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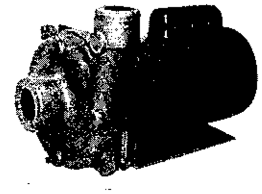
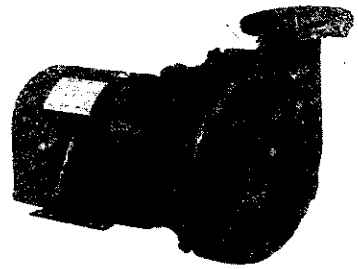
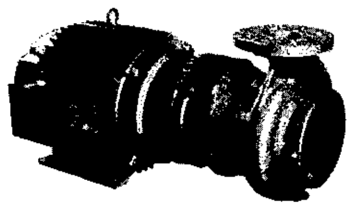
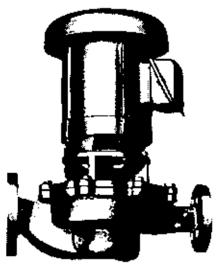
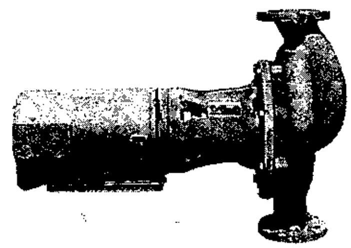


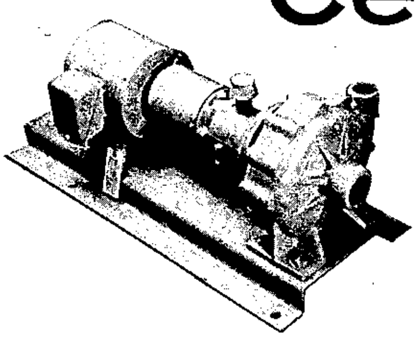


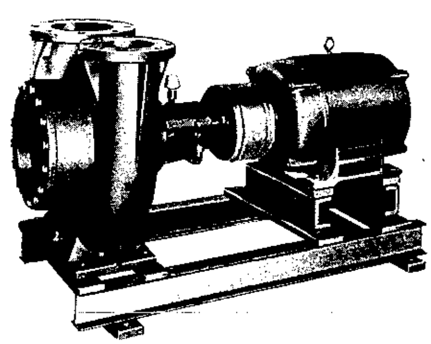


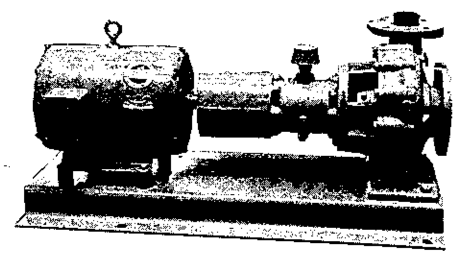




A Guide to Selecting Centrifugal Pumps







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There is no lack of variety in the kinds and sizes of centrifugal pumps available today — either from stock or through special design services. Still misapplications occur. Systems fail to operate as expected, and as required.

Getting proper operation from a piping system, whether for year 'round air conditioning or for process work, requires first of all that all the components be properly selected as to size, material, and performance characteristics. Correct pump selection is critical. Here are the things the engineer must consider.

A Guide to Selecting Centrifugal Pumps

THE CENTRIFUGAL PUMP — there is probably no other piece of mechanical equipment so deceptively simple in construction and yet so complicated in application. Further, perhaps no other piece of equipment is made in so many styles and designs, a number of which are the result of evolutionary experience which has caused a departure from theoretical design rules.

This is a general guide to centrifugal pump selection, in which we shall set forth the ground rules of determining needed pump characteristics and equating these requirements with the kinds of equipment which are, or can be made, available. Therefore, performance figures which are used to illustrate certain points must be accepted as generalities, subject to considerable variation at the discretion of the designer to fit particular applications.

Cost Can Be Controlled

Pump manufacturers have their standard lines of stock centrifugal pumps, and in addition, can supply thousands of special designs to fit unusual applications. Unfortunately such special designs must be handled from the beginning on a, say, one or two unit basis. Since the economics of mass production are therefore lost, costs of such units will increase accordingly.

Occasionally, design conditions may be so severe that even if special designs and materials are used, pump life will be measured in months. At the other extreme, a well built standard pump may do a particular job for years with little or no attention. In any event, useful pump life will be directly related to the care with which any pump is selected to suit design conditions. And obviously it behooves the engineer to select a pump by means of a set, comprehensive plan which will consider all the variables.

This article will deal with standard applications in the heating, piping, and air conditioning field, and covers most ordinary uses of centrifugal pumps.

Assuming that the capacity and head requirements have been determined, the next step is actual pump selection. What is involved?

Determining Pump Rotation

When a pump is to be installed as a replacement unit in an already existing installation, or when it is

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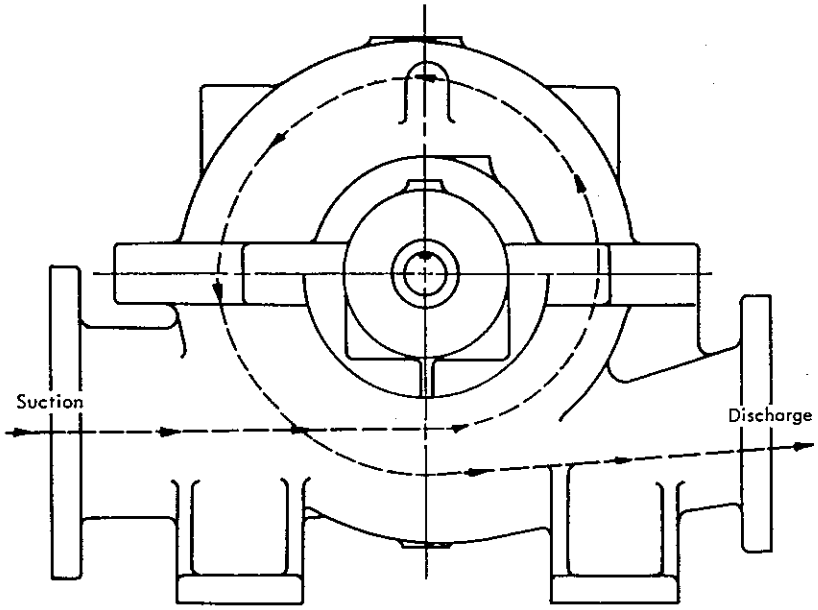


1 PUMP ROTATION is classified as either right or left hand. At left, view is *from* driver *to* pump, and particle moving through pump would follow broken line, counterclockwise around shaft. Rotation is therefore "left hand"

to go into a new installation where the piping layout has already been decided, the locations of suction and discharge are fixed. With a double suction pump (and virtually all other centrifugal pump designs) this determines the direction of rotation.

Pump rotation is normally classified as *when looking from the driver toward the pump.* From this vantage point, clockwise rotation is called right hand, and counter-clockwise, left hand.

How Does Fluid Flow?

Fig. 1 shows an end view of a typical, single stage, double suction pump, with the suction connection on the left and the discharge on the right.

Determination of rotation is made by following the path of an imaginary particle of liquid through the pump. It will follow the dotted line as shown.

This flow particle, in traversing the suction and discharge volutes, will circle the shaft in a counterclockwise direction. If this view is from the driver to the pump, then the rotation is left hand. Careful consideration of the information given regarding pump rotation in design data may frequently uncover errors and avoid expensive machining mistakes.

Approximating Pump Size

The required capacity determines the size of a pump — that is, whether a 2, 6, or 20 in. unit is needed. Pump size is designated by the diameter of the *discharge* opening. Usually, but not always, the suction opening is one or two sizes larger.

Regardless of what else the pump is called upon to do, it must have flow passages large enough to pass the required volume of liquid at a satisfactory velocity for the size of the pump and conditions under consideration. Therefore, capacity determines the size.

Allowable discharge velocity is greater for smaller pumps than for large ones.

Suction velocity is of particular importance in the case of large pumps which develop a low head and where there is no static head on the suction side. It is also important in condensate pump applications where the water is hot and there is little static suction head. Too great a suction velocity in such cases will result in cavitation.

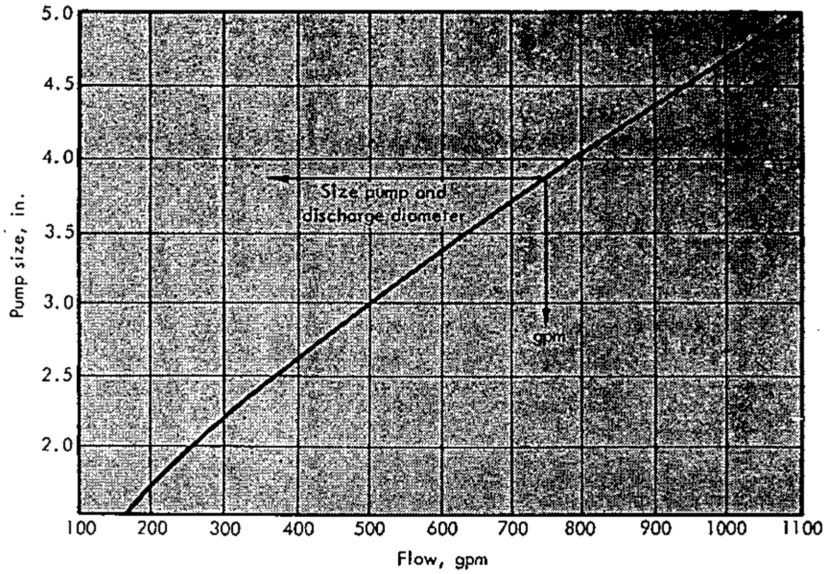
Curves Show General Range

Using some typical velocities, the curve of Fig. 2 has been plotted for small pumps, and that of Fig. 3 for larger units to show the approximate capacity which might be expected from a specific size. These curves do not rigidly fix pump size. A specific job might permit a smaller size or dictate a larger. These curves merely serve to keep us "in the ball park." One convenient point of reference is that 1000 gpm will require, roughly, a 5 in. pump.

Naturally, in using Figs. 2 or 3, one should select the nearest larger pump size available. Note also that there is a strong trend away from the use of certain pipe sizes, such as 1-¼, 3-½, 5, 7, and 9 in. lines.

The curves of Figs. 2 and 3 are based on discharge velocities, and, as noted, the discharge size normally fixes the pump size. There are, though, instances in which the liquid is highly volatile or of a high temperature, and if either or both of these conditions are coupled with a lack of head on the suction side, suction conditions may be so critical that a larger pump, or one with an oversize suction inlet and impeller eye, will be needed.

If the liquid to be pumped is particularly viscous, the pump will have to be selected with a larger capacity (based on water) to deliver the required amount of viscous fluid. Standards of the Hydraulic



2 PUMP SIZE is designated according to diameter of discharge opening. Pump capacity (flow) will determine what size passages will be needed. This graph shows *approximate* capacities which might be expected from some small pump sizes

Institute include correction factors to be applied to the water-capacity values in such cases.

Wheel Imparts Force

In a centrifugal pump, the only component which actually imparts energy to the liquid is the impeller. This energy, at the point of discharge of the impeller, is in the form of both kinetic and potential. Kinetic energy is represented by the velocity of discharge from the impeller and the potential energy by the static pressure.

Total Head Is "Constant"

Later, in passing through the casing volute or diffuser vanes, some of the kinetic energy will be converted to additional potential energy, but the total energy or total *dynamic head* is the "same" as when leaving the impeller. More precisely, the dynamic head will be slightly less because of some frictional and turbulence losses which will occur in the volute or diffuser.

How Pump Head Is Determined

The basic formula for pump head is derived from the velocity of a body (of any weight) falling from a specific height, as follows:

$$V = gt$$

and

$$H = gt^2/2$$

from which

$$V^2 = 2gH$$

or

$$V = \sqrt{2gH}$$

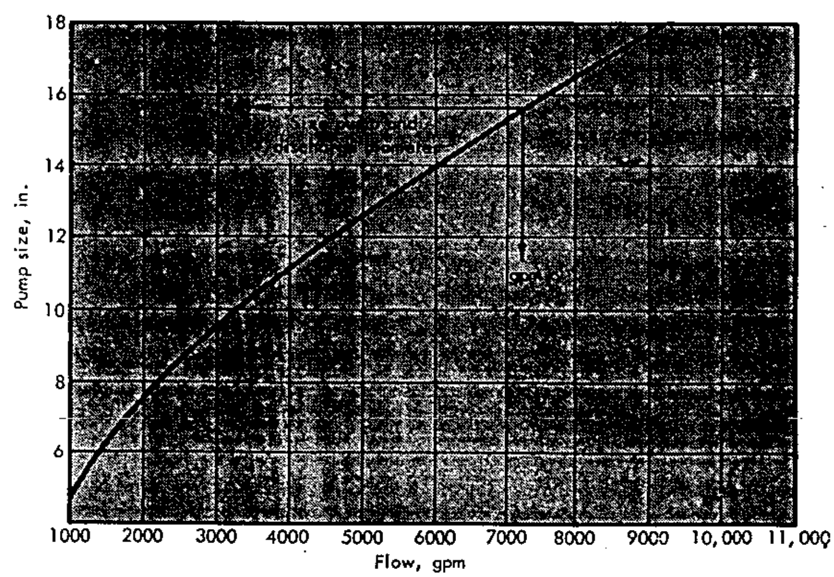
or

$$H = V^2/2g$$

where

t = time, seconds
V = final velocity after t seconds
g = acceleration due to gravity, 32.2 ft per sec^2
H = height in ft

These formulae are independent of the specific gravity of the liquid.



3 LARGE PUMP capacities will vary according to discharge opening diameters about as shown here. Designer would select next larger standard size to handle required flow

selecting centrifugal pumps

Since the energy imparted to a unit weight of liquid is independent of the weight of the liquid, then the pump will develop the same head in ft of the liquid handled, regardless of the specific gravity of the liquid. This is important. It also means that the same head-capacity curve, in ft, is valid regardless of the liquid handled. (Some exception will be made to this only if the liquid is quite viscous, *i.e.*, not "free flowing.")

Use "Ft of Liquid"

Head is expressed in various ways — ft of liquid, psi, or in. Hg. Other units should always be converted into ft of liquid handled.

The most common conversion is from psi into ft of liquid. The following are commonly used equations for such conversions:

$$\text{Head, ft} = \text{psi} \times 144/\text{density, lb per cu ft}$$

Water at 68 F weighs 62.4 lb per cu ft, so

$$\begin{aligned}\text{Head, ft} &= \text{psi} \times 144/62.4 \\ &= 2.31 \times \text{psi}\end{aligned}$$

From this it can be seen that a column of water 2.31 ft or 27.7 in. in height will exert a pressure of 1 psi.

This conversion equation may also be rewritten as

$$\text{Head, ft} = \text{Pressure, psig} \times 2.31/\text{specific gravity, liquid pumped}$$

In a similar manner, if the head is specified in in. Hg, it can be converted to ft of liquid by

$$\text{Head, ft} = \text{In. Hg} \times 1.134/\text{specific gravity, liquid pumped}$$

The effect of specific gravity is seen as follows, where it is assumed that in each instance a discharge pressure of 100 psig is required:

Substance	Specific gravity	Head required, ft
Water	1.0	231
Gasoline	0.75	308
Brine	1.20	193
Sulfur (liquid)	1.79	129

Since impeller diameter and peripheral speed are a function of

$$V = \sqrt{2gH}$$

the head generated at constant speed will vary according to the square of the impeller diameter. Impellers to deliver 100 psig of water, gasoline, brine, and liquid sulfur would be sized, then, for diameters to deliver heads of 231, 308, 193, and 129 ft, respectively.

How Many Stages?

Since head developed is related to impeller tip speed, and therefore also to the combination of impeller diameter and rpm, the next step is the determination of how many stages are required to do a particular job.

There will be no general answer to staging design, applicable to all conditions. One stage, with one large-diameter impeller, will not be as efficient as two or three stages with smaller impellers. This is because friction loss of the impeller discs rotating in the liquid increases according to the fifth power of the impeller diameter, while head increases only according to the square, or second power, of the diameter. Thus friction losses will rise at a much faster rate than will the head if, by increasing either impeller or tip speed, we try to get too much head from a single stage. This means that overall pump efficiency will decrease and the problem is resolved into another variation of the old struggle of low initial cost *vs* high efficiency.

Other Single-Stage Hazards

Certain other difficulties occur with too large an impeller. The pump may be noisy and vibrate. The impeller may be too narrow at the discharge for good castings to be made, and material strength may become a factor. It becomes difficult to clean the passages between vanes so that casting fins and bumps may be left to disturb the fluid flow.

The question of staging will be resolved on the basis of what types of pumps are available to suit a particular application. No one pump, and no generality on the number of stages needed, will cover every application.

When Suction Must Be Considered

The speed at which a pump can be run is frequently determined, or limited, by suction conditions. If such conditions dictate a low speed, it becomes impracticable to develop a high head per stage.

To avoid tying a multistage pump down to a slow speed because of suction conditions — thereby increasing the cost and number of stages required — a booster pump ahead of and in series with the main pump is often used. A booster pump operating at low speed and low liquid velocity at the impeller eye can function with a very low NPSH (Net Positive Suction Head — to be discussed later). The discharge from the booster to the main pump suction will then be at a pressure high enough to eliminate cavitation troubles resulting from too low an NPSH.

The approximate head, H, in ft, which an impeller of outside diameter D_2 in. and corresponding peripheral tip speed U_2 (fps) will develop at a particular speed N (rpm), is obtained from the relationship

$$D_2 = 1840\ \phi\sqrt{H}/N$$

where ϕ is a coefficient related to the particular pump design. It varies both above and below a value of 1.0. If we use an average value of 1.0, then the anticipated head available per stage for different impeller diameters rotating at standard electric motor speeds may be plotted as shown in Fig. 4. This should be used as an approximate guide only.

The equation above can also be written

$$U_2 = \phi\ (\sqrt{2g})\ (\sqrt{H})$$

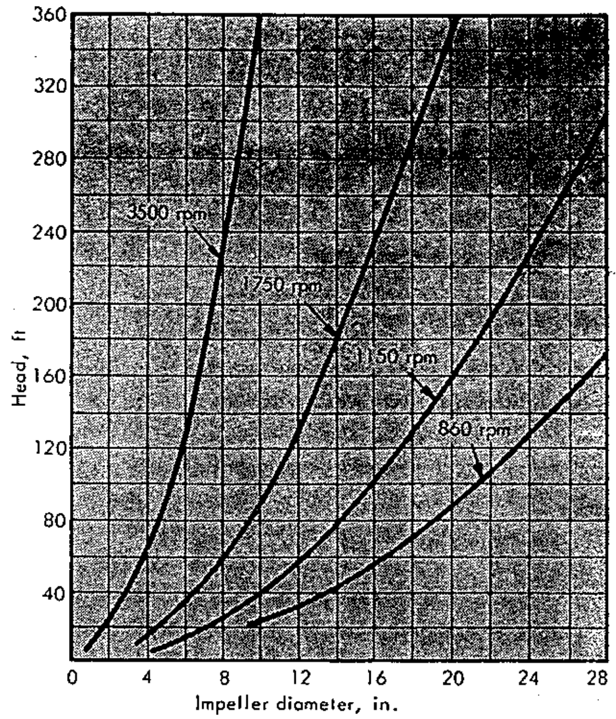
Since standard commercial pumps use various iron or bronze cast impellers, tip speeds are generally held to a moderate figure. If we assume that where suction conditions are no problem, the tip speed is held to 150 fps, we can use the above equation to find that the corresponding head is 349 ft. This is illustrated in the plot of tip speed *vs* head in Fig. 5.

For certain high speed applications, the impeller tip speed may be considerably greater than 150 fps. 220 fps would not be excessive. The latter value would result in a head of about 750 ft per stage.

Again, these limitations must be used with caution, as they are not dependent upon some single, clearcut consideration.

Power Factors

Water horsepower is the theoretical rate of work at which a pump must perform to deliver a constant quantity each interval of time against a stated head.



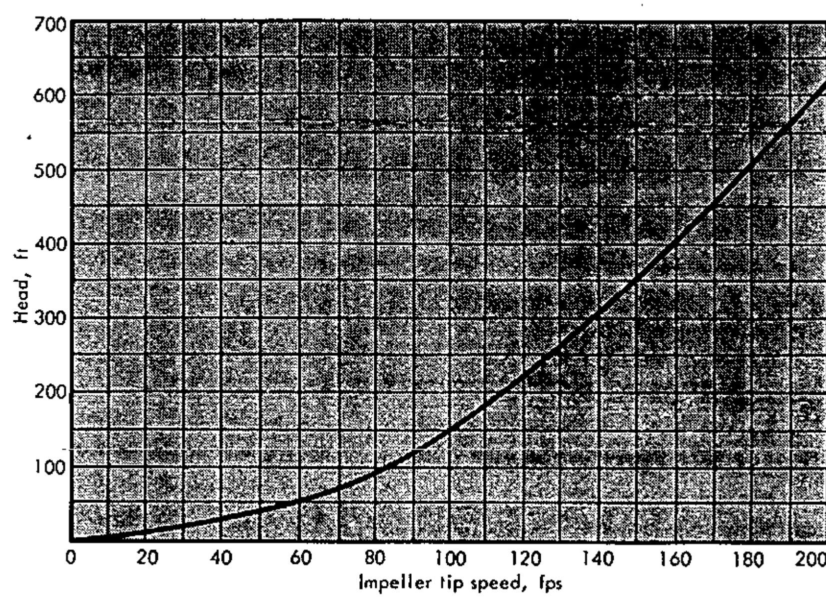
4 HEAD AVAILABLE per stage for different impeller diameters rotating at standard electric motor speeds

The quantity of liquid is usually designated in gpm, but could just as well be in lb per second or barrels per day. The term "water hp" is used, regardless of the actual liquid being handled, to express the power required at 100 percent efficiency.

If a weight, W, were to fall from a height, H, equal to 100 ft, then it would attain a velocity of

$$V = \sqrt{2gH} = 80.3 \text{ fps}$$

To return weight W to its original height would require that it be projected upward with a velocity of 80.3 fps. To do this, an amount of work equal to $W \times H$ would be performed. A pump which per-



5 PUMP HEAD as function of tip speed plots as shown here. Tip speeds are usually kept to moderate values, but may go to 220 fps or higher for special applications formed this work would be said to deliver 100 ft of head.

If this pump would deliver 1000 gpm of cool water to the 100 ft height, then the work done would equal

$$(1000)(8.33)(100) = 833{,}000 \text{ ft-lb per min}$$

where

$$8.33 = \text{weight of 1 gal water with a specific gravity of 1.0}$$

If the liquid were gasoline, then the work done would equal 1000 X 6.25 X 100 = 625,000 ft lb per min; where 6.25 is the weight of 1 gal of gasoline corresponding to a specific gravity of 0.75.

If the liquid were molten sulfur (14.9 lb per gal, specific gravity of 1.79) the work done would be 1000 X 14.9 X 100 = 1,149,000 ft lb per min.

Specific Gravity Sets Power Demand

It can be seen that to pump the same quantities of gasoline and water against the same heads, the gasoline would require only 75 percent as much work as would water. Molten sulfur would require 79 percent *more* work than water. This is to say that the work done in pumping a liquid is proportional to the specific gravity of a liquid, and, therefore, the specific gravity of the liquid will determine the power required to drive the pump (the work done).

How Liquid Weight Is Introduced

If, in the water hp equation, W represents the weight in lb of liquid delivered per unit time of 1 min, then the equation may be rewritten to express the theoretical, or water hp required by the pump as follows:

$$\textit{Water hp} = WH/33{,}000$$

where

$$33{,}000 = \text{ft-lb per min per hp}$$

To deliver 1000 gpm against a 100 ft head, the three liquids used as our examples would require the following power:

Water — 25.2 hp
Gasoline — 18.9 hp
Molten sulfur — 45.1 hp

In these examples, it is seen that, for water, there is a constant:

$$8.33/33{,}000 = 1/3960$$

so that, for water of specific gravity 1.0, the equation can be written

$$\text{Water hp} = (\text{Gpm})(\text{head, ft})/3960$$

Provide For Specific Gravity

The unit weight of gasoline is 6.25 lb per gal, and its specific gravity equals 6.25/8.33 or 0.75. Its unit weight can be represented by (Weight of water) (Specific gravity). Therefore, to include liquids other than water, the water hp equation can be rewritten to provide allowance for specific gravity:

$$\text{Water hp} = (\text{Gpm})(\text{Head, ft})(\text{Specific gravity})/3960$$

This shows again that the water hp required by a pump is directly proportional to the specific gravity of the liquid pumped.

TABLE 1 — CONVERSION FACTORS listed here can be used to arrive at standard units for centrifugal pump sizing

	Equals
1 barrel	42 U.S. gal
1000 barrel per day	29.2 gpm
1 million gal per day	694.4 gpm
1 cu ft liquid	7.48 gal
1 cu ft per second	448.8 gpm
1 cu ft per min	0.1247 gal per sec
1 gal	0.1337 cu ft
	3785.0 cc
	231.0 cu in.
	0.003785 cu meters
	0.004951 cu yd
	3.785 liters
	58,310.0 grains
1 U.S. gal	8.3453 lb (at 39 F)
	0.8327 Imperial gal
1 Imperial gal	1.209 U.S. gal
	277.0 cu in.
1 gpm	0.002228 cu ft per sec
	0.06308 liters per sec
	8.0208 cu ft per hr
	6.0086 tons water per day
	1440.0 gal per day
1 ton water per day	0.1664 gpm
1 cu meter water	264.2 gal
1 cu yd water	202.0 gal
1 cu yd per min	3.367 gal per sec
1 lb water	0.1198 gal
For heads and pressures	
1 ft water	0.8826 in. Hg
	62.43 lb per sq ft
	0.4335 psi
1 psi	27.71 in. water at 62 F
	2.3077 ft water at 62 F
	2.036 in. Hg
1 atmosphere	29.921 in. Hg at 32 F
	760.0 mm Hg
	33.9 ft water
1 in. Hg	1.134 ft Water
1 in. water at 62 F	0.735 in. Hg at 62 F

When "PSI" Is Used

Sometimes the head is given in terms of psi. Previously we said that a pump will develop the same head in ft of liquid handled, regardless of its specific gravity:

$$\begin{aligned}\text{Head, ft} &= (\text{psi})(144)/\text{Density, lb per cu ft}\\ &= (\text{psi})(2.31)/\text{Specific gravity}\end{aligned}$$

Substituting the value of head in ft in the water hp equation:

$$\text{Water hp} = [(\text{Gpm})\ (\text{psi})\ (2.31)/\text{Specific gravity}]\ [\text{Specific gravity}/3960] = (\text{Gpm})(\text{psi})/1714$$

The important point about this equation is that it shows, if the head is stated in terms of pressure (psi) that the water hp is independent of the specific gravity of the liquid.

Next: Pump Efficiency

Since, for all machines, the efficiency is the ratio of the power required to do the work at 100 percent efficiency to that actually required because of inherent losses, we can say

$$\text{Pump efficiency} = \text{Water hp}/\text{Brake hp}$$

Therefore

$$\text{Brake hp} = (\text{Gpm})(\text{head, ft})(\text{Specific gravity})/(3960)(\text{Efficiency}) = (\text{Gpm})(\text{psi})/(1714)(\text{Efficiency})$$

Certain constants relating to the stated flow of a liquid, which can be useful in making the conversions indicated above, are itemized in Table 1.

The Effect of Viscosity

In terms of a water-pumping operation, it is necessary to increase both the capacity and the head values when selecting a pump to handle viscous fluid.

It is necessary, then, that the hp required to drive the pump at the specified head and capacity of the viscous fluid is going to be greater than if it was water, even if they have the same specific gravities. Overlooking this factor has resulted in pump installations which failed miserably to deliver the required head and capacity, and in seriously overloaded drivers.

A correction factor (such as published by the Hydraulic Institute) must be inserted in the pump brake hp equation whenever viscous fluid is to be moved. The equation becomes

$$\text{Brake hp} = (\text{Gpm})(\text{Specific gravity})(\text{Head, ft})/(3960)(\text{Efficiency}_{\text{corr}})$$

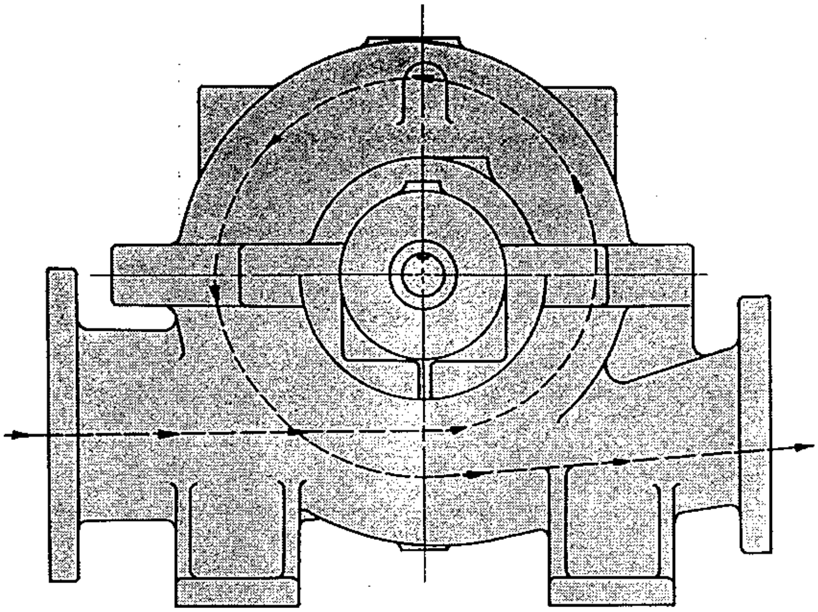
where

$$\text{Efficiency}_{\text{corr}} = (\text{Efficiency}_{\text{water}})(\text{Correction factor})$$

An important item to watch in this equation is that the capacity and head used are those for the viscous fluid, *not* the corrected water values. The corrected water values *are* used to select the pump size. +

Succeeding articles will investigate pump performance curves, the effects of speed changes and viscosity on those curves, and the subject of Net Positive Suction Head.

Cover photo of pump installation at Southdale shopping center, Minneapolis, courtesy Minneapolis-Honeywell Regulator Co.



How Performance Curves Help You Select Centrifugal Pumps

By HAROLD W. WOODHOUSE
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Mechanical Engineering Department
Stevens Institute of Technology

Knowing exactly the meanings of the various performance curves which illustrate the phenomena of centrifugal pump operation will aid the engineer in selecting and using the proper pump for his particular job.

Mr. Woodhouse continues the series begun last month[1] aimed at a better understanding of pumps and systems – and a decrease in the number of disappointing installations.

[1] *A Guide to Selecting Centrifugal Pumps*, Heating, Piping & Air Conditioning, October, 1961

A TYPICAL centrifugal pump performance curve (Fig. 1) would show

1) The head-capacity curve,
2) The brake horsepower curve,
3) The efficiency curve, and
4) The net positive suction head curve.

Let us briefly consider each of these.

Head-Capacity Curve

This shows the head which the pump will develop at any capacity. An important point to remember is that the pump *must* operate somewhere on this curve. For a specified speed, it cannot operate at any point above or below the curve without some physical change to the pump itself.

Brake Horsepower Curve

Pump manufacturers run standard tests with water as the pumped fluid, and therefore their test values will be predicated on a specific gravity of 1.0, and if a pumping system is designed to handle, say, gasoline, of specific gravity 0.75, then a second brake horsepower curve for this specific gravity should be shown.

Fig. 1 shows one brake horsepower curve for a specific gravity of 1.0 and another for 0.75, and therefore the ratio of the ordinates y' and y will be

$$y'/y = 0.75$$

This conforms to the rule which states that the power required to drive the pump will be proportional to the specific gravity of the fluid which is being moved.

Efficiency Curve

This is calculated from values of head, pump capacity, water horsepower, and brake horsepower, and represents the ratio of power required to do the work at 100 percent efficiency to that actually required because of inherent losses. Some efficiency factor is applicable, of course, to all machines.

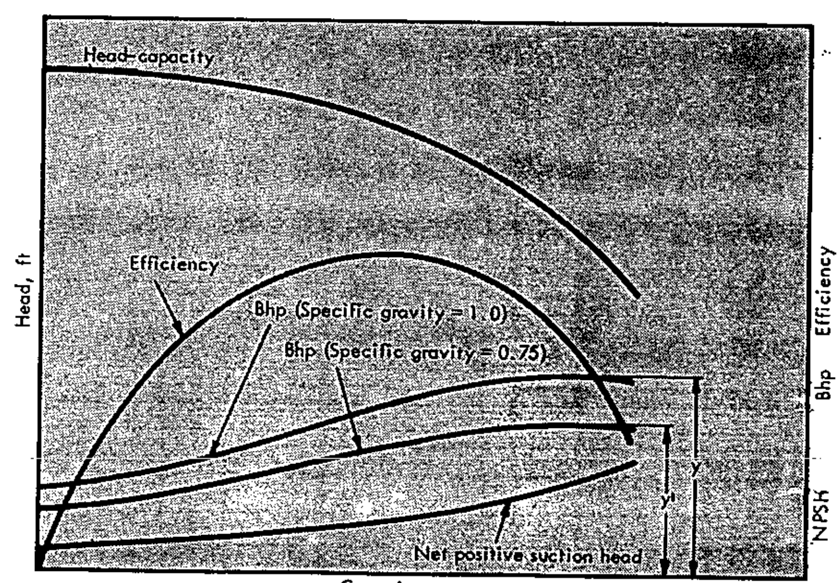
NPSH Curve

The net positive suction head curve may or may not be included in a set of general performance curves. Sometimes it is of no consequence, and sometimes it is all important. Generally speaking, for any one application, if the pumped liquid is cool and a gage *at the pump suction flange* shows a positive pressure, then the net positive suction head is of no consequence. This will be true for standard commercial speeds with about 3600 rpm as the maximum. There are certain special turbine-drive pumps in the 8000 to 12,500 rpm class where even a slight positive pressure at the pump suction might not guarantee freedom from cavitation which results from insufficient net positive suction head. The whole subject of NPSH will be discussed in more detail in the next article of this series.

What Is System Head?

As we said earlier, a centrifugal pump must operate somewhere on its head-capacity curve. Nowhere else. But where on this curve?

Assume the test curve of a particular pump to be in the form shown in Fig. 2. We know that when the pump is installed and running at full speed with the discharge valve fully closed (shutoff state) the head will be at the point marked "shutoff" on the curve, corresponding to zero flow. Then, as the discharge valve is opened, the head will move to some point along the curve which extends through *A* and *B*, un-



1 TYPICAL set of performance curves for centrifugal pump would include curves for head (capacity), efficiency, net positive suction head, and brake horsepower (plotted on basis of specific gravity of liquid pumped)

selecting centrifugal pumps

til, with the discharge valve fully opened, the head and capacity will stabilize at some point which we will call *B*. Now what causes the pump to operate at this particular point?

Reason: Total System Effect

The reason is that, in this instance, the *system head* and the *pump head* coincide when the capacity delivered by the pump is at point *B* on its performance curve.

The system head of a specific installation can be defined as the total dynamic head required to move a stated rate of flow through that system.

System head is the algebraic summation of the static head on the discharge side of the pump, *plus* the static lift on its suction side, or *minus* the static head plus all dynamic friction losses of all pipe and fittings on both suction and discharge sides, for the flow under consideration.

Curve Shows Head Coincidence

A typical system head curve is superimposed on Fig. 2. At zero flow, there will be no dynamic friction losses, so that the system head will be the static head plus the suction lift (or minus the suction head). This condition is at *S* ft of head, as shown in Fig. 2.

As flow increases through the system, the friction losses also increase, roughly according to the square of the flow, so that at some capacity point, *X*, the total system head requirement is *S* plus *PF* ft.

Assume that the pump being considered also develops a head equal to *S* plus *PF* and that this occurs at point *B* — which also is the location of capacity *X* on the system curve.

This Is Where Pump Will Operate

It now is apparent that the pump head-capacity curve and the system head curve intersect at the common point *B-X*, and that this is the only point common to both curves.

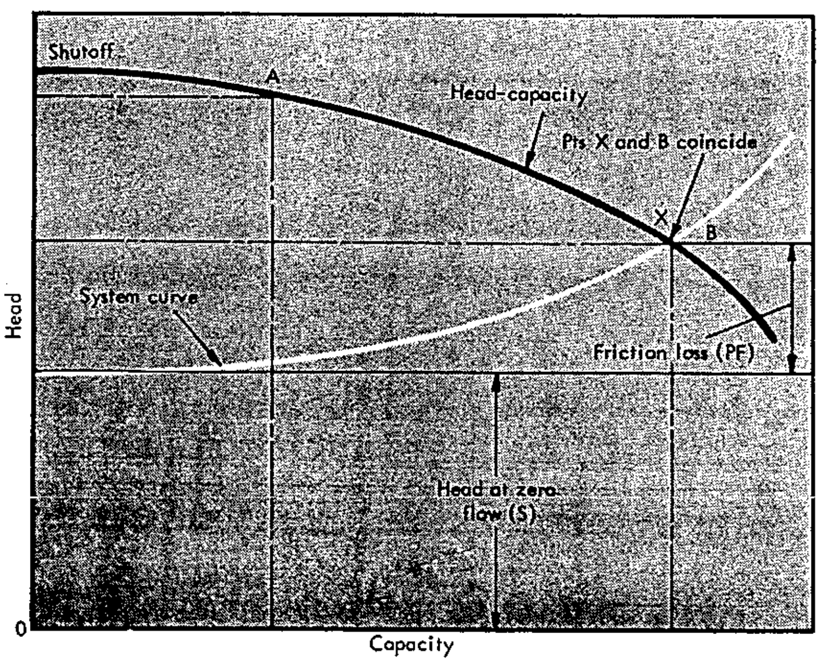
Without some physical change, this is the only point at which the pump will operate in this system, and the flow delivered will be in accordance with the location of the combined points *X* and *B*.

If the System Is Changed

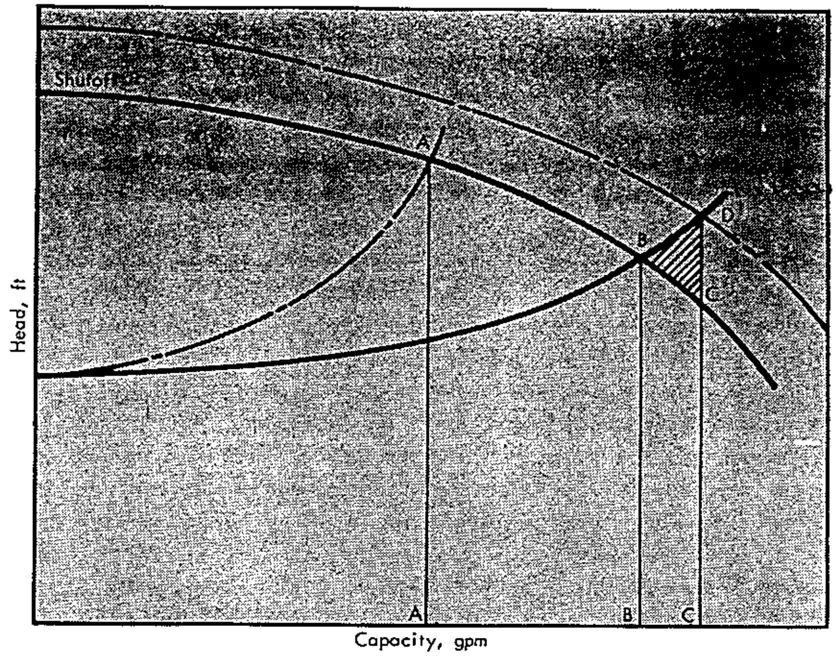
Suppose that with the same pump installation, a physical change is made to the system so that its frictional resistance is increased. Then the system head curve might intersect the pump head-capacity curve at *A*, as in Fig. 3, and the capacity at *A* would be the maximum flow the pump could deliver at this increased head. The change in frictional resistance of the system is

$$\text{Head}_A - \text{Head}_B$$

If we wish to reduce output capacity of the pump from *B* to *A* in Fig. 3, we will close the discharge valve until capacity declines to *A*. The effect of partially closing the valve is to increase artificially



2 TEST CURVE for particular pump includes point *B*, which represents full-open operation. System head curve (*white*) intersects pump curve at X (*B*), which is where pump will operate in this system



3 EFFECT of increasing system resistance against the pump of Fig. 2 is to move pump-system intersection back from *B* to *A*, which becomes new operating point. Increasing flow to *C* is impossible without decreasing system head or raising pump speed or changing impeller

the system head losses so that the system head curve passes through point *A*, by increasing the turbulence losses through the valve.

Can Flow Be Increased?

What if we'd like to deliver a capacity as represented by point *C* in Fig. 3? With the discharge valve fully open, we find that we can obtain the capacity corresponding to point *B*. What can be done about this?

While we can see from its test head-capacity curve that this pump, *at rated speed*, is capable of delivering capacity *C*, it can do so only against a resistance of head *C*. Extending the head system curve to capacity *C* we see that the head requirement increases to point *D*. There is, then, a deficiency of head put out by the pump against that required by the system of ($Head_D - Head_C$) ft. In fact, without a change in the system, the pump, at rated speed, could not operate anywhere in shaded area *BCD*.

Some Possible Solutions

Depending on the specific installation, the engineer may have several courses of action open which will permit operation at capacity *C*.

The simplest (if feasible) is to reduce the system head so that it will pass through point *C*. This might be achieved by fully opening intermediate valves in the system which had been partially closed. In old systems, much of the head loss might be due to severe constriction in the piping bore by deposits of lime. Cleaning or replacing the piping would alleviate this condition.

Another, not often too feasible an approach (it is costly) is to reduce system head by replacing small piping having a high velocity flow with piping of larger diameter.

Sometimes Speed Can Be Changed

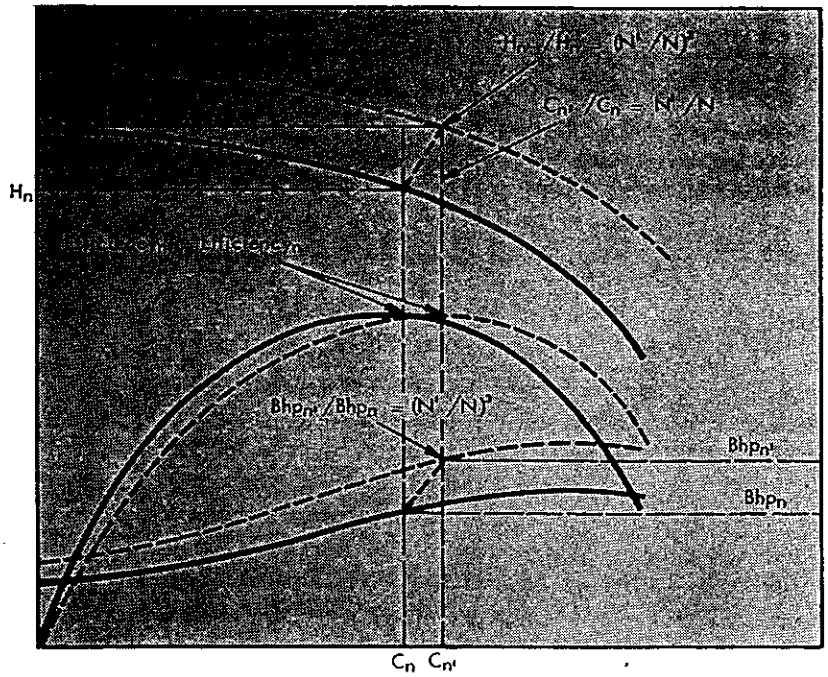
The operator might be able to increase pump speed so that it would develop a new head-capacity curve which would pass through point *D*, as illustrated by the broken-line curve in Fig. 3. Electric motor drive speeds are usually unchangeable, but such a solution may be possible with steam turbine drives, which allow for turbine speed changes over a wide range, usually with no or simple modifications.

If the driver speed only is changed, without any other change in pump or system, more horsepower will be required by the pump. This might be determined from a pump curve or calculated, since for such conditions of speed change only, the horsepower varies according to the cube of the speed over a moderate speed change.

Another Possibility: Impeller Change

Failing other means of increasing pump discharge head it is frequently possible to install a larger diameter impeller, or impellers. This also will increase required horsepower.

If a surplus of driving horsepower previously existed, the available horsepower with new, larger impellers might still be adequate. Some people solve the problem by seriously overloading the driver. This is fine only for the supplier of new drivers. An electric motor can talk back only by circulating an aromatic perfume of burning armature insulation (when it suffers the affliction of oversized fuzes or jammed



4 SPEED CHANGE (increase) from value at which test curve was run will result in changes to performance curves as shown here. Changes can be calculated so long as difference is not excessive

breaker cut-out) — the steam turbine is much more sensible, like a burro it refuses to work any harder than its normal rating plus a reasonable overload.

At any rate, the change required for a particular problem of increased head or capacity is not, in most instances, so radical that it cannot be solved in one of the ways suggested without going to the expense of a new pump.

Capacity Is Easily Determined

To determine the capacity of flow of a specific pump is a simple matter if a test curve for that particular pump (at its rated speed) is available.

The total dynamic head of the pump at any operating point will be the difference in readings between a pressure gage installed at the pump suction flange and another at the discharge flange. For accuracy, the gage readings should be corrected to the pump centerline and the difference in velocity head at discharge and suction added, but in all but low head pumps these correctives are minor.

By checking where pump head value falls on the test head-capacity curves, the capacity may be read directly. In Fig. 3, if the head corresponds to point *B* then the capacity is read directly below.

The Effect of Speed Change

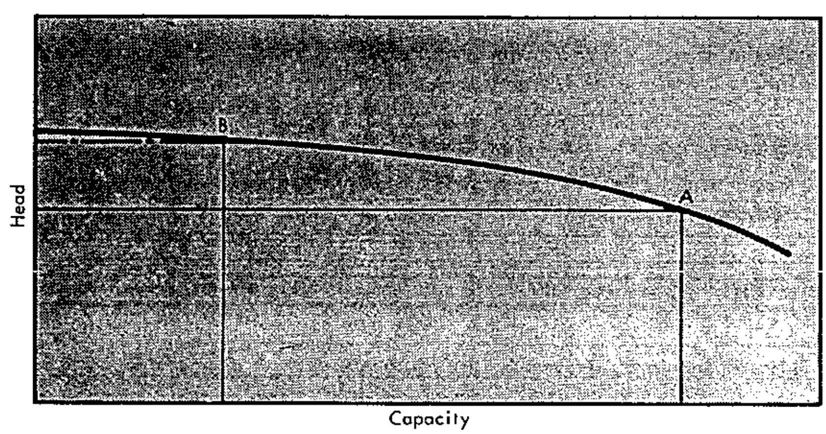
Suppose the pump doesn't run at the same speed as shown by its test curve? The curve may have been made at 1150, 1750, or 3500 rpm, whereas the pump actually runs at 1200, 1715, or 3600 rpm. Or the pump may have been purchased for 1750 rpm and a later decision made to increase driver speed to 2200 rpm. What about the new performance curve of the pump?

If N is the speed at which the test curve was run, and N' is the new speed, then at any selected capacity on the test curve

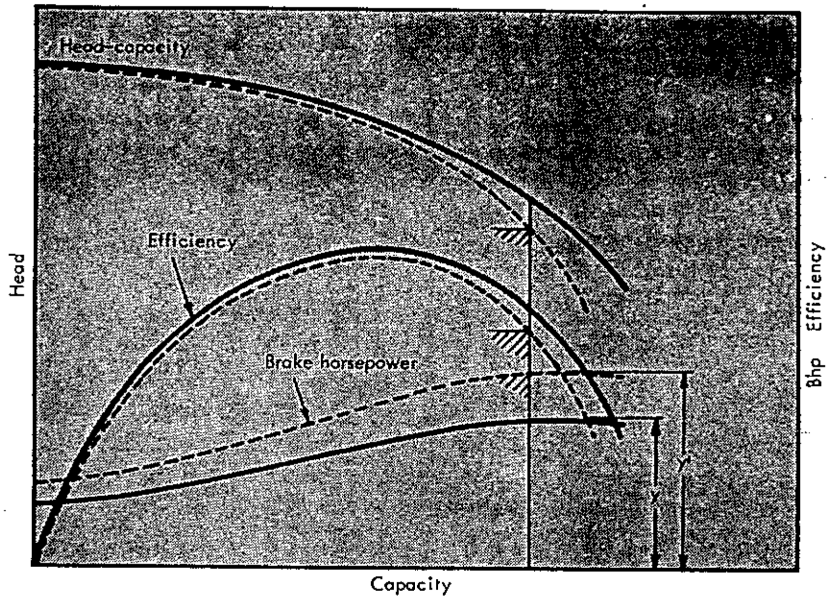
$$\text{New capacity} = \text{Old capacity} \times N'/N$$
$$\text{New head} = \text{Old head} \times (N'/N)^2$$
$$\text{New brake hp} = \text{Old brake hp} \times (N'/N)^3$$

The pump efficiency will remain the same.

All the foregoing regarding the effect of a speed



5 HEAD-CAPACITY curves for centrifugal pumps have common "falling" characteristic. As flow increases, head drops. This curve is "flat" — big change in flow produces only slight change in head



6 VISCOUS FLUIDS require changes in performance curves of nature shown here. Power requirement may be critical, since driver sized "on water" may be overloaded by more viscous fluid

change are approximate only, and are subject to correction for radical changes. For instance, because of the considerably increased hydraulic velocities, one might be disappointed in calculating anticipated performance at 3500 rpm on 1750 rpm tests.

Typical Problem, Result

Suppose the turbine drive pump which is to have its speed changed from 1750 to 2200 rpm develops 350 ft of head, 950 gpm capacity, and requires 116.5 brake hp for an efficiency of 72 percent when handling clear cold water at 1750 rpm. What will be the new conditions?

$$N'/N = 2200/1750 = 1.258$$

Therefore:

$$\text{New capacity} = (950)\ (1.258) = 1195 \text{ gpm}$$
$$\text{New head} = (350)\ (1.258)^2 = 552 \text{ ft}$$
$$\text{New brake hp} = (116.5)\ (1.258)^3 = 231 \text{ Bhp}$$

The efficiency remains "unchanged" at 72 percent. Actually, there tends to be a slight increase in efficiency because certain losses don't increase as rapidly as pump output.

The preceding relationships are illustrated in Fig. 4.

Head-Capacity Curve Shape

The head-capacity curves for an assortment of centrifugal pumps would show a common "drooping" or falling characteristic. This means that with an increase of flow there is a definite decrease in head. Sometimes this is a very desirable feature; as where two or more pumps are to operate in parallel, so that slight changes in head will not cause wide swings in the distribution of the total flow between the two pumps. It is also desirable where there are wide variations in the controlled discharge pressure, to minimize the variations in flow.

Sometimes, as with a boiler feed pump operating alone, it is desirable that the delivered capacity vary widely with only a slight variation in pressure. In Fig. 5, there is no great difference in heads between points of operation *A*, and *B*, but there is in their capacities. This pump has what is referred to as a *flat* characteristic.

Viscous Fluid Alters Curves

It has been stated earlier that corrections will be required for viscosity to the curves for pump head, capacity, efficiency, and horsepower.

The nature of the corrections to the performance curves will be as illustrated by the broken-line curves in Fig. 6.

Frequently, in order that the pump driver (motor, turbine, or engine) not be overloaded in the event that the pump accidently or purposely is called on to deliver maximum capacity, the driver may be sized so as to be "non-overloading."

In this event, the driver horsepower, normal rating, must equal or exceed the maximum required pump brake horsepower as shown on the pump performance curve for the specific gravity and viscosity of the liquid to be handled.

In Fig. 6, we see that the driver would have to have a horsepower rating y to be non-overloading on water, but y' if used with the particular viscous fluid for which the curve is drawn. Similarly, in Fig. 1, the maximum horsepower for water would be y, but for gasoline, with a specific gravity of 0.75 it would be reduced to y', where y' equals $0.75y$. ⧺

The concluding article in this series will deal with the subject of Net Positive Suction Head and what it means to the engineer selecting centrifugal pumps.

Selecting centrifugal pumps —

Net Positive Suction Head

— what it is
— what it means
— how to include it in your calculations

By HAROLD W. WOODHOUSE
Assistant Professor
Mechanical Engineering Department
Stevens Institute of Technology

This article treats, from start to finish and in detail, the subject of Net Positive Suction Head in reference to selecting centrifugal pumps. It is therefore valuable by itself as a comprehensive discussion of this important variable; especially so since it includes a number of practical examples to illustrate definitions and necessary calculations.

This is also the final article in a series of three dealing with the many things which the engineer must consider in order to be certain that the right pump is selected for the job at hand.

A SET of performance curves for a centrifugal pump may or may not include a curve of net positive suction head. Sometimes it is vital that the *available* NPSH be known so that we can be sure it is greater than the NPSH required by the pump.

What is net positive suction head? The total suction head (in ft of liquid absolute) determined at the suction nozzle and corrected to datum (pump centerline) less the vapor pressure of the liquid (in ft absolute).

There are various objections to the use of NPSH; one being that it creates the impression that the pump sucks the liquid whereas it is actually pushed into the pump by the action of atmospheric pressure plus or minus a static head or lift in an open cycle, or by the static head only if suction is from a closed vessel with the liquid at its boiling point.

Except in this last instance, NPSH does not equal static head or lift, but in that instance (closed vessel, boiling liquid) all the NPSH available is the height of the liquid level above the pump centerline (ft) minus all the piping losses in the suction side.

In Fig. 1, there is a static head of 6 ft on the suction side of the pump. The NPSH available at the pump flange might be, say, 35 ft or 2 ft, depending on the liquid, its temperature, altitude, whether the tank is closed or open, and on the pipe friction between tank and pump.

The 6 ft static head might be eliminated by the friction in the suction piping and fittings, so that the pump might have to operate as if with a suction lift. It depends — the pump might be next to the tank or 1000 ft away.

In Fig. 2, there is a theoretical 6 ft lift on the suction side. The NPSH available at the pump might be, say, 23 ft, or zero. The actual lift on the pump might not be much more than 6 ft — or it could be 25 ft or more, depending again on pipe friction.

NPSH Available *vs* Required

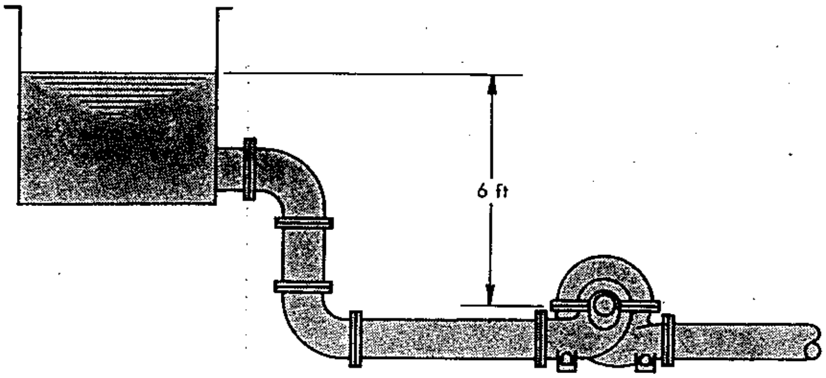
A clear distinction should be made between NPSH *available* and NPSH *required*.

The available NPSH refers to that excess of pressure of the liquid over its vapor pressure at the exact condition of the liquid as it arrives at the pump suction flange. Note, however, that head conditions have as a reference point the pump centerline. This means that all influences having any effect on the condition of the liquid on its journey from source to pump must be considered. The exact composition of the liquid and its temperature must be known to determine its vapor pressure. Again, for open cycle operations, altitude of the pump must be known since atmospheric pressure will decrease with an increase in altitude.

Every pump operating at a particular speed and capacity requires a certain minimum NPSH for satisfactory operation. This "need" is called the *required* NPSH. It represents the head loss which occurs in transporting the liquid from the suction flange to the point of minimum head just inside the impeller vanes.

If the available NPSH does not exceed the required NPSH, then cavitation will occur. Actually, to allow for discrepancies in estimating the available NPSH there should be a slight safety margin allowed.

Graphic illustrations of NPSH and the determining



1 HIGH TANK location relative to pump centerline results in "static head" condition, excluding pipe resistance. NPSH will then exist in accordance with Fig. 3, opposite

factors for the two tank/pump relationships are shown in Figs. 3 and 4.

If cavitation does occur, it can vary all the way from being barely discernible to being detrimental to the life of the impeller and/or shaft.

In passing, it might be mentioned that some condensate pumps operating in a closed cycle where the available NPSH is only 1 or 2 ft are often cavitating without noticeably detrimental effect, but that these pumps are of a special design.

Apparently, the seriousness of the cavitational effect is roughly tied to the quantity of liquid being handled, so that cavitation forces in a condensate pump of small flow are insufficient to develop offensive symptoms.

Barometric, Vapor Pressures

Looking ahead for a moment, note that Fig. 5 is a graphic presentation of the effect of altitude on atmospheric pressure, and that Fig. 6 gives water vapor pressure as a function of water temperature. Both of these direct-reading charts will be useful in calculating NPSH, as shown in the following, and in the examples to come.

It is well known that standard sea level barometric pressure equals 29.92 in. Hg. This is the height of a column of mercury which the pressure of the atmosphere will support. This value also can be expressed as 14.7 psi.

The height of a column of liquid which this atmospheric pressure will support is such that, according to its specific gravity, it will exert a downward force of 14.7 psi to balance the force exerted by the atmosphere.

For water, since 1 in. Hg equals 1.134 in. WG and 1 psi equals 2.306 ft WG, we have

$$1.34\ (29.92) = 14.7\ (2.306) = 33.9 \text{ ft. WG}$$

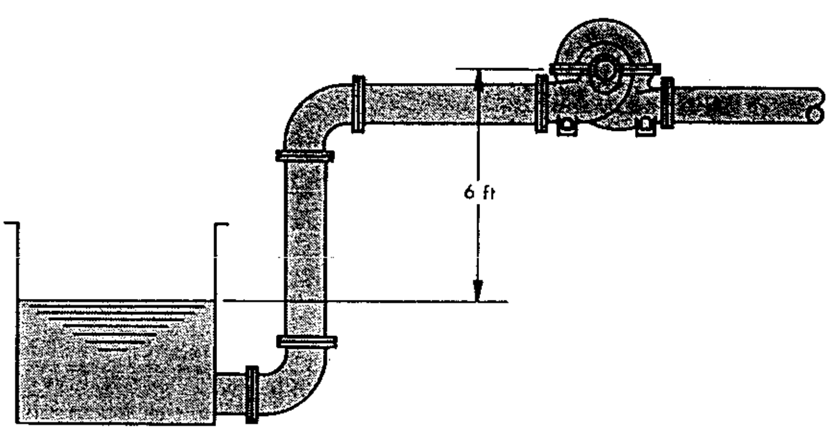
So, for normal cool water of specific gravity 1.0 at sea level there is a total pressure equivalent, H_a of 33.9 ft of head available at the surface of the water. H_a, then, can be defined in general terms as the absolute pressure on the surface of the liquid from which the pump is taking its suction, expressed in ft of the liquid.

H_a will decrease with altitude, and this reduction may be of considerable importance in some installations. Here, then, the usefulness of Fig. 5 becomes apparent.

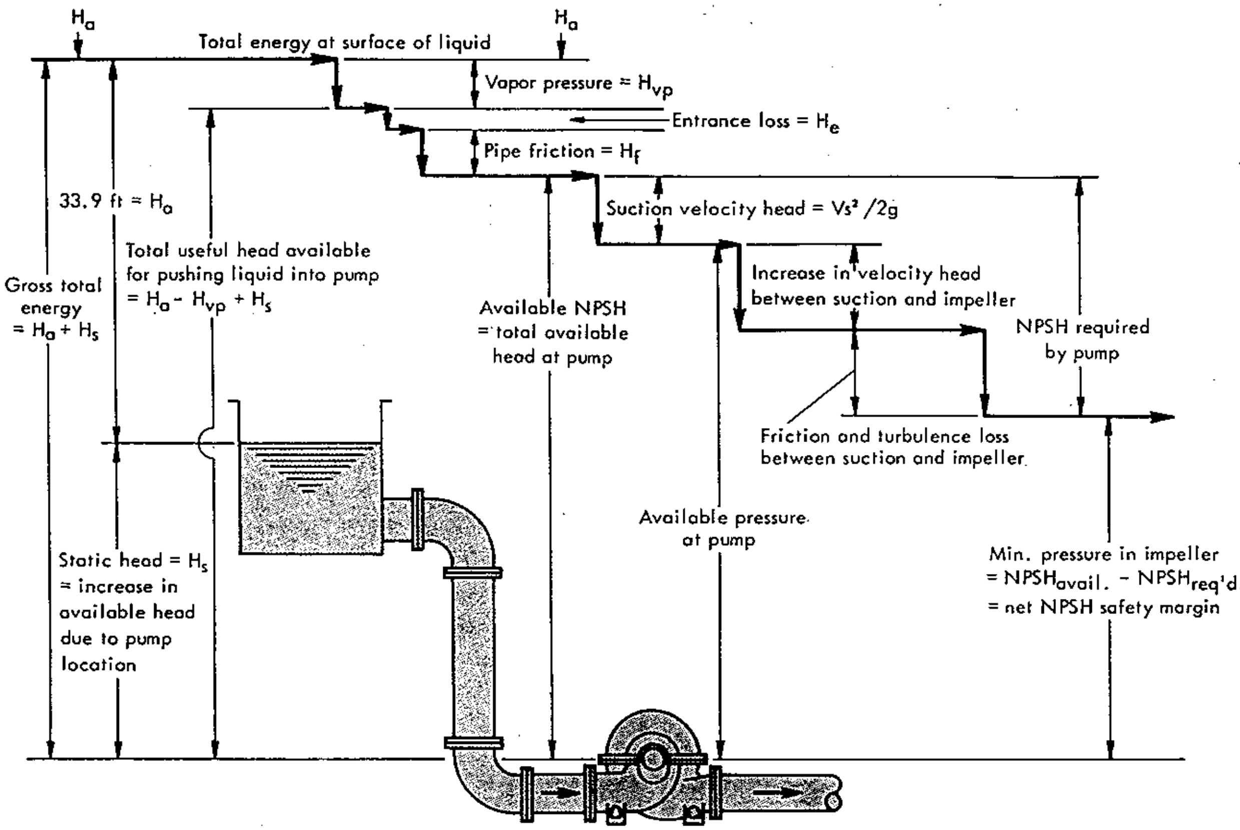
To H_a must be added the static head, H_s (Fig. 3) or the static lift, H_l (Fig. 4) must be deducted from it to obtain the gross available total energy on the suction side.

Of this, a small amount of the energy is vapor pressure, H_{vp}, which must be deducted from the gross total to obtain the useful available head. For cool water at sea level this value would equal about 1 ft.

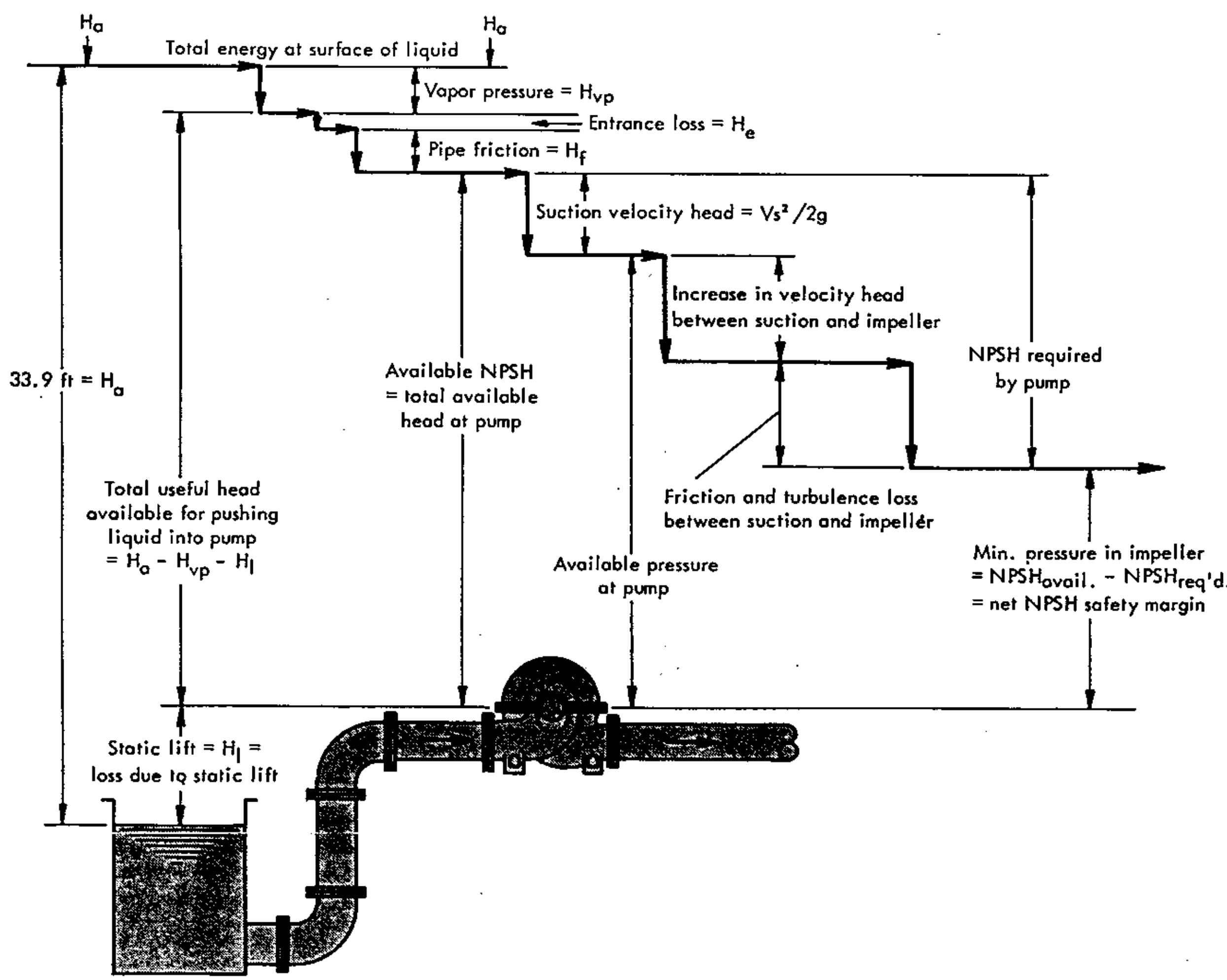
Fig. 6, which expresses vapor pressure as a function of water temperature, is therefore useful in providing this additional qualification.



2 LOW TANK relative to pump centerline means "lift" situation, disregarding system resistance. Fig. 4, opposite, shows NPSH factor relationships for this arrangement



3-4 RELATIONSHIPS bearing on determination of NPSH for two basic pump/source arrangements



Two Subtractions Remain

Referring again to Fig. 3, we see that the total useful head available for pushing the liquid into the pump equals $H_a \div H_s - H_{vp}$. From this must be deducted the losses incurred by the liquid on its way to the pump, and after this is done, the final remainder will be the net available NPSH.

The first loss to be deducted is the entrance loss, H_e, where the liquid enters the system piping. Next, the pipe friction, H_f must be subtracted.

The remainder is the available NPSH — the net available head in excess of the vapor pressure and minus the suction line losses at the pump suction, referred to the datum of pump centerline. It should be noted that the velocity heat at suction, $V_s^2/2g$, is included in this value.

Restate Equation

Referring to Fig. 3, which shows the available NPSH, the required NPSH, and the pressure margin for a pump operating with a static head on the suction (flow from an open tank), we see more clearly that the preceding may be written

$$NPSH_{avail} = (H_a - H_{vp}) + H_s - H_e - H_f$$

If a gage is installed at the pump suction flange so that the gage elevation corresponds to the centerline of the pump, then the available NPSH as measured on an actual installation is

$$NPSH_{avail} = 2.31(P_a + P_s - P_{vp})/W_g + V_s^2/g$$

where

$NPSH_{avail}$ = available net positive suction head, ft of liquid

P_a = absolute atmospheric pressure at installation elevation, psi

P_s = gage pressure at suction flange of pump, corrected to centerline, psi (minus value if reading is vacuum)

P_{vp} = vapor pressure of liquid at temperature of liquid at suction flange, psi

W_g = specific gravity of liquid at operating temperature

V_s = velocity of flow at and in pump suction flange, fps

Not only must the pump suction gage reading be corrected to centerline conditions (add ft in height above centerline, subtract if below center line) but if the suction gage is situated some distance ahead of the flange the pipe friction between gage and pump should be subtracted.

The velocity head is that which exists at the suction gage connection; it will vary with the amount of flow.

Similarly, the available NPSH, required NPSH, and the pressure margin are illustrated in Fig. 4 for a pump operating with a lift, with suction from an open tank.

Required NPSH

The *required* NPSH is made up of the suction velocity head ($V_s^2/2g$) plus the increase in velocity head between the pump suction flange and impeller vanes plus the friction and turbulence losses between suction flange and some point inside the inlet vane edge of the impeller, where the condition of minimum absolute pressure is reached. This required NPSH is determined by test of individual pump designs.

The final remainder is the minimum pressure in impeller over vapor pressure:

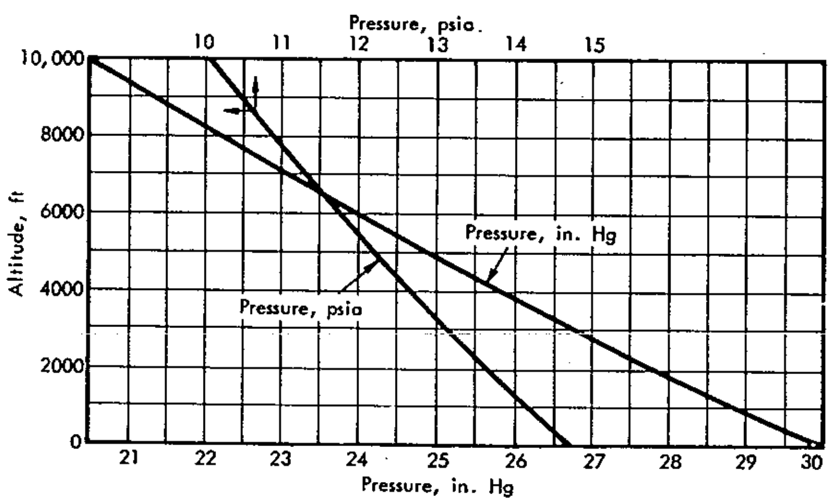
$$NPSH_{avail} - NPSH_{req'd}$$

and equals the safety margin available.

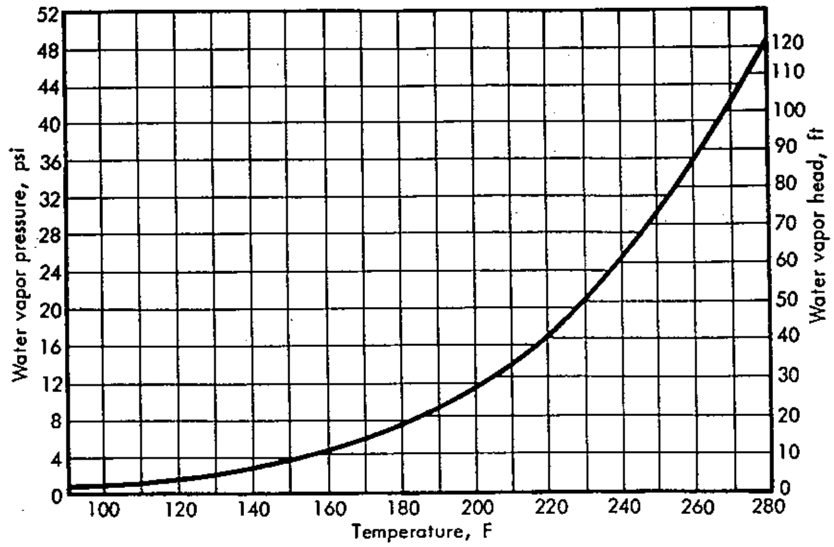
Head May "Disappear"

Inspection of Figs. 3 and 4 will indicate that the excess of available over required NPSH will normally be greater when there exists head on the suction side rather than lift. It is not quite so obvious that it may be possible to have a static head on the suction and yet not have enough available NPSH. There are two common circumstances where this might occur.

In one instance, the suction head might be small and the length of the suction piping so great that



5 ALTITUDE effects atmospheric air pressure as shown here. These values can be used directly as H_a, energy exerted on liquid surface to push fluid into pump



6 VAPOR pressure, H_{vp}, expressed as function of water temperature here, is included as part of lift or head which must be subtracted from or added to total pressure

pipe friction more than eliminates the effect of having any static head, so that, in Fig. 3

$$NPSH_{avail} - NPSH_{req'd} = 0 \text{ (or minus value)}$$

The other situation may arise when the pump is connected on the suction side to a closed vessel in which the liquid is at its boiling point. This situation is illustrated in Fig. 7. Here, the absolute pressure H_a on the surface of the liquid becomes the same as the vapor pressure H_{vp} of the liquid, and H_a—H_{vp} equals zero.

Therefore in the equation for available NPSH

$$NPSH_{avail} = (H_a - H_{vp}) + H_s - H_e - H_f$$

the first term is zero, so that for a closed vessel with the liquid at or near the boiling point we may write

$$NPSH_{avail} = H_s - (H_f + H_e)$$

which is to say that the available NPSH equals the static head minus the piping losses.

This shows that where the pump suction is from a closed vessel, with liquid at its boiling point, the only available NPSH is the static head of the liquid surface above the pump centerline, minus all the piping losses, and so, in some installations the margin between available NPSH and required NPSH may be zero or even a negative value. This spells trouble.

In a Closed System —

Consider an installation where pressures can be measured — a closed system with liquid at its boiling point. If, in Fig. 7, the gage vapor pressure on top of the tank is P_l, then the absolute pressure is

$$P_{vp} = P_a + P_l$$

and if we substitute this for P_{vp} in the equation for available NPSH:

$$NPSH_{avail} = 2.31/W_g[P_s + P_a - (P_a + P_l)] + V_s^2/2_g$$
$$= 2.31/W_g[P_s - P_l] + V_s^2/2g$$

This is to say that the measured available NPSH in a closed system equals the difference between the gage reading at the pump suction and the gage reading at the top of the vapor tank, corrected to ft of head for the liquid at its actual temperature plus the suction velocity head.

Examples in Calculating NPSH

Example 1

Assume a pump handling 500 gpm of water at 173 F from an open tank. The liquid level is 9.7 ft above the pump centerline. Installation is to be made at sea level, and the proposed pump has a required NPSH of 12 ft at the specified flow.

Solution: First, the piping entrance and friction losses must be determined. These will depend on the volume flow, stated as 500 gpm, and on the size and length of pipe, number of fittings, and valves. Such information is available in hydraulic handbooks, and must be included for a proper determination. Assume in this instance that the loss is equal to 3.6 ft of head. Also:

P_a = standard sea level atmospheric pressure
= 14.7 psia
P_{vp} = vapor pressure 6.417 psia for 173 F (from steam tables or Fig. 6)
W_g = specific gravity of water at 173 F
= 0.973 (from steam tables)

Therefore:

$$NPSH_{avail} = 2.31/0.973 \times (14.7 - 6.417) + 9.7 - 3.6$$
$$= 25.75 \text{ ft}$$

Since the required NPSH is 12 ft, there is an ample margin over this required value.

Example 2

Assume the same conditions as in Example 1, except that now the pump is raised to an altitude of 7000 ft, and piping losses equal 11.6 ft.

Solution: Atmospheric pressure at 7000 ft equals 11.38 psia (from Fig. 5). Therefore:

$$NPSH_{avail} = (11.38 - 6.417)(2.31/0.973) + 9.7 - 11.6$$
$$= 9.9 \text{ ft}$$

Since the pump has a required NPSH of 12 ft, a deficiency of available head exists, and some modification should be made to bring the available NPSH up to at least 13 ft. The solution, depending on the circumstances, might be to increase the static head on the pump by elevating the tank another 3.4 ft, or by increasing the liquid level by that amount, or by reducing the temperature of the water.

Sometimes, where it is not feasible to make such changes, a pump can be obtained which requires less NPSH. Sometimes a reduction of NPSH can be obtained by using a different impeller design in the same pump, but often a larger pump with lower flow velocities is required.

Example 3

A pump is to move water at 135 F from an open reservoir. The pump is installed at sea level, and the water surface is 8 ft below the pump centerline. Piping friction losses equal 8.9 ft.

Solution:

$$NPSH_{avail} = (14.7 - 2.537)(2.31/0.985) + (-8) - 8.9$$
$$= 11.6 \text{ ft}$$

Here again, if the required pump NPSH is 12 ft, there is a slight deficiency which should be rectified.

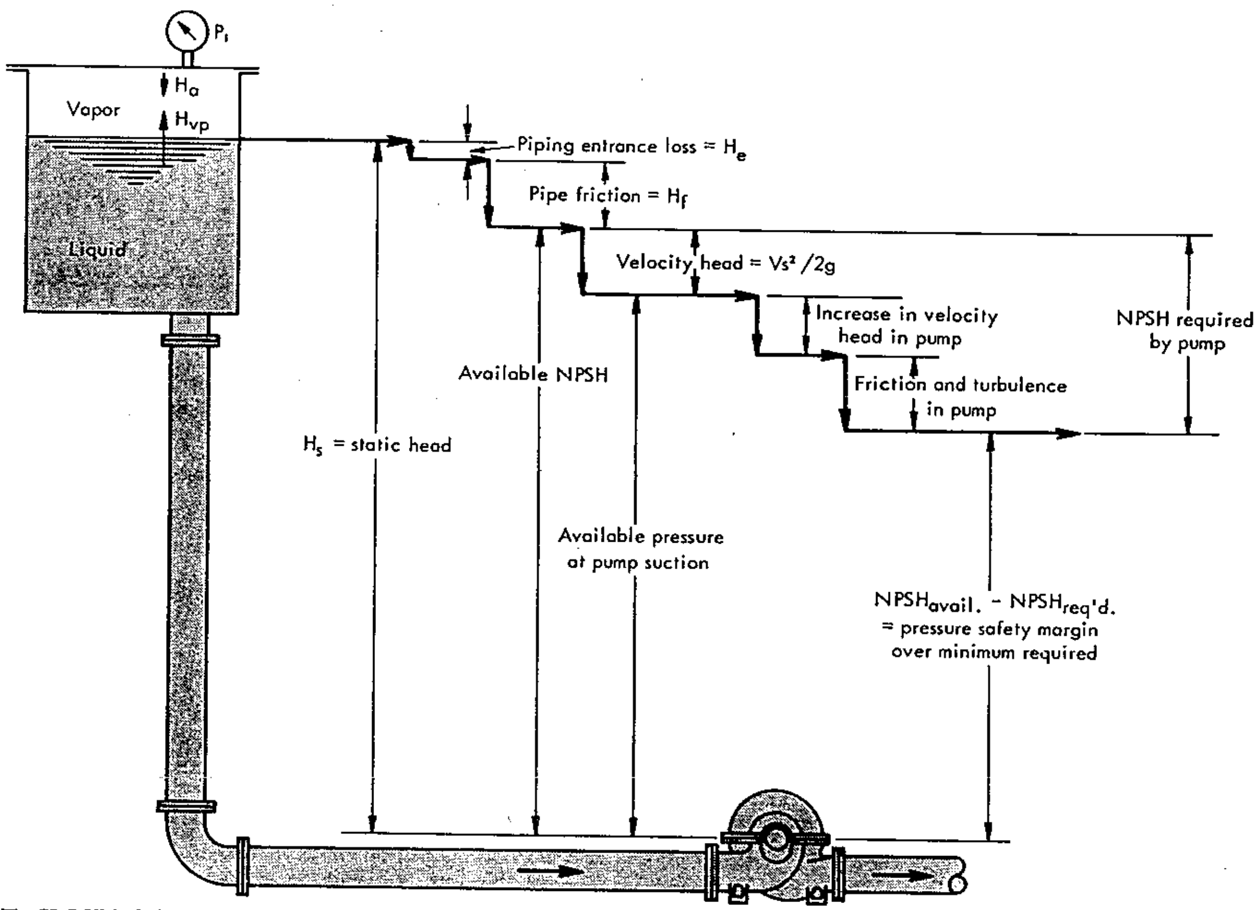
Example 4

A sea level pump is to handle 280 F water at 50 psig at the suction flange. Velocity in the suction pipe is to be 13.6 fps.

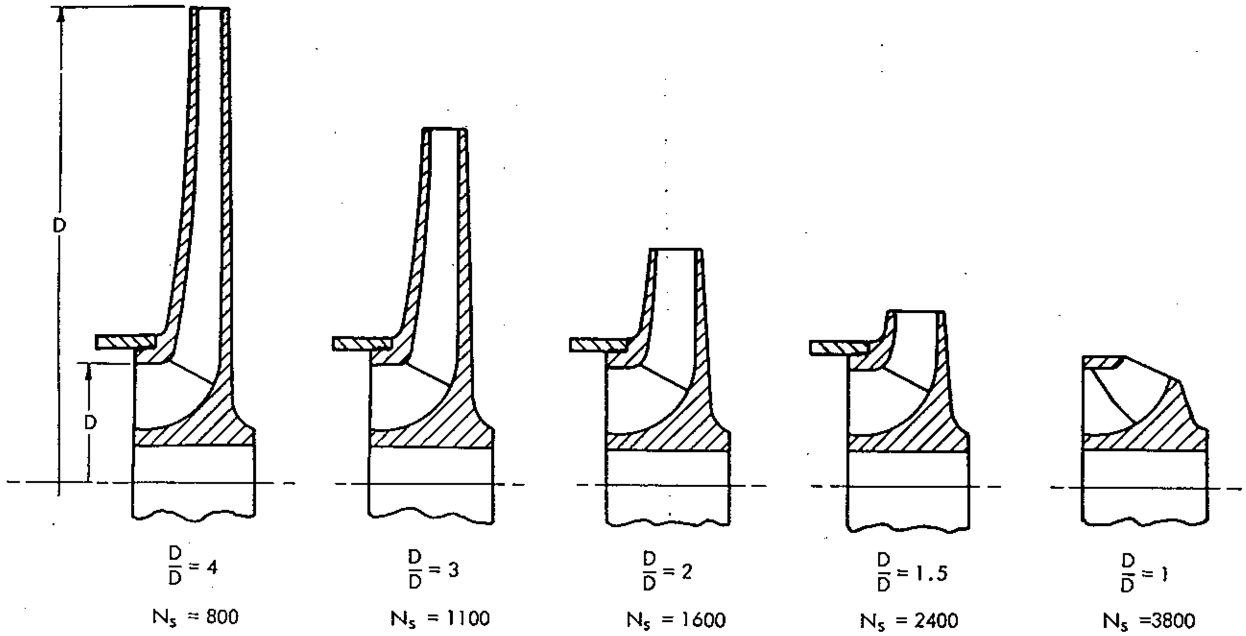
Solution:

$$\text{Velocity head} = 13.6^2/2g = 2.8 \text{ ft}$$
$$\text{Vapor pressure (280 F)} = 49.20 \text{ psia}$$
$$\text{or} \quad 49.20(2.31/0.93) = 122.1 \text{ ft}$$
$$\text{Total suction head} = (50.0 + 14.7)(2.31/0.93) + 2.8$$
$$= 163.5 \text{ ft}$$
$$NPSH_{avail} = 163.5 - 122.1 = 41.4 \text{ ft}$$

While this might seem ample, it should be kept in mind that a correction factor for high temperature will have to be applied to the required pump NPSH, and another correction for possible variations in boiler feed water temperature, so that the required



7 CLOSED TANK with liquid at boiling point means absolute pressure on liquid equals vapor pressure, so that NPSH equals merely static head minus the piping losses



8 FIVE IMPELLERS of increasing specific speed. Ratio of impeller outside diameter to inlet eye diameter (D_2/D_1) decreases with increase in specific speed

NPSH may show a considerable increase over cold water calculations.

Example 5

Water at the boiling point must be pumped from a closed vessel at 165 psig. The liquid level in the vessel is controlled by a high-low device. Maximum level of liquid above the pump centerline is 8 ft, and minimum level 6.5 ft. The pump will operate at sea level, and the pipe friction losses equal 1.5 ft.

Solution: The obvious thing to do is to use the minimum liquid level. This is an installation similar to that illustrated in Fig. 7, where $P_l = 165$, and where $H_a = H_{vp}$. Therefore

$$NPSH_{avail} = H_s - (H_f + H_e) = 6.5 - 1.5 = 5.0 \text{ ft}$$

Example 6

We are to pump 100 F butane from a closed vessel under pressure. The pressure in the vessel will be the vapor pressure of butane corresponding to 100 F, or 51.6 psia. The liquid level in the vessel is 8 ft above the pump centerline, and pipe friction losses equal 2.4 ft.

Solution:

$$NPSH_{avail} = 8 - 2.4 = 5.6 \text{ ft}$$

NPSH and Hydrocarbons

In connection with the pumping of hydrocarbons, it has been found that somewhat smaller required NPSH may be tolerated than if the pump were operating on water. The general feeling is that this is because most hydrocarbons are composed of elements having different vaporizing points, and that this softens the effect of a developing cavitation condition. For this reason, some pumps on critical hydrocarbon applications are operated with a smaller NPSH than would be used for water.

The Standards of the Hydraulic Institute include a chart giving the correction factors for non-viscous hydrocarbons. From this chart, the percentage of NPSH required relative to water requirements can be determined, when the specific gravity and vapor pressure of the hydrocarbon is known.

NPSH and Suction Lift

Obviously there must be a connection between NPSH and maximum permissible suction lift, since both relate to suction conditions. The equation for available NPSH may be rewritten as

$$(H_f + H_e) - H_s = (H_a - H_{vp}) - NPSH_{req'd}$$

but

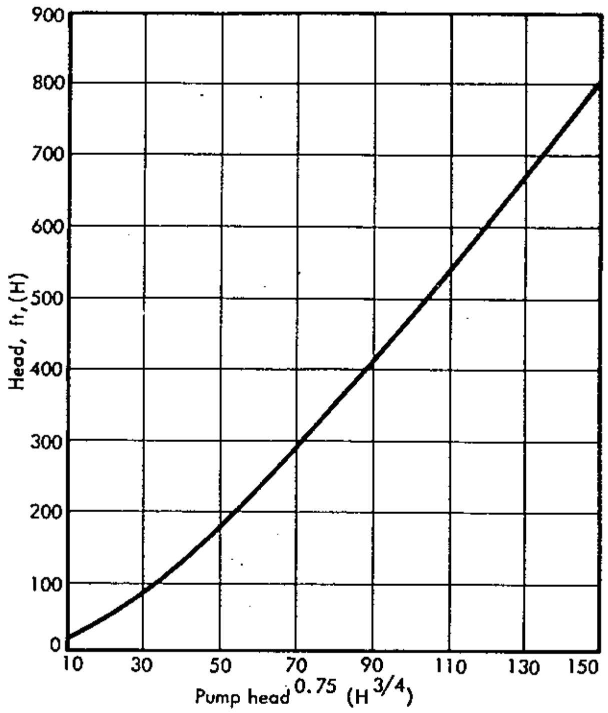
$(H_f + H_e) - H_s =$ total maximum permissible suction lift

and therefore

Total maximum permissible suction lift $= (H_a - H_{vp}) - NPSH_{req'd}$

which in turn equals the absolute pressure on the liquid surface, minus the vapor pressure, minus the required NPSH.

It should be noted that where a static lift exists on the suction side of the installation, we must substitute a $+H_l$ for the $-H_s$ on the left hand side of the equation just discussed. In effect, with a fixed maximum permissible suction lift a static head on the suction allows the presence of more piping friction losses in the suction line, whereas a static lift will put more restriction on the allowable suction



9 PUMP HEAD to ¾ power can be read from this curve for insertion in equation for specific speed

piping friction losses if the maximum total suction lift is not to be exceeded.

The Hydraulic Institute Standards include curves stating maximum practicable allowable suction lifts for single and double suction pumps operating at various specific speeds.

What Is Specific Speed?

Specific speed is defined as the speed at which a pump of a particular design would have to run to deliver 1 gpm against a head of 1 ft.

This is a guide to the type of pump impeller needed, and to the permissible operating lift. It is calculated from the following equation:

$$N_s = (\sqrt{gpm} \times rpm)/H^{3/4}$$

where

N_s = specific speed
H = head, ft per stage

With the specific speed known, the approximate proportions and type of impeller are indicated. Fig. 8 shows five impellers of increasing specific speed. Note the decreasing value of the ratio of the impeller outside diameter to the inlet eye diameter, D_2/D_1, with increasing specific speed.

At a specific speed of about 4000, the radial form of the impeller has virtually disappeared and a mixed flow has developed.

Specific Speed Is Guide

From an inspection of Fig. 8, it can be seen that a high head type will have a low specific speed, and a low head type a high specific speed. Now for maximum suction lift with suction liquid and vapor pressure fixed, the only variable is the required NPSH. The smaller the required NPSH, the greater the permissible suction lift. Specific speed is a reliable guide in determining the maximum permissible suction lift, which also means, therefore, that it serves as a guide to minimum required NPSH.

In terms of stated capacity and head, this means then that a low specific speed pump will operate with a greater suction lift than one with a higher specific speed.

If a pump (double or single suction) operating with a specified head, capacity, and specific speed, does not exceed the suction lift given by the curves in the Hydraulic Institute Standards there is not likely to be any cavitation trouble.

For convenience in calculating specific speed, Fig. 9 gives the values of $H^{3/4}$ for corresponding values of H.

Example 7

Assume a single suction pump and a suction lift of 12 ft, with a required head of 200 ft. The pump is motor driven at 3500 rpm. What will be the maximum capacity which can be handled without cavitation?

Solution: From the appropriate curve in the Hydraulic Institute Standards, we find that the maximum specific speed is 1625. Therefore

$$N_s = (\sqrt{gpm} \times rpm)/H^{3/4}$$
$$= (\sqrt{gpm} \times 3500)/200^{3/4}$$

and

$$Gpm = [(1625 \times 200^{3/4})/3500]^2$$
$$= 605 \text{ maximum}$$

Example 8

A double suction pump is to move 1000 gpm at 100 ft head, operating at 3500 rpm. What is the maximum permissible suction lift?

Solution:

$$N_s = (\sqrt{1000} \times 3500)/100^{3/4}$$
$$= 3480$$

From the Hydraulic Institute Standard curve for specific speeds of double suction pumps, we see that the maximum allowable lift is 11.5 ft.

Example 9

Pump and conditions are the same as in Example 8, except that pump speed is reduced to 1750 rpm.

Solution:

$$N_s = (\sqrt{1000} \times 1750)/100^{3/4}$$
$$= 1740$$

From the same curve the maximum lift can now be 26 ft. This points up the advantage of slower speed where lift is critical. ‡

How to match

Centrifugal Pump Needs And Bronze Alloys

for process piping

By Harold W. Woodhouse,
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• The multitude of bronze alloys available to the modern commercial-industrial user may be alternately a blessing and a problem. The accompanying article can serve as a base for bronze alloy selection and use, specifically as related to the selection of centrifugal pumps for a wide variety of process piping requirements. The major alloys are explained, with comments on some metallurgical problems in application.

Mr. Woodhouse was educated in England and has been associated in engineering design with various domestic firms in the heating, piping and air conditioning field. He is a member of the API committees on mechanical drive turbines, centrifugal compressors, and pumps.

In this modern age of myriad alloys there is no group of alloys more plentifully supplied with subtle variations in composition than the bronzes.

The too wide choice of bronzes frequently presents a problem to the mechanical, chemical, petrochemical or pump engineer who is required to specify the grade to be used.

In the remote, medieval engineering past of, say, fifty years ago, the engineer was limited, pretty much, to specifying materials as being either iron, steel, brass or bronze.

There were few alloys and special treatments. When an engineer wanted to emphasize that he would like a good iron casting he would specify "cylinder metal" — that being a close grain cast iron then in common use for the big reciprocating engine cylinders of the time. Manganese bronze had come into use for ship's propellers. Nowadays one could easily compile a list of 20,000 different ferrous and non-ferrous alloys.

What is bronze? What is brass? Basically, each has copper as its major ingredient with tin as an alloy for bronze and zinc for brass.

Types Multiply Quickly

From this starting point one can proceed to the selection of the particular variation most suited to the application. For specific industries there is the choice of architectural bronze, coinage bronze, hardware bronze, instrument bronze, medal bronze, statuary bronze and optical bronze as samples.

In broad categories there are also: aluminum bronze, lead bronze, phosphor bronze, manganese bronze, nickel bronze, vanadium bronze, hard bronze, Naval bronze; with each of these sub-divided into numerous variations.

centrifugal pumps and the bronzes

In the absence of specific field operational experience it would not be rare for a user to purchase several centrifugal pumps identical in all respects except materials so that material failures might be compared under identical operating conditions. This might include two or more types of bronzes. While there are many bronzes from which to select there is an infinitely greater number of liquids and conditions with which a centrifugal pump has to contend and often a customer (and pump manufacturer) has had no prior experience with the particular set of conditions faced.

Particular Alloys Sometimes Avoided

Sometimes a particular bronze is required, not for what it contains but for what it does not. The chief examples of this to be met in centrifugal pump use will be where a bronze must either be lead or zinc free.

The following are common alloying elements used with bronze:

Zinc, in spite of the basic definitions of brass and bronze, is frequently used in bronzes where its effect is to add strength and corrosion resistance to the copper..

Tin is used in bronze in amounts ranging from 5 to 20 percent. Its effect is to strengthen and harden the copper, making it tougher and more wear resistant.

Lead is added up to 35 percent for automotive bearings. In amounts of 5 to 25 percent it improves machinability. Combined with tin as an alloy it improves the antifriction qualities.

Aluminum is the base of a series of high strength alloys.

Iron is added to the silicon, aluminum and manganese bronzes for strength and wear resistance.

Phosphorus is added principally as a deoxidizer. In greater amounts it improves hardness and wear resistance.

Nickel refines the grain and toughens the alloy. It improves strength, corrosion resistance and whitens the alloy. Longer life is claimed for nickel-aluminum bronze ship's propellers over the older manganese bronze.

Silicon bronze alloys have high corrosion resistance, high strength and toughness.

Beryllium bronzes form an age-hardening group and they are the strongest of the copper alloys. They can present problems in machining not common to the other bronzes.

Manganese is used to produce a high strength alloy. It has a long history in marine applications.

While there are many ramifications to the subject the present article attempts only to consider a few salient facts relating to the bronzes and their use and non-use in centrifugal pumps.

List Available Bronze Alloys

Table 2 is a convenient reference list of some of the bronzes made by American manufacturers. Some discretion must be exercised in its use. Local availability may be important. A hard bronze available only in bar stock would be useful for shaft sleeves but not pump casings.

Metallurgical problems relating to bronzes and their application in centrifugal pumps might be broken down into the following broad categories:

1) Applications where almost any kind of bronze is considered satisfactory.

2) Where bronzes are undesirable because of the chemical or electrolytic action set up.

3) Where bronzes are acceptable provided certain elements are absent.

4) Where bronzes are unacceptable because of their susceptibility to erosion.

Industrial Pumps Use Bronze

The standard type of industrial pump is generally bronze fitted. This means that the impeller, impeller wearing rings, case wearing rings and shaft sleeves are bronze. This is the pump of category (1). This does not necessarily mean that in all instances the bronze in a standard fitted pump is unaffected by its operating medium.

Sometimes a material, or materials, which would be completely impervious to a particular medium would be so expensive it would be considered more economical to replace at intervals the most affected parts.

Occasionally, a bronze impeller is replaced by another, harder, ferrous material because of excessive erosion resulting from cavitation, which is purely a mechanical impact in its effect on the impeller. Using a harder material will lengthen the life of the impeller but the proper solution in such instances is to correct the flow conditions on the suction side of the pump.

Brine Causes Galvanic Action

In category (2) the most common example to be met is a pump handling sea water or brine. In the standard bronze fitted pump if used in such service there is a combination sea water, cast iron and bronze. The result is that a galvanic action is set up.

Table 1 is a tabulation of metals arranged as a galvanic series in decreasing order of potential as measured in sea water.

This galvanic table shows that cast iron and carbon steel rank higher in the series, in the presence of sea water, than such alloys as the yellow, aluminum or Admiralty brasses or "G" bronze. The result, with a bronze fitted pump, is an electrolytic action takes place which converts the cast iron casing interior surface into a mixture of graphite and iron oxide. This takes place most actively where the water velocity is highest. The process is known as graphitization.

The graphitized layer becomes increasingly impervious to the penetration of the water as its depth increases. The rate of attack on the underlying cast iron base is correspondingly reduced.

If, at certain points in the liquid flow the velocity or impact is sufficiently high to keep the graphitized layer washed away, the cast iron will be continuously exposed and this can lead to local failure of the casing.

It is for this reason that most pump users in this type of application request that the pump manufacturer, at the time the pump is purchased, specify a "retiring" thickness. In other words, when the casing wall has decreased in operation to the retiring thickness the casing must be scrapped and replaced.

The approach to the problem depends to a large extent on the initial cost limitation. Some will take a standard bronze fitted pump and replace worn parts periodically.

The electrolytic action can be reduced by using an all iron or all bronze pump. More costly solutions are to use monel or stainless steel in place of bronze. One recommendation for a compromise between optimum life and minimum cost is a 1 to 2 percent nickel cast iron casing, S monel impeller and impeller rings, Ni-Resist Type 2 case wearing rings, aged K monel shaft and shaft sleeves.

TABLE 1. — GALVANIC SERIES for various metals arranged in decreasing order of potential as measured in sea water

	Volts
Magnesium (H Alloy)	1.48
Zinc	1.03
Aluminum (Alclad 3 S)	0.94
Aluminum 52 S-H	0.74
Cast Iron	0.61
Carbon Steel	0.61
Stainless Steel. Type 430	0.57
Ni-Resist Cast Iron	0.54
Yellow Brass	0.36
Aluminum Brass	0.32
Composition 'G' Bronze (88-10-2)	0.31
Admiralty Brass	0.29
70/30 Cupro Nickel (0.47% Fe.)	0.25
Nickel	0.20
Stainless Steel. Type 316	0.18
Inconel	0.17
Stainless Steel. Type 410	0.15
Titanium (Commercial)	0.15
Silver	0.13
Stainless Steel. Type 304	0.08
Hastelloy 'C'	0.08
Monel	0.08

The following is a partial list of liquids for which standard bronze fitted pumps are *not* recommended and for which an all iron pump is usually supplied:

Acetaldehyde
Ammonia, Aqua
Ammonium Bicarbonate
Carbon Bisulfide
Enamel
Lime Water (Milk of Lime)
Lithium Chloride
Methyl Chloride
Miscella (20% Soybean Oil and Solvent)
Potassium Bichromate
Potassium Carbonate
Soap Liquor
Soda Ash (Cold)
Sodium Phosphate: Tribasic
Sodium Plumbite
Sulphur — Molten

Liquids Also Harmful

There are a great many liquids for which bronze is not recommended and one of a series of special alloys is necessary. Some of these are:

Acetic Acid (Vinegar)
Chromic Acid
Formic Acid
Hydrochloric Acid
Nitric Acid
Sulfuric Acid
Aluminum Sulphate
Brine, Sodium Chloride (Over 3% Salt, Hot)
Calcium Chlorate
Copper Nitrate
Copper Sulfate
Hydrogen Peroxide
Hydrogen Sulfide
Lead Acetate (Sugar of Lead)
Liquor — Pulp Mill: Sulfite
Magnesium Chloride
Photographic Developers
Silver Nitrate
Soda Ash (Hot)
Sodium Chlorate
Zinc Chloride

In category (3) where the absence from bronze of certain elements such as lead or zinc is desirable, and is frequently so specified by the ultimate user, the engineer can be confused by the numerous alloys from which he can choose.

Table 2 begins on page 148. Text continues on page 150.

Use this table to match centrifugal pump

TABLE 2. — REFERENCE LIST of some of the bronzes made by American manufacturers. Figures indicate percentage of each element present in alloy. Where total does not equal 100 percent, traces of other elements are present

	Name	Copper	Tin	Zinc	Lead	Phosphorus	Iron	Other Elements	Uses	Remarks
1	Bronze, Iron	82.5	8.6	4.4	...	...	4.0	...	Shafts, piston rods, screws, marine parts	Resists sea water corrosion
2	Elephant Brand Phosphor Bronze Grade S	Bal-ance	9.0-11.0	...	8.0-11.0	0.7-1.0	...	...	Bearings for locomotive, marine and stationary engines, roll neck bearings, piston rings	Very hard and durable
3	Nickel Bronze	82.0	8.0	2.0	...	...	...	8.0 Nickel	Superheated steam parts	Corrosion resistant
4	Valve Bronze	83.0	4.0	7.0	6.0	...	...	...	Valves	Corrosion resistant
5	Dudleys Phosphor Bronze	80.0	10.0	...	9.6	0.8	...	...	Bearings	Heavy duty
6	Bridge Bronze C	80.0	10.0	...	10.0	0.7-1.0	...	...	Bearings	Heavy duty
7	Arsenic Bronze	80.0	10.0	...	9.2	...	...	0.8 Arsenic	Bearings	Heavy duty
8	Leaded Phosphor Bronze	80.0	10.0	...	10.0	...	...	...	Bearings for high speed and heavy pressures	Resists shock and vibrations
9	Standard Phosphor Bronze	80.0	10.0	...	10.0	Up to 0.25	...	...	High speed bearings	Heavy duty
10	Ajax Phosphor Bronze	79.7	10.0	..	9.5	0.7	...	...	Bearings	...
11	Phosphor Bronze English No. 1	79.5	10.2	...	9.6	0.7	...	...	Fittings exposed to sea water, Small Springs, Bearings	High resistance to corrosion and fatigue
12	Semi Plastic Bronze	79.0 75.0	7.0 9.0	...	13.5-16.5	...	...	...	Machine bearings	Leaded bronze
13	Kern's Hydraulic Bronze	78.0	12.0	10.0	...	...	...	...	Vessels, pressure castings, valves, fittings	High strength
14	Kuhne Phosphor Bronze	78.0	11.0	...	10.0	0.6	...	0.3 Nickel	Hard bearings	Heavy duty
15	S.A.E. No. 67 Semi-Plastic Bronze	76.5 79.5	5.00 7.00	4.0 Max.	14.5-17.5	...	0.40 Max.	0.4 Max. Antimony 1.00 Max impurities	Soft bronze bearing, water pump bearings	Good antifriction qualities
16	Bronze, Carbon	75.0	9.7	...	15.0	...	...	...	Bearings	...
17	Phosphor Bronze American No. 1	70.0	13.0	...	16.0	1.0		...	Fittings exposed to sea water ,small springs	High resistance to corrosion and fatigue
18	Plastic Bronze	66.0	5.0	...	28.0	...	...	1.0 Nickel	Bearings	Heavy duty
19	Plastic Bronze Ajax	64.0	5.0	...	31.0	...	...	...	Bearings	...
20	Ajax Plastic Bronze	64.0	5.0	...	30.0	...	...	1.0 Nickel	Bearings	...
21	Kramer "X.X." Manganese Bronze	63.0	...	Bal.	...	...	2.0	2.5 Manganese, 5.0 Aluminum	Piston rods, valve stems, propellers, fittings	Corrosion resistant
22	Bronze-Vanadium	61.0	...	38.5	...	...	...	0.5 Vanadium	Pipes, tubes, fittings	High strength
23	Titan Bronze "B"	60.5	1.02	38.5	...	...	...	...	Marine shafts, under water fittings	Non-corrosive
24	Mueller Bronze Blue Tip	60.36	0.94	38.6	...	...	...	...	Nuts, bolts, condenser tubes, marine parts	Corrosion resistant
25	Chamet Bronze Type A	60.0	0.75	39.25	...	...	...	...	Welding rod, bolts, screws	Resists sea water
26	Bronze Leaded Chamet	60.0	0.75	38.5	0.75	...	...	...	Pipes, tubes, fittings	Free cutting
27	Bridgeport Bronze	60.0	0.75	Bal.	...	...	...	...	Condenser tubes, water pipes, nuts, bolts	Corrosion resistant
28	Ajax Special Valve Stem Bronze	60.0	...	37.0	1.0	...	...	2 Special hardeners	Valve stems	Free cutting
29	Titan Manganese Bronze	59.2	0.9	Bal.	...	..	0.9	0.4 Manganese	Bolts, rods forgings	Corrosion resistant high tensile
30	Bronze Victor	59.0	39.0	...	...	...	1.0	1.5 Aluminum, 0.03 Vanadium	Pipes, tubes, fittings	Corrosion Resistant
31	S.A.E. No. 73 Naval Brass Rod (Tobin Bronze)	59.0-62.0	0.5-1.0	Bal.	0.3 Max.	...	0.10 Max.	...	Piston rods, propeller shafts, plates, bolts, nuts, gears, gear bearings	Resistant to sea water corrosion
32	Chase Manganese Bronze	58.0	1.0	38.9	...	...	1.0	0.1 Manganese	High strength bolts	Tough
33	Valve Bronze	89.0	5.0	3.0	3.0	...	...	...	Valves	Corrosion resistant
34	Valve Bronze	89.0-85.0	2.5-10.0	0.9	0.-6.0	...	...	...	Valves	Corrosion Resistant
35	Tantalum Bronze	Bal.	...	...	...	...	...	1.25 Molybdenum, 0.20 Tantalum, 10.0 Aluminum	Steam fittings, valves	Corrosion resistant
36	Bronze Steam Valve	88.0	10.0	2.0	...	...	...	...	Steam valves	...
37	Bronze, Hard	88.0	10.0	2.0	...	...	...	...	Gears, bushings	...
38	Standard Admiralty Bronze	88.0	10.0	2.0	...	...	...	...	Bronze castings, gears, worm wheels	Tough, wear resistant
39	Bronze, Lafonds Pump	88.0	10.0	2.0	...	...	...	...	Gears, worm wheels, pump parts	Tough
40	Bronze, Cocks	88	10.0-8.0	2.0 6.0	...	...	...	...	Cocks, fittings	Corrosion resistant
41	Naval Phosphor Bronze (P-c) Cast	88.0	8.0	4.0	...	0.5	..	...	Bearings, gears marine parts	corrosion U.S.N.-46-B5f

requirements with available bronze alloys

TABLE 2 — Continued

	Name	Copper	Tin	Zinc	Lead	Phosphorus	Iron	Other Elements	Uses	Remarks
42	Bronze Steam Fitting	88.0	8.0	2.0	2.0	...	...	...	Steam fittings	Pressure tight
43	Naval Bronze	88.0	8.0	4.0	...	...	...	...	Steam and structural bronze, expansion joints, gears, valves, bearings	Tough
44	Hard Bronze	88.0	7.0	3.0	2.0	...	...	...	Gears and bushings	Tough
45	Naval Valve Bronze	88.0	6.5	4.0	1.5	...	...	...	Valve stems, valve seats, valve bodies	"Composition M" U.S.N.-46 B8d
46	Standard Nickel Bronze	88.0	5.0	2.0	...	...	...	5.0 Nickel	Valves, corrosion resistant parts	Corrosion resistant
47	Chase "444" Bronze	88.0	4.0	4.0	4.0	...	...	...	Bearings, condenser tubes	Corrosion resistant to brine, alkalis
48	Special Free Cutting Phosphor Bronze 610	Bal.	4.0	4.0	4.0	...	...	...	Bearings, bushings, valve and pump parts gears, pinions	Free cutting, wear resistant
49	Bronze, Nickel Aluminum	88.0	..	...	...	...	...	10.0 Nickel 2.0 Aluminum, Tin	Heat and corrosion resisting parts	High strength
50	Bronze, Diamond	88.0	...	...	...	...	...	10.0 Aluminum, 2.0 Silicon	High strength corrosion resisting castings	Wear resistant
51	Bronze Leaded	88.0-62.0	...	10.0-35.0	...	...	...	...	Fittings, screws, nuts, bolts	Free cutting
52	Bronze, Sillman	86.4	...	...	...	...	3.9	9.7 Aluminum	Strong corrosion resisting parts	Aluminum bronze
53	Bronze, Screw Nut	86.0	11.0	2.3	...	...	...	...	Screws and nuts	Corrosion resistant
54	Admiralty Bronze Modified	86.0	8.0	4.0	2.0	...	...	...	Valves, fittings	Pressure tight
55	Government Bronze	86.0-89.0	7.5-11.0	1.5-4.5	0.3	...	...	0.75 Nickel	Valves,gears, fittings, steam valves	Composition "G"
56	Apex Bronze	86.0	...	...	...	...	Bal.	9.5 Aluminum	Marine parts	...
57	Reich's Bronze	85.0	...	...	...	...	7.5	0.6 Aluminum, 0.5 Manganese	Strong corrosion resistant parts	Corrosion resistant
58	Dawsons Bronze	83.9	15.9	...	0.10	...	...	0.5 Arsenic	Journal bearings	Very fluid
59	Bronze, Chinese	83.0-72.0	1.0-13.0	0.7-14.0	10.0-20.0	...	Bal.	...	Bearings	Heavy duty
60	Corvic Bronze	98.5	1.5	...	...	...	...	...	Chemical engineering equipment	Corrosion resistant
61	Bronze Lafonds Malleable	98.0	2.0	...	...	...	...	...	Sheets, tubes, wire	Corrosion resistant
62	Phosphor Bronze 5% (Gr.A) 351	95.0	5.0	...	...	...	...	...	Tubes, sheets, wire, general parts	Resists fatigue corrosion and abrasion
63	Phosphor Bronze American No. 2	95.0	4.9	...	...	0.1	...	...	Screw propellers, worm wheels, bearings	High resistance to corrosion and fatigue
64	Bierman Tungsten Bronze	95.0	3.4	...	...	...	...	1.6 Tungsten	Strong corrosion resistant parts	Corrosion resistant
65	Sun Bronze	95.0	...	...	...	...	...	5.0 Aluminum	Pump rods, bushings, propeller blade bolts	Corrosion resistant
66	Bronze Screw	94.0	1.0	5.0	0.5	...	...	...	Screws, nuts, bolts	Free cutting
67	Phosphor Bronze Russian	93.7	5.8	0.34	...	0.16	...	...	Screw propellers, fittings exposed to sea water, bearings, worm wheels	High resistance to corrosion and fatigue
68	Bronze, Tetmajer Aluminum	93.0-86.0	...	...	...	...	0.7-1.0	4.6-10.0 Aluminum 1.0-2.7 Silicon	Bushings, fittings	Tough, high strength
69	Bronze Yale	92.5-90.0	0.5-1.5	7.5 8.0	0.7-1.5	...	...	...	Screw machine parts bolts, nuts, bushings	Free-cutting
70	Bronze Uchatius (Uchatin's)	92.0	8.0	...	...	...	...	...	Bearings, gears, worm wheels	Wear resistant
71	Chase Nickel Aluminum Bronze	92.0	...	...	...	...	...	4.0 Nickel, 4.0 Aluminum	Condenser tubing, pipe an corrosion resistant purposes	Corrosion resistant
72	Omega Brand Phosphor Bronze No. 209	90.0	10.0	...	...	...	...	Trace Phosphor In Sulphide Solution	In paper mill machinery	...
73	Phosphor Bronze 10% (Gr.D) 354	90.0	10.0	...	...	...	...	...	Tubes, sheets, wire, general parts	Resist fatigue and corrosion
74	Frontier No. 11	90.0	6.5	2.0	1.5	...	...	...	Bearings, screws, bolts	Formerly Bronze 11
75	Lumen No. 14	90.0	6.5	2.0	1.5	...	...	...	Valve bodies, oil pumps	Known as valve or Steam Bronze
76	Ajax Plastic Bronze	90.0	5.0	...	5.0	...	...	...	Bearings	...
77	Litinum Bronze	90.0	4.7	3.86	1.6	...	...	...	Nuts, bushings, bearings screw machine parts	Free cutting
78	Bronze Phosphor Grade "D"	89.5	10.5	...	...	...	...	...	Worm wheels, pumps	Tough
79	Bronze, Tin	89.0	11.0	...	...	...	...	...	Bearings, gears, worm wheels	Wear resistant
80	Lumen No. 9	57.5	0.75	40.0	...	...	1.0	0.5 Aluminum, 0.25 Manganese	Propeller blades and hubs, valves, pump bodies	Known as Manganese Bronze
81	Marine Bronze	57.5	0.15	Bal.	...	...	...	0.8 Nickel 0.5 Aluminum	Marine parts	...

TABLE 2 — Continued

	Name	Copper	Tin	Zinc	Lead	Phosphorus	Iron	Other Elements	Uses	Remarks
82	McKechnie's Bronze	57.0	1.0	41.0	0.5	...	1.0	...	Rods, nuts, bolts	High strength
83	Manganese Bronze 937	57.0-62.0	0.5-1.5	36.0-40.0	...	...	0.5-1.0	0.5 Manganese	Valve stem forgings, slotted and perforated screens	High strength and toughness, resists action of salt water.
84	Turbiston Bronze	55.0	...	41.0	...	...	0.84	2.0 Nickel, 1.0 Aluminum, 0.16 Manganese	Pistons	Highly resistant to sea water. Also called a Brass
85	Nickel Manganese Bronze	53.0	2.6	39.0	...	...	...	2.5 Nickel, 1.7 Manganese, Aluminum Lead	Corrosion resistant strong castings	Corrosion resistant
86	Bronze Steel (Stahl)	52.0-59.0	...	36.0-43.0	...	...	1.0	2.5-3.0 Manganese 1.0 Aluminum	Propellers, marine parts	Same as "Uchatius Bronze"
87	Machine Bronze	50.0	25.0	...	...	...	...	25.0 Nickel	Bearings	Wear resistant
88	Bronze, Sea Water	45.0	16.0	5.5	...	...	...	33.0 Nickel 1.0 Bismuth	Marine parts	Corrosion resistant
89	Delta Bronze 111	40.0-98.0	0.1-10.0	1.8-45.0	...	...	0.1-5.0		Solid drawn tubes for hydraulic purposes, condensers, gears, valves, pump parts	Corrosion resistant
90	Sun Bronze	30.0-50.0	...	...	...	...	...	60.0-40.0 Cobalt, 10.0 Aluminum	High temperature fittings	Heat resistant
91	Johnson Bronze Babbitt No. 11	6.0	87.0	...	...	...	...	7.0 Antimony	Bearings	Bronze Babbitt
92	Johnson Bronze Babbitt No. 10	5.0	90.0	...	...	...	...	5.0 Antimony	Bearings	Bronze Babbitt
93	Naval Bronze No. 4	4.0	36.0	...	44.0	...	...	16.0 Antimony	Bearings	Anti-friction
94	Johnson Bronze Babbitt No. 97	2.5	90.0	...	...	...	...	7.5 Antimony	Bearings	Bronze Babbitt
95	Buckeye Bronze	0.6	41.6	37.8	15.6	...	...	4.3 Aluminum	Bearings	Babbitt

While materials should be selected by a metallurgist many companies do not employ such a specialist and, even if they did, he would, wherever possible, rely heavily on the evidence of past field behavior.

Selection May Be a Compromise

The old cliches that "nothing is perfect in this world" and "one can't have everything" could very well have been said by a pump engineer with an unusual liquid to handle.

While there are thousands of centrifugal pumps which have continuously operated for 40 years on water there are the unusual applications where, with the very best design and materials, pump life is measured in months.

TABLE 3 — LEAD FREE BRONZES. Numbers refer only to Table 2

1, 3, 13, 21, 22, 23, 24, 25, 27, 29, 30, 32, 35, 36, 37, 38, 39, 40, 41, 43, 46, 49, 50, 51, 52, 53, 56, 57, 61, 62, 63, 64, 65, 66, 68, 69, 71, 72, 73, 74, 79, 80, 81, 82, 84, 85, 87, 88, 89, 90, 91, 92, 93, 95.

TABLE 4 — ZINC FREE BRONZES. Numbers refer only to Table 2

2, 5, 6, 7, 8, 9, 10, 11, 12, 14, 16, 17, 18, 19, 20, 30, 35, 49, 50, 52, 56, 57, 58, 61, 62, 63, 64, 65, 66, 69, 71, 72, 73, 74, 77, 79, 80, 88, 91, 92, 93, 94, 95.

For convenience, Table 3 is a list of those numbers in Table 2 which are lead free bronze.

Table 4 is a list of those bronzes which are zinc free.

Erosion May Exclude Bronze

Applications in category (4) are those where bronze is rejected in favor of a harder material, usually because the harder material will resist the erosive action longer and by this resistance prolong pump life.

Erosive action may be the result of abrasives in the liquid or of cavitation resulting from inadequate pump suction conditions. Substitution of harder materials will normally increase pump life but in the case of cavitation the proper solution is to change suction conditions.

In some instances the necessity for another material can be determined only by field experience. On occasion, a material which would seem to be harder and generally better is rejected in favor of one of the tougher bronze alloys. ⌗

References

1) International Nickel Publications
2) Engineering Alloys — Woldman
3) S.A.E. Handbook
4) Hydraulic Institute Standards