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.

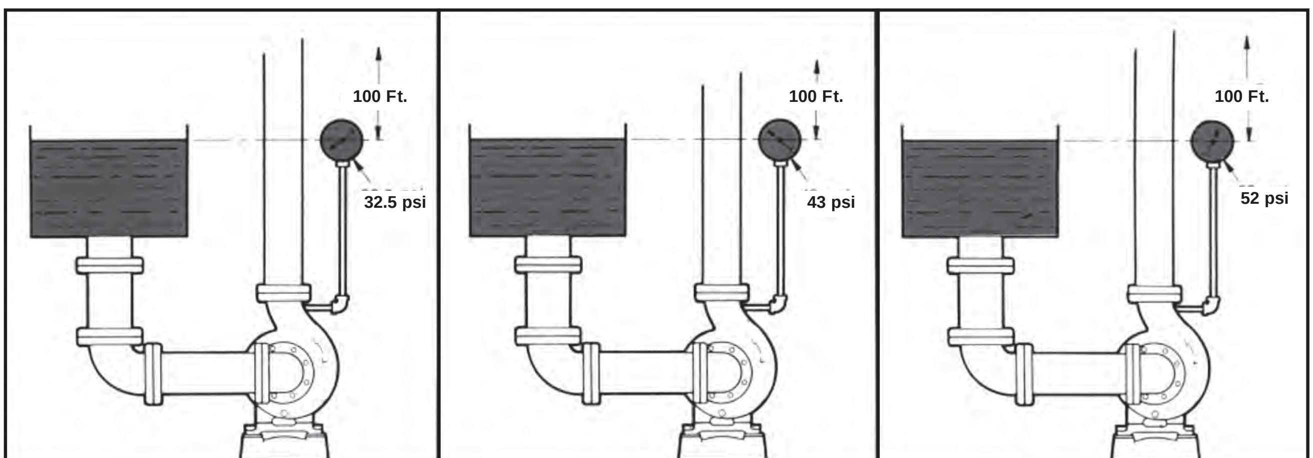
v = Velocity at periphery of impeller in feet per sec.

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:

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

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.



Gasoline, Sp. Gr. = 0.75	Water, Sp. Gr. = 1.0	Brine, Sp. Gr. = 1.2
$\text{Discharge Pressure} = \frac{100' \times 0.75}{2.31} = 32.5 \text{ PSI}$	$\text{Discharge Pressure} = \frac{100' \times 1.0}{2.31} = 43 \text{ PSI}$	$\text{Discharge Pressure} = \frac{100' \times 1.2}{2.31} = 52 \text{ PSI}$

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 center line 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. 2-a & 2-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:

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

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.

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

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.

TDH = h_d + h_s (with a suction lift)
TDH = h_d – h_s (with a 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, or flow rate, and the pipe sze and fluid velocity. This relationship is as follows:

$$\mathbf{Q = V \times (ID^2) \times 2.445}$$

Where	ID = inside diameter of the pipe, inches
	V = Velocity of the liquid, feet per second
	Q = capacity, gallons per minute

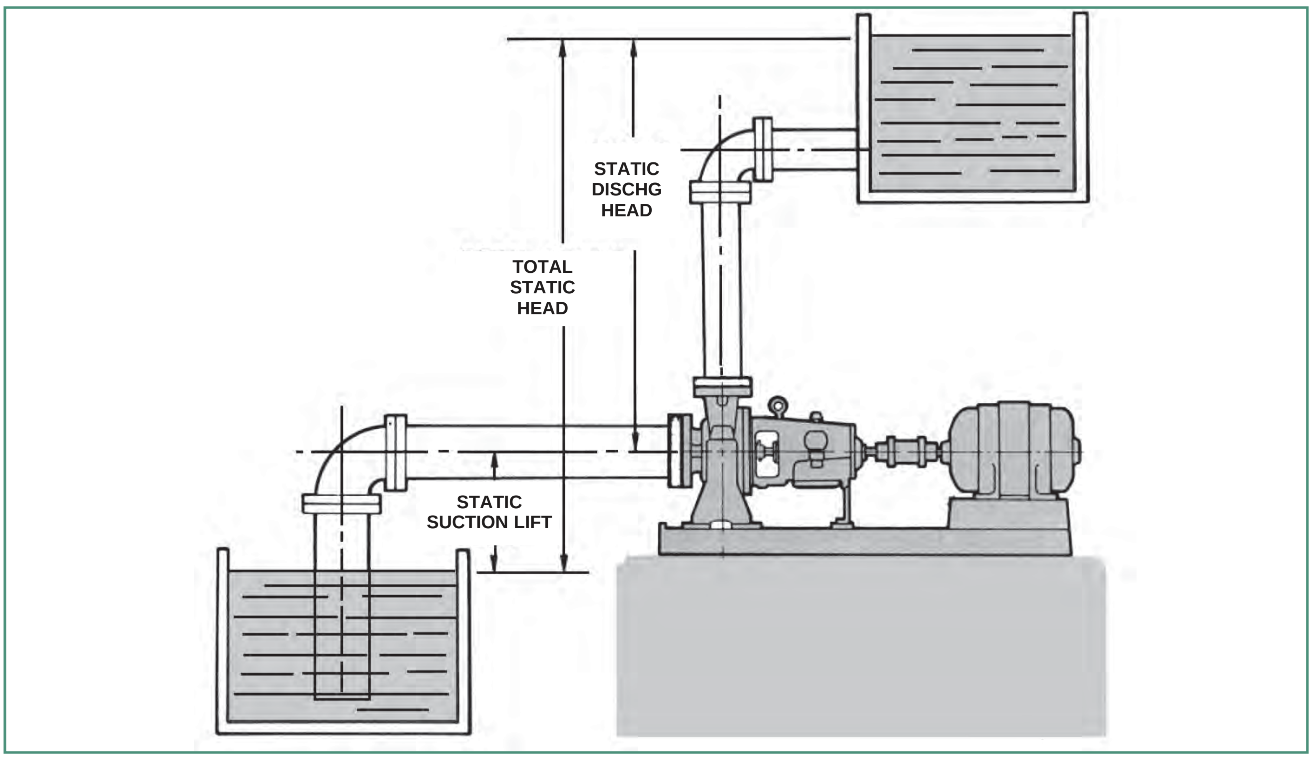


Fig. 2-a Suction Lift – Showing Static Heads in a Pumping System Where the Pump is Located Above the Suction Tank. (Static Suction Head)

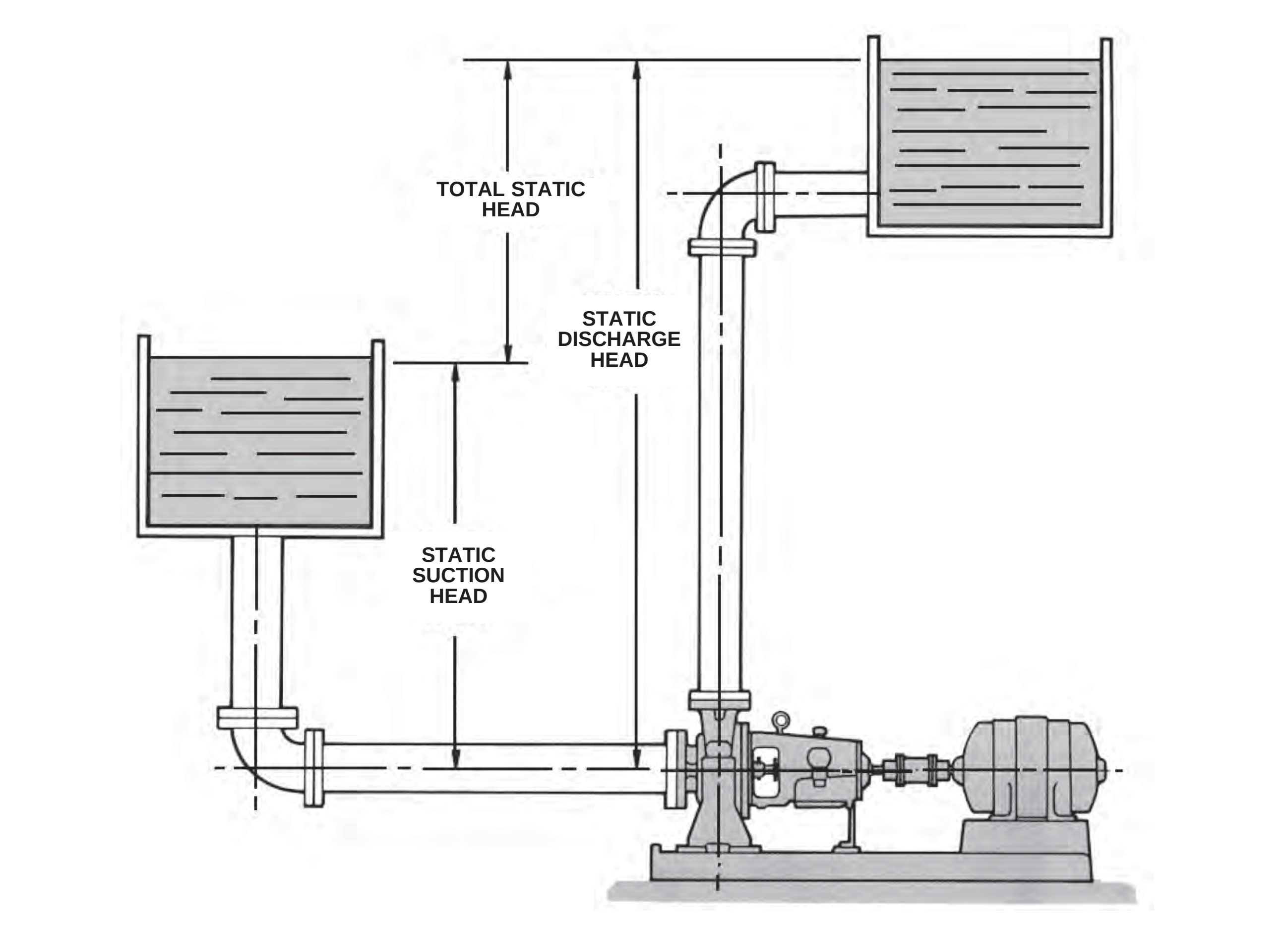


Fig. 2-b Suction Head –
Showing Static Heads in a Pumping System Where the Pump is Located Below the Suction Tank. (Static Suction Head)

TECH-A-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.

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

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

The constant 3960 is obtained by dividing the number or 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.

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

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.

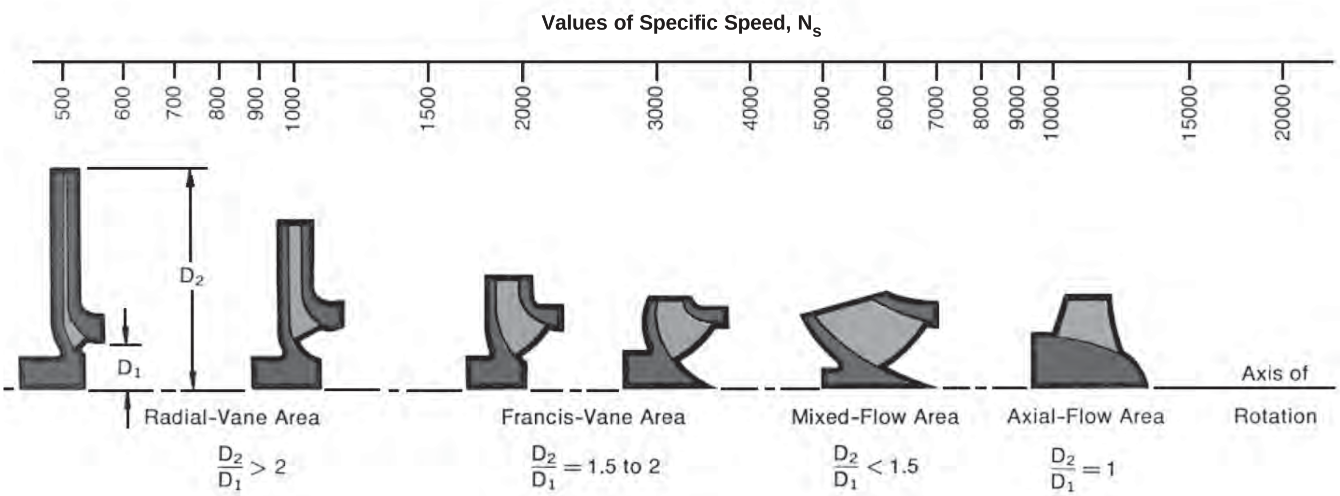


Fig. 3 Impeller Design vs Specific Speed

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.

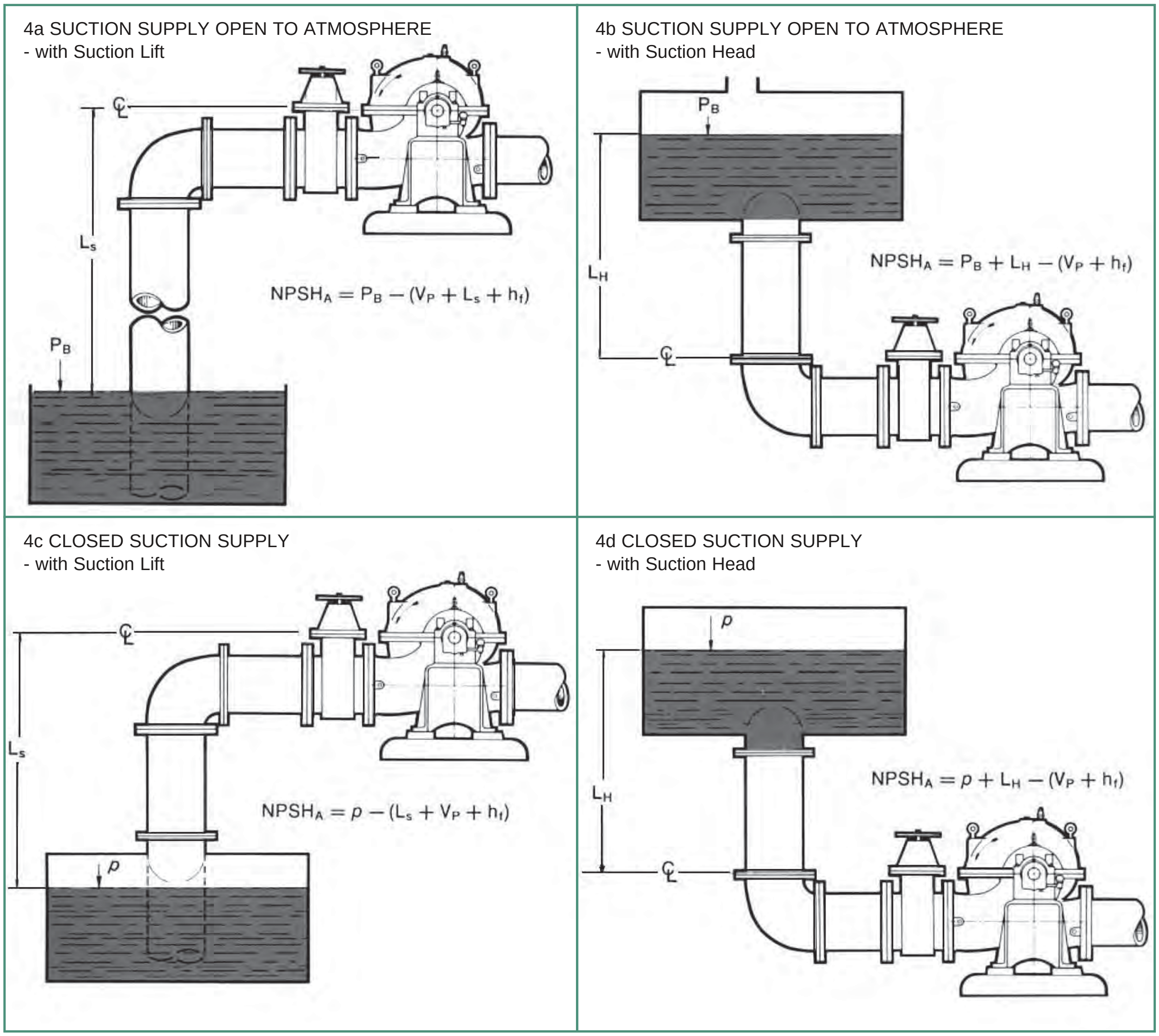
The pressure which a liquid exerts on its surroundings is dependent upon its temperature. This pressure, called vapor pressure, is a unique characteristic of every fluid and increases with increasing temperature. When the vapor pressure within the fluid reaches the pressure of the surrounding medium, 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:

$$NPSH_A = P_B - V_p \pm Gr + h_v$$

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:

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

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.

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

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

$$NPSH_R^{3/4} = 17.7$$

$$NPSH_R = 46 \text{ ft.}$$

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?

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

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

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

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.

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

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

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

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:

Suction Energy = D_e x N x S x Sg

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

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

High Suction Energy starts at 160 x 10^6 for end suction pumps and 120 x 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 x 6" = 5.4"
Suction Energy = D_e x N x S x Sg
= 5.4 x 3550 x 9,000 x 1.0
= 173 x 10^6

Since 173 x 10^6 > 160 x 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.

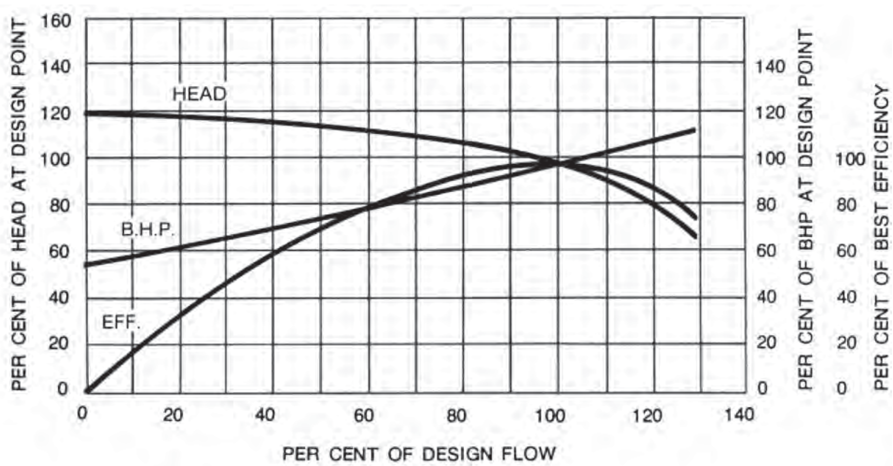


Fig. 5 Radial Flow Pump

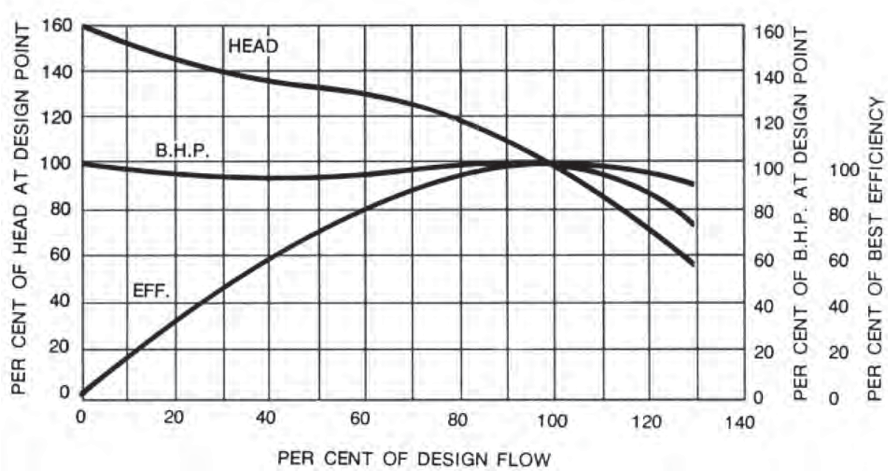


Fig. 6 Mixed Flow Pump

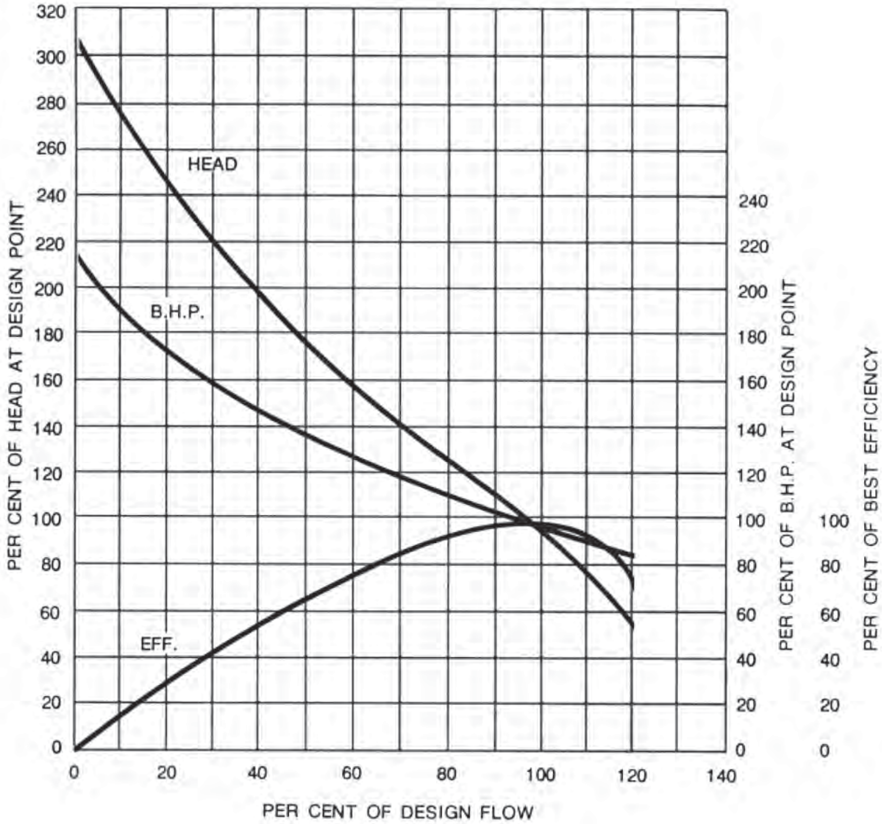


Fig. 7 Axial Flow Pump

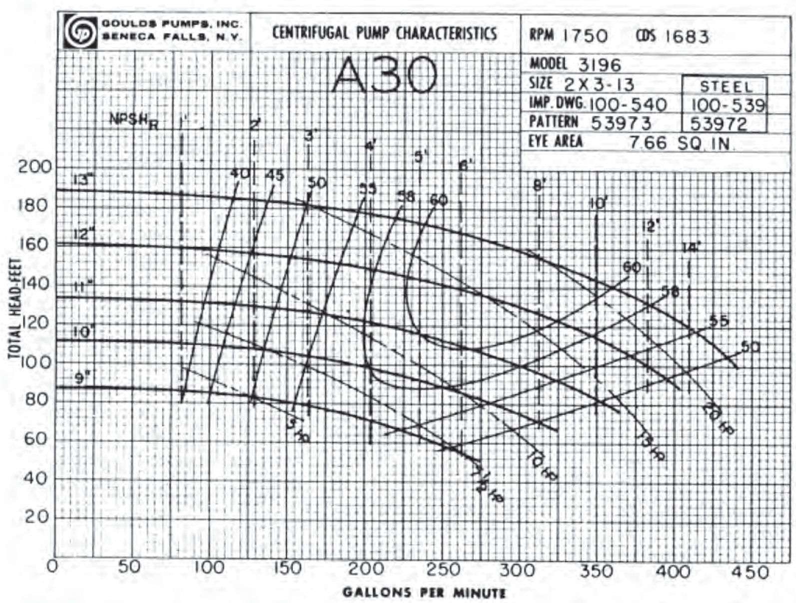


Fig. 8 Composite Performance Curve

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" diameter 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.

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

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

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

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.

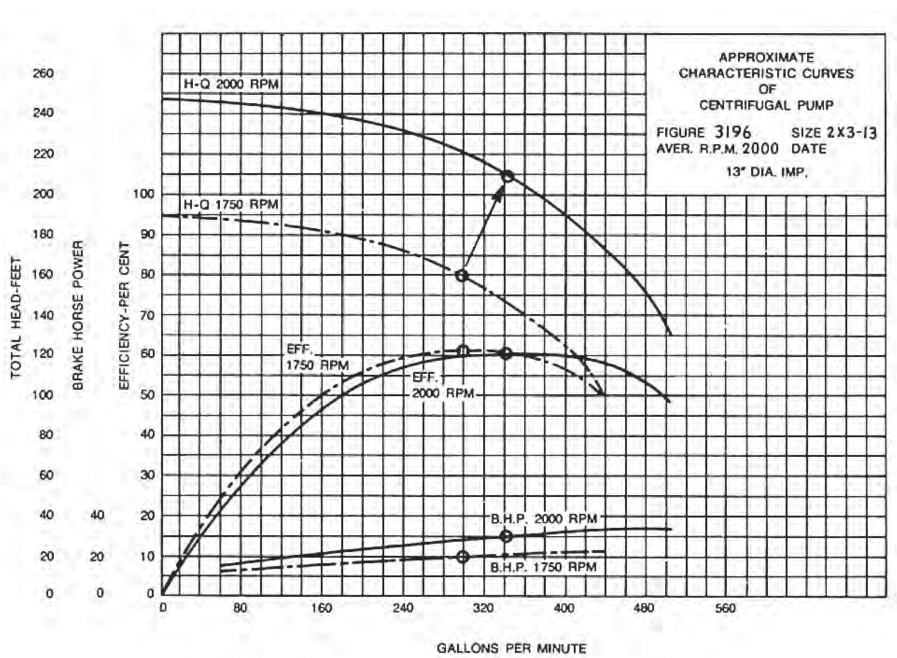
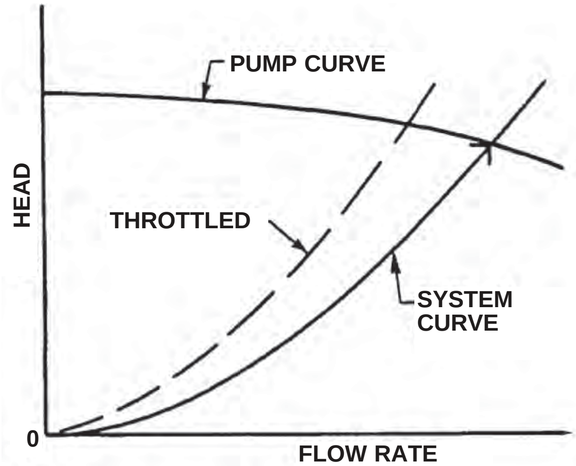


Fig. 9

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.

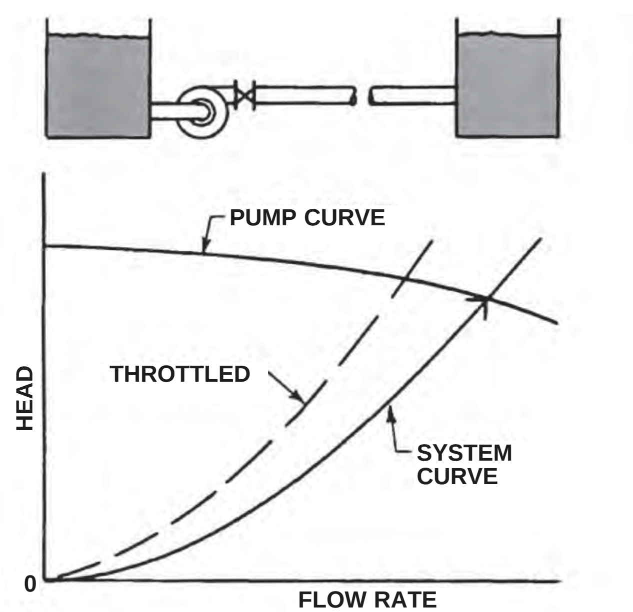


Fig. 1 No Static Head - All Friction

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.

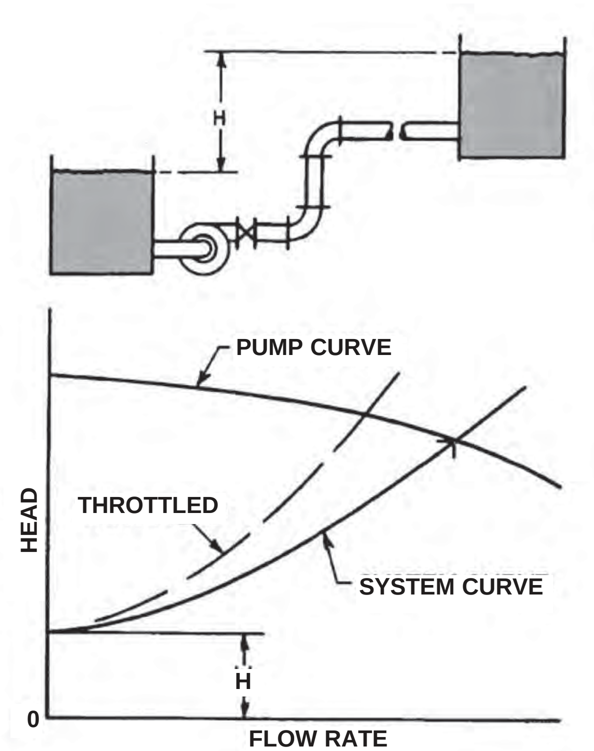


Fig. 2 Positive Suction Head

* 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.

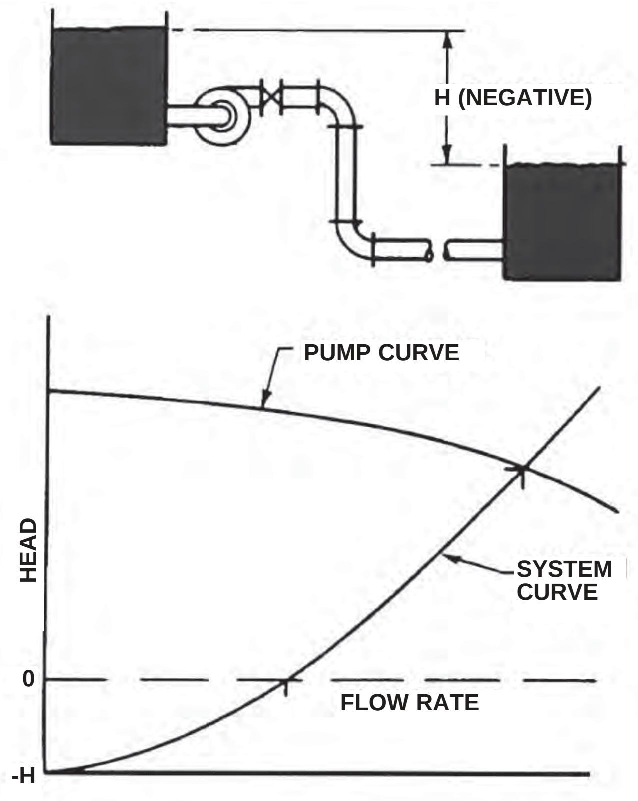


Fig. 3 Negative (Gravity) Head

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.

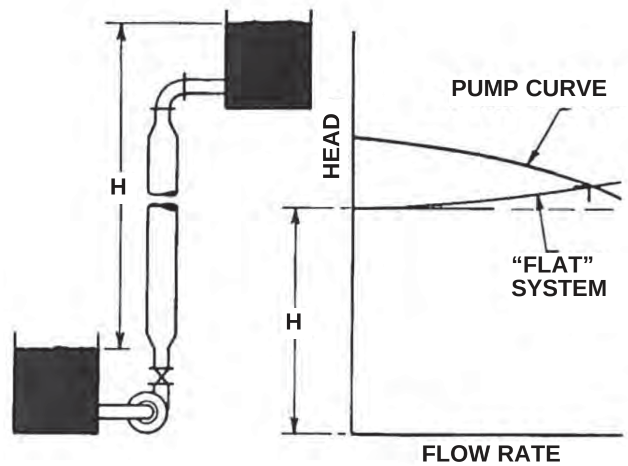


Fig. 4 Mostly Lift - Little Friction Head

TECH-A-10 Basic Formulas and Symbols

Formulas

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

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

$$GPM = 449 \times CFS$$

$$GPM = 0.7 \times BBL/Hr.$$

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

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

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

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

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

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

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

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

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

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

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

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

DEG. C = (*DEG. F* - 32) x 5 / 9

DEG. F = (*DEG. C* x 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/sec^2 (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