12	Liters/min.
Grains (troy)	0.06480	Grams
Grains (troy)	0.04167	Pennyweights (troy)
Grains (troy)	2.0833×10^{-3}	Ounces
Grains/US gal.	17.118	Parts/million
Grains/US gal.	142.86	Lbs./million gal.
Grains/Imp. gal.	14.254	Parts/million
Grams	980.7	Dynes
Grams	15.43	Grains
Grams	.001	Kilograms
Grams	1000	Milligrams
Grams	0.03527	Ounces
Grams	0.03215	Ounces (troy)
Grams	2.205×10^{-3}	Pounds
Grams/cm.	5.600×10^{-3}	Pounds/ inch
Grams/cu. cm.	62.43	Pounds/cubic foot
Grams/cu. cm.	0.03613	Pounds/cubic inch
Grams/liter	58.416	Grains/gal.
Grams/liter	8.345	Pounds/1000 gals.
Grams/liter	0.06242	Pounds/cubic foot
Grams/liter	1000	Parts/million
Hectares	2.471	Acres
Hectares	1.076×10^5	Square feet
Hectograms	100	Grams
Hectoliters	100	Liters
Hectometers	100	Meters
Hectowatts	100	Watts
Horsepower	42.44	B.T.U./min.
Horsepower	33,000	Foot-lbs./min.
Horsepower	550	Foot-lbs./sec.
Horsepower	1.014	Horsepower (metric)
Horsepower	10.547	kg.-calories/min.
Horsepower	0.7457	Kilowatts
Horsepower	745.7	Watts
Horsepower (boiler)	33, 493	B.T.U./hr.
Horsepower (boiler)	9.809	Kilowatts
Horsepower (boiler)	9.2994	B.T.U./sec.
Horsepower (boiler)	9809.5	Watts
Horsepower-hours	2546	B.T.U.
Horsepower-hours	1.98×10^6	Foot-lbs.
Horsepower-hours	641.6	Kilogram-calories
Horsepower-hours	2.737×10^5	Kilogram-meters
Horsepower-hours	0.7457	Kilowatt-hours
Inches	2.540	Centimeters
Inches	0.083333	Feet
Inches	0.0254	Meters
Inches	25.4	Millimeters
Inches	0.0277778	Yards
Inches of mercury	0.03342	Atmospheres
Inches of mercury	0.03386	Bars
Inches of mercury	13.6	Inches H ₂ O

Multiply	By	To Obtain
Inches of mercury	0.034531	kg/cm ²
Inches of mercury	3374.1	Pascals
Inches of mercury	70.727	Pounds/ft. ²
Inches of mercury	0.49116	Pounds/in. ²
Inches of mercury	1.133	Feet of water
Inches of mercury	345.3	kgs./sq. meter
Inches of mercury	70.73	Lbs./sq. ft.
Inches of mercury (32° F)	0.491	Lbs./sq. inch
Inches of water	0.002458	Atmospheres
Inches of water	0.07355	Inches of mercury
Inches of water	25.40	kgs./sq. meter
Inches of water	0.578	Ounces/sq. inch
Inches of water	5.202	Lbs./sq. foot
Inches of water	0.03613	Lbs./sq. inch
Joules	9.479 x 10 ⁻⁴	B.T.U
Joules	0.239006	Calories (Thermo)
Joules	0.73756	Foot-lb.f.
Joules	3.725 x 10 ⁻⁷	HP-hr. (US)
Joules	2.7778 x 10 ⁻⁷	Kilowatt-hr.
Joules	1	Newton-m
Joules	2.7778 x 10 ⁻⁴	Watt-hr.
Kilograms	35.274	Ounces (avoir)
Kilograms	32.151	Ounces (troy)
Kilograms	980,665	Dynes
Kilograms	2.205	Lbs.
Kilograms	1.102 x 10 ⁻³	Tons (short)
Kilograms	34.286	Tons (assay)
Kilograms	9.8421 x 10 ⁻⁴	Tons (long)
Kilograms	0.001	Tons (metric)
Kilograms	10 ³	Grams
Kilograms-cal./sec.	3.968	B.T.U./sec.
Kilograms-cal./sec	3086	Foot-lbs./sec.
Kilograms-cal./sec	5.6145	Horsepower
Kilograms-cal./sec	4186.7	Watts
Kilograms/cm	0.96783	Atmospheres
Kilograms/cm	0.980665	Bars
Kilograms/cm	28.959	Inches Hg@ 32° F
Kilograms/cm	14.223	Pounds/in. ²
Kilograms-cal./min.	3085.9	Foot-lbs./min.
Kilograms-cal./min	0.09351	Horsepower
Kilograms-cal./min	69.733	Watts
kgs/meter	0.6720	Lbs./foot
kgs/sq. meter	9.678 x 10 ⁻⁵	Atmospheres
kgs/sq. meter	3.281 x 10 ⁻³	Feet of water
kgs/sq. meter	2.896 x 10 ⁻³	Inches of mercury
kgs/sq. meter	0.2048	Lbs./sq. foot
kgs/sq. meter	1.422 x 10 ⁻³	Lbs./sq. inch
kgs/sq. millimeter	10 ⁶	kgs./sq. meter
Kiloliters	10 ³	Liters
Kilometers	10 ⁵	Centimeters
Kilometers	3281	Feet
Kilometers	10 ³	Meters
Kilometers	0.6214	Miles
Kilometers	1094	Yards
Kilopascal	.145	Pounds/in. ²
Kilometers/hr.	27.78	Centimeters/sec.
Kilometers/hr.	54.68	Feet/min.
Kilometers/hr.	0.9113	Feet/sec.
Kilometers/hr.	.5399	Knots
Kilometers/hr.	16.67	Meters/min.

Multiply	By	To Obtain
Kilometers/hr.	0.6214	Miles/hr.
Kms./hr./sec.	27.78	Cms./sec./sec.
Kms./hr./sec.	0.9113	Ft./sec./sec.
Kms./hr./sec.	0.2778	Meters/sec./sec.
Kilowatts	56.907	B.T.U./min.
Kilowatts	4.425 x 10 ⁴	Foot-lbs./min.
Kilowatts	737.6	Foot-lbs./sec.
Kilowatts	1.341	Horsepower (US)
Kilowatts	1.3597	Horsepower (metric)
Kilowatts	1000	Joules/sec.
Kilowatts	3412.9	B.T.U./hr.
Kilowatts	0.94827	B.T.U./sec.
Kilowatts	14.34	kg.-calories/min.
Kilowatts	10 ³	Watts
Kilowatt-hours	3414.4	B.T.U
Kilowatt-hours	2.655 x 10 ⁶	Foot-lbs.
Kilowatt-hours	1.341	Horsepower-hrs.
Kilowatt-hours	860.4	Kilogram-calories
Kilowatt-hours	3.671 x 10 ⁵	Kilogram-meters
Liters	10 ³	Cubic centimeters
Liters	0.03531	Cubic feet
Liters	61.02	Cubic inches
Liters	10 ⁻³	Cubic meters
Liters	1.308 x 10 ⁻³	Cubic yards
Liters	0.2642	Gallons
Liters	2.113	Pints (liq.)
Liters	1.057	Quarts (liq.)
Liters/min.	5.886 x 10 ⁻⁴	Cubic ft./sec.
Liters/min.	4.403 x 10 ⁻³	Gals./sec.
Lumber width (in) x Thickness (in)	Length (ft.)	Board feet
12		
Meters	100	Centimeters
Meters	3.281	Feet
Meters	39.37	Inches
Meters	10 ⁻³	Kilometers
Meters	10 ³	Millimeters
Meters	1.094	Yards
Meters/min.	1.667	Centimeters/sec.
Meters/min.	3.281	Feet/min.
Meters/min.	0.05468	Feet/sec.
Meters/min.	0.06	Kilometers/hr.
Meters/min.	0.03728	Miles/hr.
Meters/sec.	196.8	Feet/min.
Meters/sec.	3.281	Feet/sec.
Meters/sec.	3.6	Kilometers/hr.
Meters/sec.	0.06	Kilometers/min.
Meters/sec.	2.287	Miles/hr.
Meters/sec.	0.03728	Miles/min.
Meters/sec. ²	3.2808	Feet/sec. ²
Meters/sec. ²	0.101972	G (gravity)
Meters/sec. ²	39.37	Inches/sec. ²
Meters/sec. ²	134.214	Miles/hr.-min.
Meter-kg. (force)	9.8067	Joules
Microns	10 ⁻⁶	Meters
Miles	1.609 x 10 ⁵	Centimeters
Miles	5280	Feet
Miles	1.609	Kilometers
Miles	1760	Yards
Miles/hr.	44.70	Centimeters/sec.

Multiply	By	To Obtain
Miles/hr.	88	Feet/min.
Miles/hr.	1.467	Feet/sec.
Miles/hr.	1.609	Kilometers/hr.
Miles/hr.	0.8689	Knots
Miles/hr.	26.82	Meter/min.
Miles/min.	2682	Meters/min.
Miles/min.	88	Feet/sec.
Miles/min.	1.609	Kilometers/min.
Miles/min.	60	Miles/hr.
Milliers	10 ³	Kilograms
Milligrams	10 ⁻³	Grams
Milliliters	10 ⁻³	Liters
Millimeters	0.1	Centimeters
Millimeters	0.03937	Inches
Milligrams/liter	1	Parts/million
Million Gals./day	1.54723	Cubic ft./sec.
Miner's inches	1.5	Cubic ft./min.
Minutes (angle)	2.909 x 10 ⁻⁴	Radians
Newtons (N)	.225	Pounds-force
Ounces	16	Drams
Ounces	437.5	Grains
Ounces	0.0625	Pounds
Ounces	28.3495	Grams
Ounces	0.9115	Ounces (troy)
Ounces	2.790 x 10 ⁻⁵	Tons (long)
Ounces	2.835 x 10 ⁻⁵	Tons (metric)
Ounces (troy)	480	Grains
Ounces (troy)	20	Pennyweights (troy)
Ounces (troy)	0.08333	Pounds (troy)
Ounces (troy)	31.10348	Grams
Ounces (troy)	1.09714	Ounces (avoir)
Ounces (fluid)	1.805	Cubic inches
Ounces (fluid)	0.02957	Liters
Ounces/sq. inch	0.0625	Lbs./sq. inch
Ounces/gal (US)	7.4892	kg/m ³
Ounces/gal (US)	0.25	Ounces/quart
Ounces/gal (US)	0.46753	Pounds/ft. ³
Ounces/gal (US)	2.7056 x 10 ⁻⁴	Pounds/in. ³
Parts/million	0.0584	Grains/US gal.
Parts/million	0.07015	Grains/Imp. gal.
Parts/million	8.345	Lbs./million gal.
Pennyweights (troy)	24	Grains
Pennyweights (troy)	1.55517	Grams
Pennyweights (troy)	0.05	Ounces (troy)
Pennyweights (troy)	4.1667 x 10 ⁻³	Pounds (troy)
Pounds	16	Ounces
Pounds	256	Drams
Pounds	7000	Grains
Pounds	0.0005	Tons (short)
Pounds	453.5924	Grams
Pounds	1.21528	Pounds (troy)
Pounds	14.5833	Ounces (troy)
Pounds (troy)	5760	Grains
Pounds (troy)	240	Pennyweights (troy)
Pounds (troy)	12	Ounces (troy)
Pounds (troy)	373.2417	Grams
Pounds (troy)	0.822857	Pounds (avoir.)
Pounds (troy)	13.1657	Ounces (avoir.)
Pounds (troy)	3.6735 x 10 ⁻⁴	Tons (long)
Pounds (troy)	4.1143 x 10 ⁻⁴	Tons (short)

Multiply	By	To Obtain
Pounds (troy)	3.7324 x 10 ⁻⁴	Tons (metric)
Pounds of water	0.01602	Cubic feet
Pounds of water	27.68	Cubic inches
Pounds of water	0.1198	Gallons
Pounds of water/min.	2.670 x 10 ⁻⁴	Cubic ft./sec
Pounds/cubic foot	0.01602	Grams/cubic cm.
Pounds/cubic foot	16.02	kgs./cubic centimeters
Pounds/cubic foot	5.787 x 10 ⁻⁴	Lbs./cubic inch
Pounds/cubic inch	27.68	Grams/cubic inch
Pounds/cubic inch	2.768 x 10 ⁻⁴	kgs./cubic meter
Pounds/cubic inch	1728	Lbs./cubic foot
Pounds/foot	1.488	kgs/meter
Pounds/inch	1152	Grams/cm.
Pounds/sq. in.	0.06895	Bars
Pounds/sq. in.	5.1715	Cm Hg @ 0° C
Pounds/sq. in.	0.070307	kg./cm ²
Pounds/sq. in.	6895	Newtons/m ²
Pounds/sq. in.	6895	Pascals
Pounds/sq. foot	0.01602	Feet of water
Pounds/sq. foot	4.882	kgs./sq. meter
Pounds/sq. foot	6.944 x 10 ⁻³	Pounds/sq. inch
Pounds/sq. inch	0.06804	Atmospheres
Pounds/sq. inch	2.307	Feet of water
Pounds/sq. inch	2.036	Inches of mercury
Pounds/sq. inch	703.1	kgs./sq. meter
Pounds/sq. foot	4.788 x 10 ⁻⁴	Bars
Pounds/sq. foot	0.035913	Cm Hg @ 0°C
Pounds/sq. foot	0.014139	In Hg @ 32°C
Pounds/sq. foot	4.8824 x 10 ⁻⁴	kg/cm ²
Pounds/sq. foot	47.880	Newtons/m ²
Pounds/sq. foot	47.880	Pascals
Quadrants (angle)	90	Degrees
Quadrants (angle)	5400	Minutes
Quadrants (angle)	1.571	Radians
Quarts (dry)	67.20	Cubic inches
Quarts (liq.)	57.75	Cubic inches
Quintal, Argentine	101.28	Pounds
Quintal, Brazil	129.54	Pounds
Quintal, Castile, Peru	101.43	Pounds
Quintal, Chile	101.41	Pounds
Quintal, Mexico	101.47	Pounds
Quintal, Metric	220.46	Pounds
Quires	25	Sheets
Radians	57.30	Degrees
Radians	3438	Minutes
Radians	0.637	Quadrants
Radians/sec.	57.30	Degrees/sec.
Radians/sec.	0.1592	Revolutions/sec.
Radians/sec.	9.549	Revolutions/min.
Radians/sec./sec.	573.0	Revs./min./min.
Radians/sec./sec.	0.1592	Revs./sec./sec.
Reams	500	Sheets
Revolutions	360	Degrees
Revolutions	4	Quadrants
Revolutions	6.283	Radians
Revolutions/min.	6	Degrees/sec.
Revolutions/min.	0.1047	Radians/sec.
Revolutions/min.	0.01667	Revolutions/sec.
Revolutions/min./min.	1.745 x 10 ⁻³	Rads./sec./sec.
Revolutions/min./min.	2.778 x 10 ⁻⁴	Rev./sec./sec.

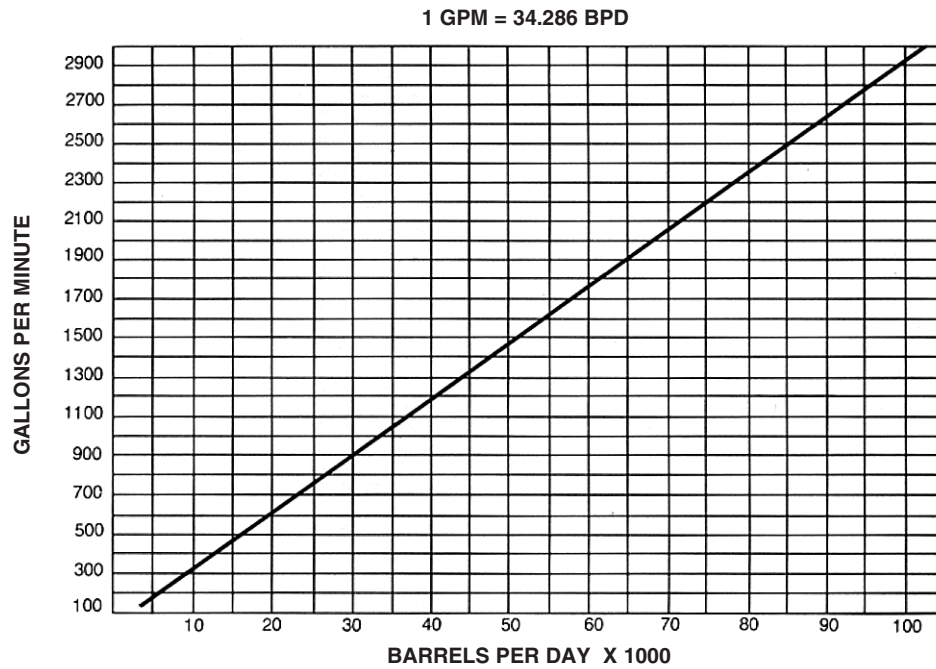
Multiply	By	To Obtain
Revolutions/ sec	360	Degrees/ sec.
Revolutions/ sec	6.283	Radians/ sec.
Revolutions/ sec	60	Revolutions/ min.
Revolutions/sec/sec	6,283	Radians/sec./sec
Revolutions/ sec/sec.	3600	Revs. / min/ min
Seconds (angle)	4.848 X 10 ⁻⁶	Radians
Square centimeters	1.076 X10 ⁻³	Square feet
Square centimeters	0.1550	Square inches
Square centimeters	10 ⁴	Square meters
Square centimeters	100	Square millimeters
Square feet	2.296 X 10 ⁻⁵	Acres
Square feet	929.0	Square centimeters
Square feet	144	Square inches
Square feet	0.09290	Square meters
Square feet	3.587 X10 ⁻⁴	Square miles
Square feet	1/9	Square yards
1		
Sq. ft./ gal. Min	8.0208	Overflow rate (ft. / hr.)
Square inches	6.542	Square centimeters
Square inches	6.944 X 10 ⁻³	Square feet
Square inches	645.2	Square millimeters
Square kilometers	247.1	Acres
Square kilometers	10.76 X 10 ⁶	Square feet
Square kilometers	10 ⁶	Square meters
Square kilometers	0.3861	Square miles
Square kilometers	1.196 X 10 ⁶	Square yards
Square meters	2.471 X10 ⁻⁴	Acres
Square meters	10.76	Square feet
Square meters	3.861 X 10 ⁻⁷	Square miles
Square meters	1.196	Square yards
Square miles	640	Acres
Square miles	27.88 x 10 ⁶	Square feet
Square miles	2.590	Square kilometers
Square miles	3.098 x 10 ⁶	Square yards
Square millimeters	0.01	Square centimeters
Square millimeters	1.550 x 10 ⁻³	Square inches
Square yards	2.066x 10 ⁻⁴	Acres
Square yards	9	Square feet

Multiply	By	To Obtain
Square yards	0.8361	Square Meters
Square yards	3.228 X 10 ⁻⁷	Square miles
Temp. (°C.) + 273	1	Abs. Temp. (° C.)
Temp. (° C.) +17.78	1.8	Temp. (° F.)
Temp. (° F.) + 460	1	Abs. Temp (° F.)
Temp (° F.) -32	5/9	Temp. (° C.)
Tons (long)	1016	Kilograms
Tons (long)	2240	Pounds
Tons (long)	1.12000	Tons (short)
Tons (metric)	10 ³	Kilograms
Tons (metric)	2205	Pounds
Tons (short)	2000	Pounds
Tons (short)	32,000	Ounces
Tons (short)	907. 1843	Kilograms
Tons (short)	2430.56	Pounds (troy)
Tons (short)	2430.56	Tons (long)
Tons (short)	29166.66	Ounces (troy)
Tons (short)	0.90718	Tons (metric)
Tons of water/ 24 hrs.	83.333	Pounds water/ hr.
Tons of water/24 hrs	0.16643	Gallons/ min.
Tons of water/ 24 hrs	1.3349	Cu. Ft. / hr.
Watts	0.05686	B.T.U./ min
Watts	44.25	Foot- Lbs. / min.
Watts	0.7376	Foot- Lb/sec.
Watts	1.341 X 10 ⁻³	Horsepower (U .S)
Watts	0.001360	Horsepower(metric)
Watts	1	Joules/ sec
Watts	0.01434	Kg- calories/ min.
Watts	10 ⁻³	Kilowatts
Watt- hours	3.414	B.T.U
Watt- hours	2655	Foot- Lbs
Watt- hours	1.341 X 10 ⁻³	Horsepower- hrs
Watt- hours	0.8604	Kilogram-calories
Watt- hours	367.1	kilogram- meters
Watt- hours	10 ⁻³	Kilowatt- hours
Yards	91.44	Centimeters
Yards	3	Feet
Yards	36	Inches
Yards	0.9144	Meters

TECH-H-5 Qwik Convert Tables

AREA		
inch ² x 645.16 = mm ²	mm ² x .00155 = inch ²	cm ² = square centimetre
inch ² x 6.4516 = cm ²	cm ² x 0.1550 = inch ²	mm ² = square millimetre
BENDING MOMENT (Torque)		
in- lbf x 0.113 = N • m	N • m x 8.85 = in-lbs	N • m = Newton- metre
ft- lbf x 1.356 = N • m		
CAPACITY (Volume per Unit Time)		
gpm x 0.2271 = m ³ /h	m ³ /h x 4.403 = gpm	m ³ /h = cubic metre per hour
gpm x 0.638 = liters per second	liters/ second x 15.85 = gpm	
FORCE		
lbf x 0.00448 = kN		kN = kilonewton
HEAD (& NPSH)		
foot x 0.3048 = m	m x 3.28084 = foot	m = metre
LENGTH		
foot x 304.8 = mm	mm x 0.003281 = feet	mm = millimetre
inch x 25.4 = mm	mm 0.03937 = inch	m = metre
foot x 0.3048 = m	m x 3.281 = foot	
MASS (Weight)		
ounce x 0.02853 = kg	kg x 2.205 = pound	kg = kilogram
pound x 0.4536 = kg	g x 0.03527 = ounce	g = gram
ounce x 28.35 = g		
POWER		
hp x 0.7457 = kW	kW x 1.340483 = hp	kW = kilowatt
PRESSURE		
psi x 0.0703 = kg/cm ²	kg/cm ² x 14.233578 = psi	kg/cm ² = kilogram/ square centimetre
psi x 6.895 = kPa	kPa x .145 = psi	kPa = kiloascal
kg/cm ² x 98.07 = kPa	kPa x 0.010197 = kg/cm ²	
psi x 0.06895 = Bar	Bar x 14.50377 = psi	
TEMPERATURE		
°C = 0.556 (°F - 32)	°F = (1.8 x °C) + 32	°C = degrees Celsius
VOLUME		
ft ³ x 0.02832 = m ³	m ³ x 35.31 = ft ³	m ³ = cubic metre
Gallon x 0.003785 = m ³	m ³ x 264 .17 = gallon	L = litre
Quart x 0.9464 = L	L x 1.057 = quart	mL = millilitre
Ounce x 29.57 = mL		
Gallon x 3.7854 = L	L X 0.26418 = gallon	

TECH-H-6 Conversion Chart—Gallons Per Minute to Barrels Per Day



TECH-H-7 Decimal and Millimeter Equivalents of Fractions

Inches		Millimeters	Inches		Millimeters
Fractions	Decimals		Fractions	Decimals	
1/64	.015625	.397	33/64	.515625	13.097
1/32	.03125	.794	17/32	.53125	13.494
3/64	.046875	1.191	35/64	.546875	13.891
1/16	.0625	1.588	9/16	.5625	14.288
5/64	.078125	1.984	37/64	.578125	14.684
3/32	.09375	2.381	19/32	.59375	15.081
7/64	.109375	2.778	39/64	.609375	15.487
1/8	.125	3.175	5/8	.625	15.875
9/64	.140625	3.572	41/64	.640625	16.272
5/32	.15625	3.969	21/32	.65625	16.669
11/64	.171875	4.366	43/64	.671875	17.066
3/16	.1875	4.763	11/16	.6875	17.463
13/64	.203125	5.159	45/64	.703125	17.859
7/32	.21875	5.556	22/32	.71875	18.256
15/64	.234375	5.953	47/64	.734375	18.653
1/4	.250	6.350	3/4	.750	19.050
17/64	.265625	6.747	49/64	.765625	19.447
9/32	.28125	7.144	25/32	.78125	19.844
19/64	.296875	7.541	51/64	.796875	20.241
5/16	.3125	7.938	13/16	.8125	20.638
21/64	.328125	8.334	53/64	.828125	21.034
11/32	.34375	8.731	27/32	.84375	21.431
23/64	.359375	9.128	55/64	.859375	21.828
3/8	.375	9.525	7/8	.875	22.225
25/64	.390625	9.922	57/64	.890625	22.622
13/32	.40625	10.319	29/32	.90625	23.019
27/64	.421875	10.716	59/64	.921875	23.416
7/16	.4375	11.113	15/16	.9375	23.813
29/64	.453125	11.509	61/64	.953125	24.209
15/32	.46875	11.906	31/32	.96875	24.606
31/64	.484375	12.303	63/64	.984375	25.003
1/2	.500	12.700	1	1.000	25.400

TECH-H-8 Atmospheric Pressures and Barometer Readings at Different Altitudes*

Altitude Below or Above Sea Level (Feet)	Barometer Reading Inches Mercury at 32° F	Atmospheric Pressure (PSI)	Equivalent Head of Water (75°) (Feet)	Boiling Point of Water ° F	° C
-1000	31.02	15.2	35.2	213.8	101.0
-500	30.47	15.0	34.7	212.9	100.5
0	29.921	14.7	34.0	212.0	100.0
+500	29.38	14.4	33.4	211.1	99.5
+1000	28.86	14.2	32.8	210.2	99.0
1500	28.33	13.9	32.2	209.3	98.5
2000	27.82	13.7	31.6	208.4	98.0
2500	27.31	13.4	31.0	207.4	97.4
3000	26.81	13.2	30.5	206.5	96.9
3500	26.32	12.9	29.9	205.6	96.4
4000	25.84	12.7	29.4	204.7	95.9
4500	25.36	12.4	28.8	203.8	95.4
5000	24.89	12.2	28.3	202.9	94.9
5500	24.43	12.0	27.8	201.9	94.4
6000	23.98	11.8	27.3	201.0	94.4
6500	23.53	11.5	26.7	200.1	93.9
7000	23.09	11.3	26.2	199.2	92.9
7500	22.65	11.1	25.7	198.3	92.4
8000	22.22	10.9	25.2	197.4	91.9
8500	21.80	10.7	24.8	196.5	91.4
9000	21.38	10.5	24.3	195.5	90.8
9500	20.98	10.3	23.8	194.6	90.3
10,000	20.58	10.1	23.4	193.7	89.8
15,000	16.88	8.3	19.1	184	84.4
20,000	13.75	6.7	15.2	-	-
30,000	8.88	4.4	10.2	-	-
40,000	5.54	2.7	6.3	-	-
50,000	3.44	1.7	3.9	-	-

*Approximate Values



Section TECH-I

Pump Operation and Maintenance

TECH-I-1 Pump Safety Tips

Maintenance personnel should be aware of potential hazards to reduce the risk of accidents...

Safety Apparel:

- Insulated work gloves when handling hot bearings or using bearing heater.
- Heavy work gloves when handling parts with sharp edges, especially impellers.
- Safety glasses (with side shields) for eye protection, especially in machine shop areas.
- Steel-toed shoes for foot protection when handling parts, heavy tools, etc.
- Other personal protective equipment to protect against hazardous/toxic fluids.

Couplings Guards:

- Never operate pump without a coupling guard properly installed.

Flanged Connections:

- Never force piping to make a connection with a pump.
- Use only fasteners of the proper size and material.

- Ensure there are no missing fasteners.
- Beware of corroded or loose fasteners.

Operation:

- Do not operate below minimum rated flow, or with suction/discharge valves closed.
- Do not open vent or drain valves, or remove plugs while system is pressurized.

Maintenance Safety:

- Always lockout power.
- Ensure pump is isolated from system and pressure is relieved before disassembling pump, removing plugs, or disconnecting piping.
- Use proper lifting and supporting equipment to prevent serious injury.
- Observe proper decontamination procedures.
- Know and follow company safety regulations.
- Never apply heat to remove impeller.
- Observe all cautions and warnings highlighted in pump instruction manual.

TECH-I-2 PRO Service Centers: An Economical Alternative



Goulds offers an economical alternative to high maintenance costs. Goulds PRO Service Centers are experienced with reconditioning all types of pumps and rotating equipment, restoring equipment to original specifications. Users continually utilize PRO Service Centers for economical repair versus replacement, decreased downtime, reduced inventory of replacement parts and the advantage of updated engineering technology.

Benefits/Services:

- Factory trained service personnel
- 24-hour emergency service
- Machine shop facilities
- Inventory of replacement parts
- Repairs to all makes and manufacture of pumps
- Pickup and delivery service
- Pump installation supervision
- Technical advisory services
- Turnkey field service capability
- Vertical turbine rebowling

Contact your nearest Goulds sales office for location of your nearest PRO Service Center.

TECH-I-3 Symptoms and Causes of Hydraulic and Mechanical Pump Failure

Cause	Hydraulic Failure					Mechanical Failure				
	1	2	3	4	5	6	7	8	9	10
	Pump does not deliver liquid	Pump does not deliver sufficient capacity	Pump does not deliver sufficient pressure	Pump delivers flow intermittently	Bearings run hot and/or fail on a regular basis	High rate of mechanical seal failure	Packing has short life	Pump vibrates at higher-than-normal levels	Pump is drawing too much power	Wear of internal wetted parts is accelerated
Pump not primed or prime lost										
Suction and/or discharge valves closed or clogged										
Suction piping incorrect										
Insufficient NPSH available										
Excessive air entrapped in liquid										
Speed (RPM) too low										
Incorrect rotation										
Broken impeller or bent vanes										
Incorrect impeller or impeller diameter										
System head too high										
Instruments give erroneous readings										
Air leaks in suction line										
Excessive shaft misalignment										
Inadequate lubrication										
Lubricant contamination										
Inadequate lubricant cooling										
Axial thrust or radial loads higher than bearing rating										
Improper coupling lubrication										
Suction pressure too high										
Bearing incorrectly installed										
Impeller out of balance										
Overheating of seal faces										
Excessive shaft deflection										
Lack of seal flush at seal faces										
Incorrect seal installation										
Pump is run dry										
Pump run off design point										
Shaft/shaft sleeve worn										
Packing gland not properly adjusted										
Packing not properly installed										
Impeller clogged										
Coupling out of balance										
Baseplate not installed properly										
Pump operating speed too close to system's natural frequency										
Bearing failing										
Piping not properly anchored										
Pump and/or driver not secured to baseplate										
Specific gravity higher than specified										
Viscosity higher than specified										
Internal clearances too tight										
Chemicals in liquid other than specified										
Pump assembled incorrectly										
Higher solids concentration than specified										

TECH-I-4 Troubleshooting Centrifugal Pumps

Problem	Probable Cause	Remedy
No liquid delivered.	Pump not primed.	Reprime pump, check that pump and suction line are full of liquid.
	Suction line clogged.	Remove obstructions.
	Impeller clogged with foreign material.	Back flush pump to clean impeller.
	Wrong direction of rotation.	Change rotation to concur with direction indicated by arrow on bearing housing or pump casing.
	Foot valve or suction pipe opening not submerged enough.	Consult factory for proper depth. Use baffle to eliminate vortices.
	Suction lift too high.	Shorten suction pipe.
Pump not producing rated flow or head.	Air leak through gasket.	Replace gasket.
	Air leak through stuffing box.	Replace or readjust packing/mechanical seal.
	Impeller partly clogged.	Back flush pump to clean impeller.
	Worn suction sideplate or wear rings.	Replace defective part as required.
	Insufficient suction head.	Ensure that suction line shutoff valve is fully open and line is unobstructed.
	Worn or broken impeller.	Inspect and replace if necessary.
Pump starts then stops pumping.	Improperly primed pump.	Reprime pump.
	Air or vapor pockets in suction line.	Rearrange piping to eliminate air pockets.
	Air leak in suction line.	Repair (plug) leak.
	Improper alignment.	Re-align pump and drive.
Bearings run hot.	Improper lubrication.	Check lubricate for suitability and level.
	Lube cooling.	Check cooling system.
	Improper pump/driver alignment.	Align shafts.
Pump is noisy or vibrates.	Partly clogged impeller causing imbalance.	Back-flush pump to clean impeller.
	Broken or bent impeller or shaft.	Replace as required.
	Foundation not rigid.	Tighten hold down bolts of pump and motor or adjust stilts.
	Worn bearings.	Replace.
	Suction or discharge piping not anchored or properly supported.	Anchor per Hydraulic Institute Standards Manual recommendation.
	Pump is cavitating.	System problem.
	Packing gland improperly adjusted.	Tighten gland nuts.
	Stuffing box improperly packed.	Check packing and repack box.
Excessive leakage from stuffing box/seal chamber.	Worn mechanical seal parts.	Replace worn parts.
	Overheating mechanical seal.	Check lubrication and cooling lines.
	Shaft sleeve scored.	Remachine or replace as required.
	Head lower than rating. Pumps too much liquid.	Consult factory. Install throttle valve, trim impeller diameter.
	Liquid heavier than expected.	Check specific gravity and viscosity.
Motor requires excessive power.	Stuffing packing too tight.	Readjust packing. Replace if worn.
	Rotating parts bind.	Check internal wearing parts for proper clearances.

TECH-I-5 Abrasive Slurries and Pump Wear

THE EFFECTS OF OPERATING AT DIFFERENT ZONES ON THE PUMP CHARACTERISTIC CURVE

The rate of wear is directly influenced by the system point on the characteristic curve. These condition points can be divided into four significant zones of operation (Fig. 1).

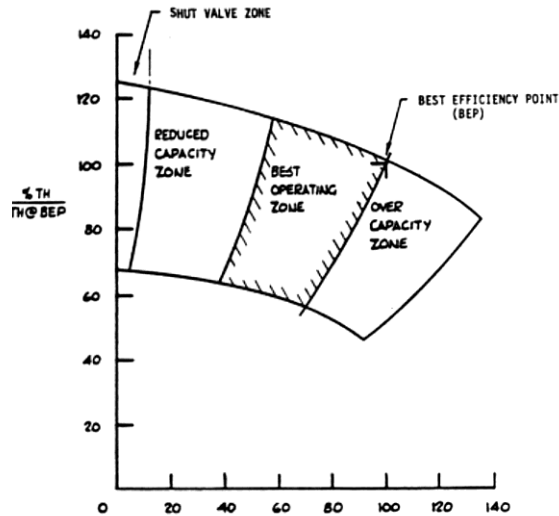


Fig. 1 Slurry Pump Characteristic Curve

Overcapacity Zone:	The velocities within the pump are usually very high and recirculation occurs causing excessive wear. The radial hydraulic loads on the impeller increase.
Recommended Operation Zone:	The velocities within the pump are reduced (but not enough to cause settlement). Recirculation is minimal and the flow in the suction nozzle should be axial (no induced vortex). The radial hydraulic loads are minimized.
Reduced Capacity Zone	The velocities within the pump are low, separation and recirculation occurs, causing excessive wear. Reducing the capacity should be limited because a certain minimum velocity must be maintained to avoid settling out; with the consequence of increased wear and clogging. The hydraulic radial loads will increase and the pump efficiency will decrease.
Shut Valve Zone:	This is the point of zero flow, and pump should not be operated at this point for any length of time. Wear and tear will be rapid due to separation and recirculation, the hydraulic forces will be at their highest, and settlement and plugging will occur. The pump will rapidly heat up, which is particularly serious in rubber constructed pumps.

PRINCIPAL WEAR AREAS

As the abrasive mixture passes through the pump, all the wetted surfaces which come in contact will be subject to varying degrees of wear. It is very important to note that the performance of a conventional centrifugal pump, which has been misapplied to a slurry service, will be significantly effected by a relatively small degree of abrasive wear.

The areas most prone to wear, in order of severity, are:

1. Suction sideplate, particularly at the nozzle region.
2. Impeller, particularly at the eye vane inlets, suction side impeller shroud, and the vane tips.
3. Casing cutwater and side walls adjacent to the impeller tip.
4. Stuffing box packing and sleeve.

NOTE: In the case of a conventional pump with radial wear rings on the impeller, this is where the worst wear occurs.

On severely abrasive services where there are high concentrations of hard, larger, sharp particles, the suction side liner life can be increased if it is rotated periodically to equalize the effects of wear.

In hard iron pumps applied to severely abrasive service, the relative wear rates of the suction side liner, casing, and impeller are in the order of 3 to 1.5 to 1, e.g. the life of the casing is three times that of a suction side wear plate.

Recognizing that due to the nature of the mixtures being pumped, the complete elimination of wear is impossible, the life of the parts can be appreciably prolonged and the cost of maintenance reduced by a good pump design and selection, e.g.:

- Construct the pump with good abrasion resistant materials.
- Provide generous wear allowances on all parts subject to excessive wear.
- Adopt a hydraulic design which will minimize the effects causing wear.
- Adopt a mechanical design which is suitable for the materials of construction and has ready access to the parts for renewal.
- Limit the head to be generated and select a low speed pump.

TECH-I-6 Start-Up and Shut-Off Procedure for Heated and Unheated Mag Drive Pumps

(This procedure does not replace the operation instruction handbook.)

A. CHECKLIST BEFORE START-UP

1. The nominal motor power must not exceed the pump's allowed maximum capacity (compare rating plates of motor and pump).
2. Check direction of rotation with disconnected coupling.
3. Check alignment of coupling.
4. Check ease of pump operation by hand.
5. Attach coupling protection.
6. Connect thermocouples, dry run protection, pressure gauges, etc.
7. Connect heater for heated pumps.
8. Connect cooling system (if required).
9. Attention: Insulation must not cover roller bearings.

B. START-UP

1. Preheat heated pumps for a minimum of 2 hours.
2. Open pressure valve.
3. Open suction valve completely and fill pump.
4. After 2-3 minutes close pressure valve.
5. In case of external cooling, switch on coolant flow.
6. Start motor.
7. Subsequently open pressure valve slowly until pump reaches specified performance level.

C. SHUT-OFF

1. Close pressure valve.
2. Shut off motor. Allow pump to slow down smoothly.
3. In case of external cooling, shut off coolant flow.
4. Close suction valve.

NOTE:

- Throttling must not be done with the suction valve.
- Never shut off the pump with the suction valve.
- Pump must never run dry.
- Never run the pump against a closed pressure valve.
- The pump motor unit must run vibration free.
- Temperature of roller bearings must not exceed tolerated limit.

TECH-I-7 Raised Face and Flat Face Flanges (Mating Combinations)

Pumps of cast iron construction are furnished with 125 or 250 lb. flat face (F.F.) flanges. Since industry normally uses fabricated steel piping, the pumps are often connected to 150 or 300 lb. 1/16" raised face (R.F.) steel flanges.

Difficulty can occur with this flange mating combination. The pump flange tends to pivot around the edge of the raised face as the flange bolts are tightened. This can cause the pump flange to break allowing leakage at the joint. (Fig. 1).

A similar problem can be encountered when a bronze pump with F.F. flanges is connected to R.F. steel flanges (Fig. 2). Since the materials are not of equal strength, the bronze flange may distort, resulting in leakage.

To avoid problems when attaching bronze or cast iron F.F. pump flanges to R.F. steel pipe flanges, the following steps should be taken (refer to Fig. 3).

1. Machine off the raised face on the steel pipe flange.
2. Use a full face gasket.

If the pump is steel or stainless steel with F.F. flanges, no problem arises since materials of equal strength are being connected. Many customers, however, specify R.F. flanges on steel pumps for mating to R.F. companion flanges. This arrangement is technically and practically not required.

The purpose of a R.F. flange is to concentrate more pressure on a smaller gasket area and thereby increase the pressure containment capability of the joint. To create this higher gasket load, it is only necessary to have one-half of the flanged joint supplied with a raised face - not both. The following illustrations show 4" steel R.F. and F.F. mating flange combinations and the gasket loading incurred in each instance.

Assuming the force (F) from the flange bolts to be 10,000 lbs. and constant in each combination, the gasket stress is:

$$P \text{ (Stress)} = \frac{\text{Bolt Force (F)}}{\text{Gasket Area}}$$

$$P_1 \text{ (Fig. 4)} = \frac{10,000 \text{ lbs.}}{49.4 \text{ sq. in.}} = 203 \text{ psi}$$

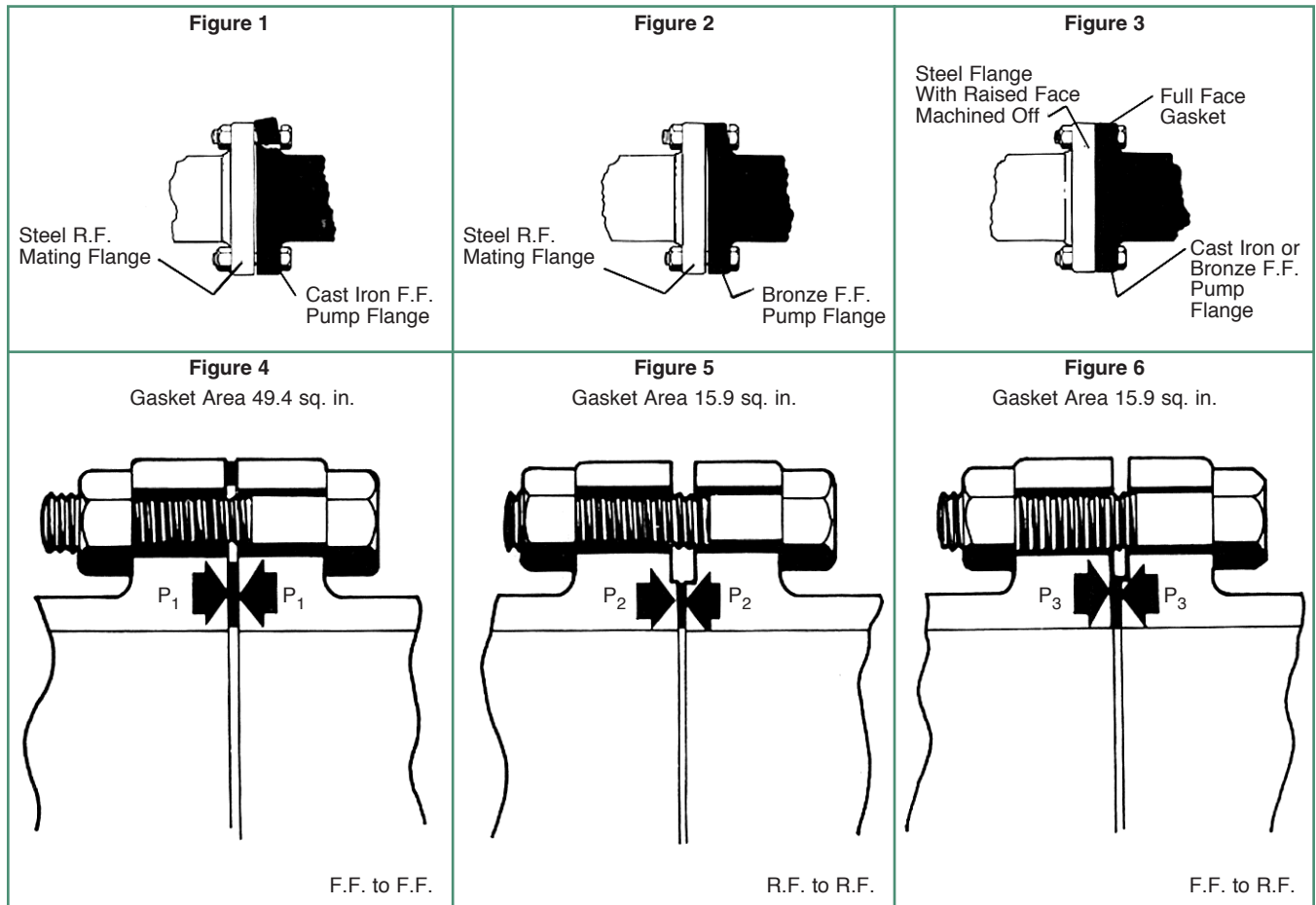
$$P_2 \text{ (Fig. 5)} = \frac{10,000 \text{ lbs.}}{15.9 \text{ sq. in.}} = 630 \text{ psi}$$

$$P_3 \text{ (Fig. 6)} = 15.9 \text{ sq. in.}$$

It can be readily seen that the smaller gasket, used with a raised face flange, increases the pressure containment capability of a flanged joint. However, it can also be noted that there is no difference in pressure capability between R.F.-to-R.F. and R.F.-to-F.F. flange combinations.

In addition to being technically unnecessary to have a R.F.-to-R.F. mating combination, the advantages are:

1. The elimination of the extra cost for R.F. flanges.
2. The elimination of the extra delivery time required for a non-standard casing.



TECH-I-8 Keep Air Out of Your Pump

Most centrifugal pumps are not designed to operate on a mixture of liquid and gases. To do so is an invitation to serious mechanical trouble, shortened life and unsatisfactory operation. The presence of relatively small quantities of air can result in considerable reduction in capacity, since only 2% free air will cause a 10% reduction in capacity, and 4% free air will reduce the capacity by 43.5%.

In addition to a serious loss in efficiency and wasted power, the pump may be noisy with destructive vibration. Entrained air is one of the most frequent causes of shaft breakage. It also may cause the pump to lose its prime and greatly accelerate corrosion.

Air may be present in the liquid being pumped due to leaky suction lines, stuffing boxes improperly packed, or inadequately sealed on suction lift or from other sources.

Refer also to Section TECH-D-7, Pumping Liquids with Entrained Gas.

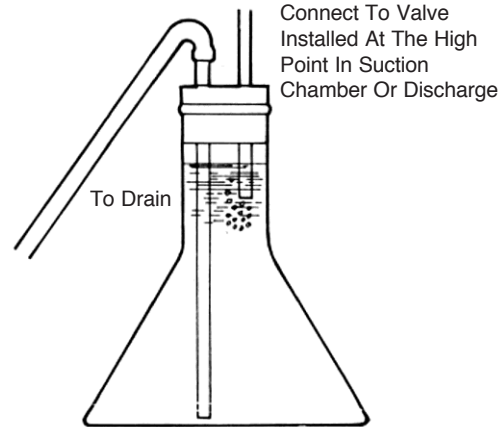
On the other hand, very small amounts of entrained air (less than 1%) can actually quiet noisy pumps by cushioning the collapse of cavitation bubbles.

TESTING FOR AIR IN CENTRIFUGAL PUMPS

The amount of air which can be handled with reasonable pump life varies from pump to pump. The elimination of air has greatly improved the operation and life of many troublesome pumps. When trouble occurs, it is common to suspect everything but air, and to consider air last, if at all.

In many cases a great deal of time, inconvenience, and expense can be saved by making a simple test for the presence of air. We will assume that calculations have already been made to determine that there is sufficient NPSH Margin (2 - 5 times the NPSHR) to insure that the noise is not due to cavitation. The next step should be to check for the presence of entrained air in the pumpage.

When the source of suction supply is above the centerline of the pump, a check for air leaks can be made by collecting a sample in a "bubble bottle" as illustrated. Since the pressure at the suction chamber of the pump is above atmospheric pressure, a valve can be installed in one of the tapped openings at the high point in the chamber and liquid can be fed into the "bubble bottle." The presence of air or vapor will show itself in the "bubble bottle."



This test can also be made from a high point in the discharge side.

Obviously, the next step is to eliminate the source of air since quantities present in insufficient amount to be audible are almost certain to cause premature mechanical failure.

NOTE: The absence of bubbles is not proof that the pumpage doesn't contain air.

TECH-I-9 Ball Bearings – Handling, Replacement and Maintenance Suggestions

Ball bearings are carefully designed and made to watch-like tolerances. They give long, trouble-free service when properly used. They will not stand abuse.

KEEP CLEAN

Dirt causes 90% of early bearing failures. Cleanliness is a must when working on bearings. Some things which help:

1. Do not open housings unless absolutely necessary.
2. Spread clean newspapers on work benches and at pump. Set tools and bearings on papers only.
3. Wash hands. Wipe dirt, chips and grease off tools.
4. Keep bearings, housings, and shaft covered with clean cloths whenever they are not being worked on.
5. Do not unwrap new bearings until ready to install.
6. Flush shaft and housing with clean solvent before reassembly.

PULL BEARINGS CAREFULLY

1. Use sleeve or puller which contacts just inner race of bearing. (The only exception to this is some double suction pumps which use the housing to pull the bearing.)
2. Never press against the balls or ball cages, only against the races.
3. Do not cock bearing. Use sleeve which is cut square, or puller which is adjusted square.
4. When using a bearing housing to pull a bearing, pull evenly, do not hammer on housing or shaft. With both races locked, shock will be carried to balls and ruin bearing.

INSPECT BEARINGS AND SHAFT

1. Look bearing over carefully. Scrap it if there are any flat spots, nicks or pits on the balls or races. Bearings should be in perfect shape.
2. Turn bearing over slowly by hand. It should turn smoothly and quietly. Scrap if "catchy" or noisy.

3. Whenever in doubt about the condition of the bearing, scrap it. Five or ten dollars worth of new bearings may prevent serious loss from downtime and pump damage. In severe or critical services, replace bearings at each overhaul.
4. Check condition of shaft. Bearing seats should be smooth and free from burrs. Smooth burrs with crocus cloth. Shaft shoulders should be square and not run over.

CHECK NEW BEARINGS

Be sure bearing is of correct size and type. For instance, an angular contact bearing which is dimensionally the same as a deep groove bearing may fit perfectly in the pump. However, the angular contact bearing is not suitable for end thrust in both directions, and may quickly fail. Also check to see that shields (if any) are the same as in the original unit. Refer to the pump instruction manual for the proper bearing to use.

INSTALL CAREFULLY

1. Oil bearing seat on shaft lightly.
2. Shielding, if any, must face in proper direction. Angular contact bearings, on pumps where they are used, must also face in the proper direction. Duplex bearings must be mounted with the proper faces together. Mounting arrangements vary from model to model. Consult instruction manual for specific pump.
3. Press bearing on squarely. Do not cock it on shaft. Be sure that the sleeve used to press the bearing on is clean, cut square, and contacts the inner race only.
4. Press bearing firmly against shaft shoulder. The shoulder helps support and square the bearing.
5. Be sure snap rings are properly installed, flat side against bearing, and that lock nuts are tight.
6. Lubricate properly, as directed in instruction manual.

TECH-I-10 Impeller Clearance

IMPELLER CLEARANCE

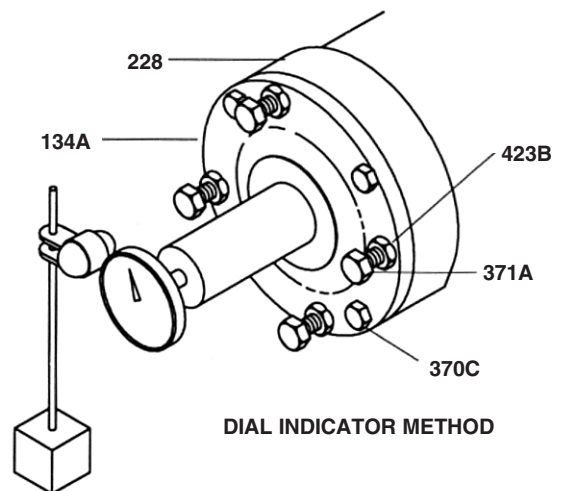
Open impeller centrifugal pumps offer several advantages. They're particularly suited but not restricted to liquids which contain abrasive solids. Abrasive wear on an open impeller is distributed over the diametrical area swept by the vanes. The resulting total wear has less effect on performance than the same total wear concentrated on the radial ring clearance of a closed impeller.

The open impeller permits restoration of "new pump" running clearance after wear has occurred without parts replacement. Many of Goulds open impeller pumps feature a simple positive means for axial adjustment without necessity of disassembling the unit to add shims or gaskets.

SETTING IMPELLER CLEARANCE (DIAL INDICATOR METHOD)

1. After locking out power, remove coupling guard and coupling.
2. Set dial indicator so that button contacts shaft end.
3. Loosen jam nuts (423B) on jack bolts (371A) and back bolts out about two turns.
4. Tighten each locking bolt (370C) evenly, drawing the bearing housing toward the bearing frame until impeller contacts casing.
5. Set indicator to zero and back locking bolt about one turn.
6. Thread jack bolts in until they evenly contact the bearing frame. Tighten evenly backing the bearing housing away from the frame until indicator shows the proper clearance established in instruction manual.*

7. Evenly tighten locking bolts, the jack bolts keeping indicator at proper setting.
 8. Check shaft for free turning.
- *Established clearance may vary due to service temperature.



TECH-I-11 Predictive and Preventative Maintenance Program

This overview of Predictive and Preventative Maintenance (PPM) is intended to assist the pump users who are starting a PPM program or have an interest in the continuous improvement of their current programs.

There are four areas that should be incorporated in a PPM program. Individually each one will provide information that gives an indication of the condition of the pump; collectively they will provide a complete picture as to the actual condition of the pump.

PUMP PERFORMANCE MONITORING

There are six parameters that should be monitored to understand how a pump is performing. They are Suction pressure (P_s), discharge pressure (P_d), flow (Q), pump speed (N_r), pumpage properties, and power. Power is easiest measured with a clip on amp meter but some facilities have continuous monitoring systems that can be utilized. In any event, the intent is to determine the BHP of the pump. When using a clip on amp meter the degree of accuracy is limited. It

should not be used to determine the efficiency of the pump. Clip on amp meters are best used for trouble shooting where the engineer is trying to determine the operating point of the pump.

The most basic method of determining the TDH of the pump is by utilizing suction and discharge gauges to determine P_S and P_d . The installation of the taps for the gauges is very important. Ideally, they should be located normal to the pipe wall and on the horizontal centerline of the pipe. They should also be in a straight section of pipe. Avoid locating the taps in elbows or reducers because the readings will not indicate the true static pressure due to the velocity head component. Avoid locating taps in the top or bottom of the pipe because the gauges can become air bound or clogged with solids.

Flow measurements can be difficult to obtain but every effort should be made to do so, especially when trouble shooting. In some new installations permanent flow meters are installed which make the job easier. When this is the case, make sure the flow meters are working properly and have been calibrated on a regular schedule. When flow meters are not installed, pitot tubes can be used. Pitot tubes provide a very accurate measure of flow, but this in an obtrusive device and provisions must be made to insert the tube into the piping. The other method of determining flow is with either a doppler or transimite device. Again, provisions must be made on the piping for these instruments, but these are non-obtrusive devices and are easier to use than the pitot tube. Caution must be exercised because each device must be calibrated, and independent testing has shown these devices are sensitive to the pumpage and are not 100% accurate.

An accurate power measurement reading can also be difficult to obtain. Clip on map meters are the most common tool available to the Field Engineer who is trouble shooting a pump problem. In most cases this has proven to be accurate. However, as previously mentioned, this tool must be used and applied properly. Clip on map meters are not accurate enough to determine the actual efficiency of a pump. If accurate horsepower readings are necessary, a torque shaft must be installed but is not very practical in an actual field installation and lends itself to use in a laboratory environment much better. In some critical installations where the user has provided a permanent power monitor, these have varying degrees of accuracy and they must be understood up front.

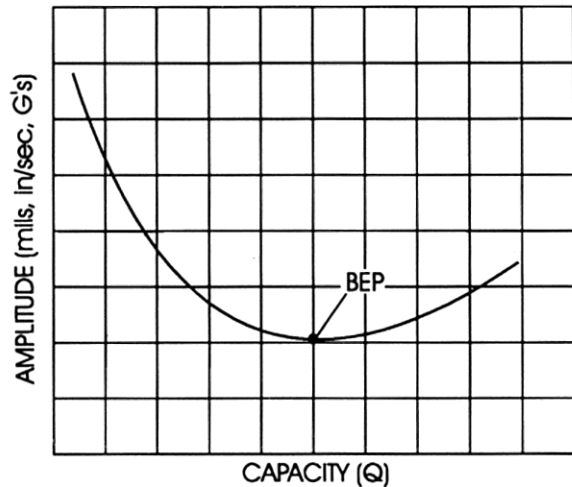
Finally, the properties of the pumpage must be known to accurately determine the actual pump performance. Pumpage temperature (T_p), viscosity, and specific gravity (S.G.), must be known.

When all of the above parameters are known, it becomes a simple matter of calculating the pump performance. There are instances when it proves to be a very difficult if not an impossible task to determine all of the above parameters in the field, therefore, the Field Engineer must rely on his or her ability to understand where a compromise must be made to get the job done. The basic document the Field Engineer must have is the pump performance curve. With this it can be determined where the pump is performing in some cases without all of the information.

PUMP VIBRATION AND BEARING ANALYSIS

Vibration analysis is the cornerstone of all PPM programs. Perhaps the question asked most often is "What is the vibration level that indicates the pump is in distress?". The answer is that there is no absolute vibration amplitude level that is indicative of a pump in distress. However, there are several guidelines that have been developed as target values that enable the analyst to set alarm levels. Also many users have developed their own site criteria that is used as a guideline. Institutions such as the Hydraulic Institute and API have developed independent vibration criteria. Caution should be exercised when applying the published values...each installation is unique and should be handled accordingly. When a machine is initially started, a baseline vibration reading should be taken and trended over time.

Typically, readings are taken on the motor outboard and inboard bearing housings in the vertical and horizontal directions and on the pump outboard and inboard bearing housings in the vertical and horizontal directions. Additionally, an axial vibration measurement is taken on the pump. The inboard location is defined as the coupling end of the machine. It is critical that when the baseline vibration measurement is taken that the operating point of the pump is also recorded. The vibration level of a pump is directly related to where it is operating and in relation to its Best Efficiency Point (BEP). The further away from the BEP, the higher the vibrations will be. See the following chart for a graphical representation of vibration amplitude-vs- flow.



Typical Vibration Level Characteristic vs Capacity

The engineer must also look at the frequency where the amplitude is occurring. Frequency identifies what the defect is that is causing the problem, and the amplitude is an indication of the severity of the problem. These are general guidelines and do not cover every situation. The spectrum in the chart is a typical spectrum for a pump that has an unbalance condition.

Bearing defect analysis is another useful tool that can be used in many condition monitoring programs. Each component of a roller bearing has its own unique defect frequency. Vibration equipment available today enables the engineer to isolate the unique bearing defects and determine if the bearing is in distress. This allows the user to shut the machine down prior to a catastrophic failure. There are several methods utilized but the most practical from a Field Engineering perspective is called bearing enveloping. In this method, special filters built into the analyzer are used to amplify the repetitive high frequency signals in the high frequency range and amplify them in the low frequency part of the vibration spectrum. Bearing manufacturers publish the bearing defect frequency as a function of running speed which allows the engineer to identify and monitor the defect frequency. Similar to conventional vibration analysis, a baseline must be established and then trended. There are other methods available such as High Frequency Detection (HFD), and Spike Energy but the enveloping technology is the latest development.

It is a common practice to monitor bearing temperature. The most accurate method to monitor the actual bearing temperature is to use a device that will contact the outer race of the bearing. This requires holes to be drilled into the bearing housings which is not always practical. The other method is the use of an infrared 'gun' where the analyst aims the gun at a point on the bearing housing where the temperature reading is going to be taken. Obviously, this method is the most convenient but there is a downside. The temperature being measured is the outside surface of the bearing housing, not the actual bearing temperature. This must be considered when using this method.

To complete the condition monitoring portion of a PPM program, many users have begun an oil analysis program. There are several tests that can be performed on the lubricant to determine the condition of the bearing or determine why a bearing failed so appropriate corrective action can be taken. These tests include Spectrographic Analysis, Viscosity Analysis, Infrared Analysis, Total Acid Number, Wear Particle Analysis and Wear Particle Count. Most of these tests have to be performed under laboratory conditions. Portable instruments are now available that enable the user to perform the test on site.

PUMP SYSTEM ANALYSIS

Pump system analysis is often overlooked because it is assumed the system was constructed and operation of the pumps are in accordance with the design specifications. This is often not the case. A proper system analysis begins with a system head curve. System head curves are very difficult to obtain from the end user and, more often than not, are not available. On simple systems, they can be generated in the field but on more complicated systems this can't be done. As has been stated previously, it is imperative to know where the pumps are being operated to perform a correct analysis and this is dependent on the system.

A typical system analysis will include the following information; $NPSH_A$, $NPSH_R$, static head, friction loss through the system, and a complete review of the piping configuration and valving. The process must also be understood because it ultimately dictates how the pumps are being operated. All indicators may show the pump is in distress when the real problem is it is being run at low or high flows which will generate high hydraulic forces inside the pump.

CONCLUSION

A PPM program that incorporates all of the topics discussed will greatly enhance the effectiveness of the program. The more complete understanding the engineer has of the pumping system, the more effective the PPM program becomes.

TECH-I-12 Field Alignment

Proper field alignment of pumps and drivers is critical to the life of the equipment. There are three methods used in industry: rim and face, reverse dial indicator, and laser alignment.

RIM AND FACE

This method should not be used when there is no fixed thrust bearing or on pumps/drivers that have axial shaft movement.

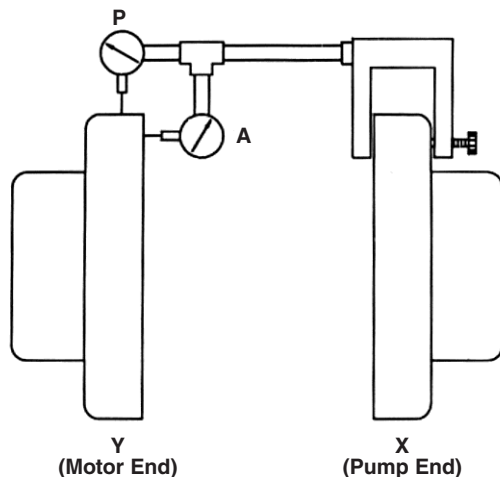


Fig. 1 Rim and Face Dial Indicator Alignment
(Criteria: 0.002 in. T.I.R. rim and face reading)

REVERSE DIAL INDICATOR

This method is the most widely used and is recommended for most situations.

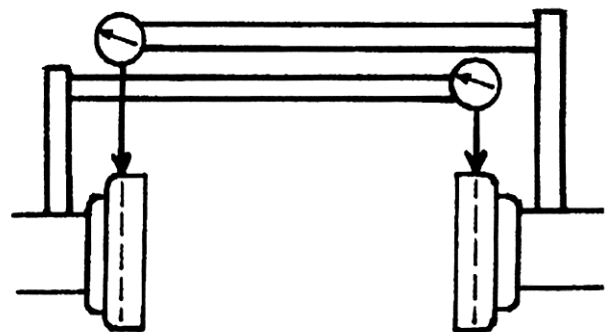


Fig. 2 Reverse Dial Indicator Alignment
(Criteria: 0.0005 in. per inch of dial indicator separation)

LASER ALIGNMENT

Although a popular method, it's not any more accurate than either dial indicator method. Instruments are expensive and require frequent calibration.

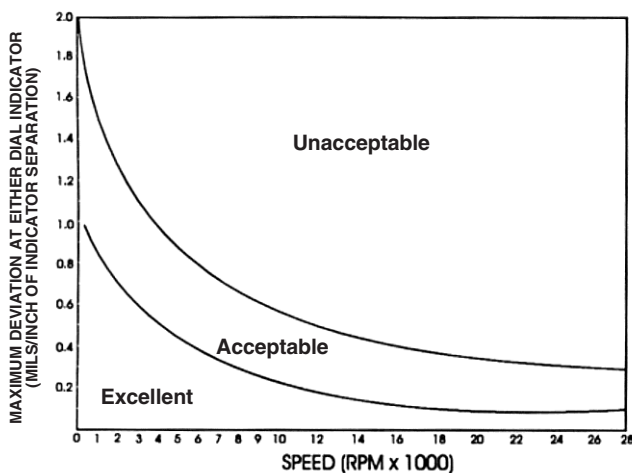


Fig. 3 Guideline for Alignment Tolerances

MECHANICAL ALIGNMENT PROCEDURE

This procedure assumes the presenter knows how to align a pump and has a basic understanding of pump baseplates and piping installation. There are many alignment systems available. We will be using the plotting board with dial indicators developed by M.G. Murray. The plotting board is as accurate as any method available today and gives the best representation of the actual position of the machines that are being aligned. The actual procedure that will be discussed is the reverse dial indicator procedure because it is the most versatile and widely used alignment procedure used today.

PREPARING FOR ALIGNMENT

A. Baseplate Inspection

1. Inspect all mounting surfaces to make sure they are clean and free of any paint, rust, grime, burrs, etc.
 - a. Thoroughly clean mounting surfaces. Debar using a honing stone if necessary.
 - b. At this point, it is assumed that the baseplate has been installed correctly and is level.

B. Pump and Driver Inspection

1. Inspect all mounting surfaces to make sure they are clean and free of any paint, rust, grime, burrs, etc.

C. Shim Inspection

1. Inspect all shims to make sure they are clean and free of any paint, rust, grime, burrs, etc.
2. Dimensionally inspect ALL shims to be used and record the reading on the individual shims.

DO NOT ASSUME THAT THE SHIMS ARE TO THE EXACT DIMENSIONS THAT ARE RECORDED ON THEM.

SETTING EQUIPMENT

A. Pump

1. Set pump on pump mounting pads. Insert pump hold-down bolts but do not tighten.
 - a. If there is existing piping, line up pump flanges with pipe flanges. DO NOT CONNECT THE PIPING AT THIS POINT.

2. Level the pump off of the shaft extension. Do not level off of the pump casing flanges. Remember, the piping must come to the pump. You are aligning the pump shaft and the driver shaft. Shafts are the datum, not flanges.

- a. Use a STARRET No.135 level to level the shaft.
- b. Leveling the pump should be accomplished by shimming under the bearing frame foot.

B. Motor

1. Set the motor on the baseplate.
2. Using a straight edge, approximate the shaft alignment.
 - a. This will require setting shims of the same thickness under the motor feet; you are just trying to get close so you can use the dial indicators. Get the rough alignment within 0.0625".
 - b. If the motor is higher, there is something wrong or it is a special case. This situation must be inspected. Do not shim the pump. The pump is connected to the piping and it will present difficulties with future work on the installation.
 - c. Make sure you have the proper shaft separation.

3. Remove soft foot.

C. Alignment. (Reverse indicator Method)

1. Install reverse dial indicator tooling on shafts.
2. Measure and record the following dimensions on a worksheet, SA, AI, IO. These parameters are defined as follows:
 - a. SA = Distance between the dial indicators which are located at the respective planes of correction.
 - b. AI = Distance between the adjustable plane of correction and the inboard foot of the adjustable machine.
 - c. IO = Distance between the inboard foot and outboard foot of the adjustable machine.
2. Correct for dial indicator sag.
 - a. Remove dial indicator tooling from the unit.
 - b. Install reverse dial indicator tooling on a pipe or piece of round bar stock in the exact configuration that you removed it from the unit that is being aligned. The dial indicators must be set to the SA distance.
 - c. Zero the dial indicator while they are in the vertical up position.
 - d. Rotate the entire set-up 180° and record dial indicator readings. This is the sag, the correction will be made when you take the alignment readings.
3. Reinstall the reverse dial indicator tooling back to the configuration it was in Step 1.
 - a. The SA dimension must be held.
4. Establishing the datums.
 - a. You must take readings from the same position relative to the fixed machine or the moveable machine. Choose the position that is the most comfortable. DO NOT CHANGE THE ORIENTATION ONCE YOU BEGIN TO TAKE READINGS.
 - b. All dial indicator readings must be taken 90° apart from each other and at the same relative position each time. Either mark the couplings in 80° increments or use a two dimension bubble level with a magnetic pad. The level is the most accurate method.

- c. The shafts must be rotated together and readings taken from the same exact locations every time; therefore, if the coupling spacer is removed, the stationary and adjustable machines coupling hubs must be marked in 90° increments.
5. Take the initial set of readings.
 - a. Zero the dial indicators at the 0° position.
 - b. Rotate the shafts simultaneously taking readings every 90°, (0°, 90°, 180°, 270°). Record readings on the reverse dial indicator worksheet.
6. Determine if the initial readings are good.
 - a. Add top (T) and bottom (B) together for both planes and the two side readings (S) together for both planes.
 - b. Take the difference of the two readings. If the difference exceeds 0.002", there is something wrong with the readings. Inspect the set up and make any necessary adjustments.
7. Algebraically zero the side readings. Be consistent on which side you zero; it is usually easier to zero the 90° side.
8. Make dial indicator sag correction on worksheet.
 - a. Dial indicator sag only effects vertical readings. Since the dial indicator is going to read negative on the bottom, add the sag to the dial indicator reading on the bottom.
9. Divide all corrected readings by two because they are TIR readings taken on the outside of a circle.
 - a. Remember, when the dial indicator reads positive, the probe is being pushed in. When it reads negative, the probe is extended.
10. Determine shim change.
 - a. Lay out the machine dimensions on the plotting board transparency.
 1. Once the scale is determined you must be consistent and use only that particular scale.
 - b. Referring to our example, you must use the "C" scale on the bottom horizontal axis. The bottom horizontal axis represents the physical dimensions of the machine.
- c. The left vertical axis represents the misalignment/shim correction scale.
- d. Locate , S, A, IB, OB,
 1. S is located where the vertical and horizontal axis of the overlay intersect. S represents the location of the stationary reference plane.
 2. A is marked on the horizontal axis and represents the location of the adjustable reference plane. In our example, it is marked at 7" on the C scale.
 3. B is marked on the horizontal axis and represents the location of the inboard foot of the adjustable machine. In our example it is marked at 15" on the C scale.
 4. OB is marked on the horizontal axis and represents the location of the outboard foot of the adjustable machine. In our example it is marked at 36" on the C scale.
 5. Mark reference on the plotting board transparent vertical scale.
11. Plot shim change for vertical correction first.
 - a. Transform worksheet data to the plotting board.
 1. Set S at 0.009" low mark based on the E vertical scale
 2. Set at 0.0035" high mark based on the E vertical scale
 - b. Draw vertical lines from the IB and OB locations on the red line to the horizontal zero line on the plotting board.
 - c. Count the vertical distances from the IB and OB marks to the horizontal zero line using the correct scale, in our case the E scale, these values are the shim changes at the inboard (IB) and outboard (OB) feet of the adjustable machine.
12. Make shim change.
13. Repeat Step 11 for horizontal correction.
14. Check alignment.
 - a. The machines should be aligned at this point; if not, repeat Steps 11 and 12.
15. Inspect final alignment and record all results.

NOTES

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