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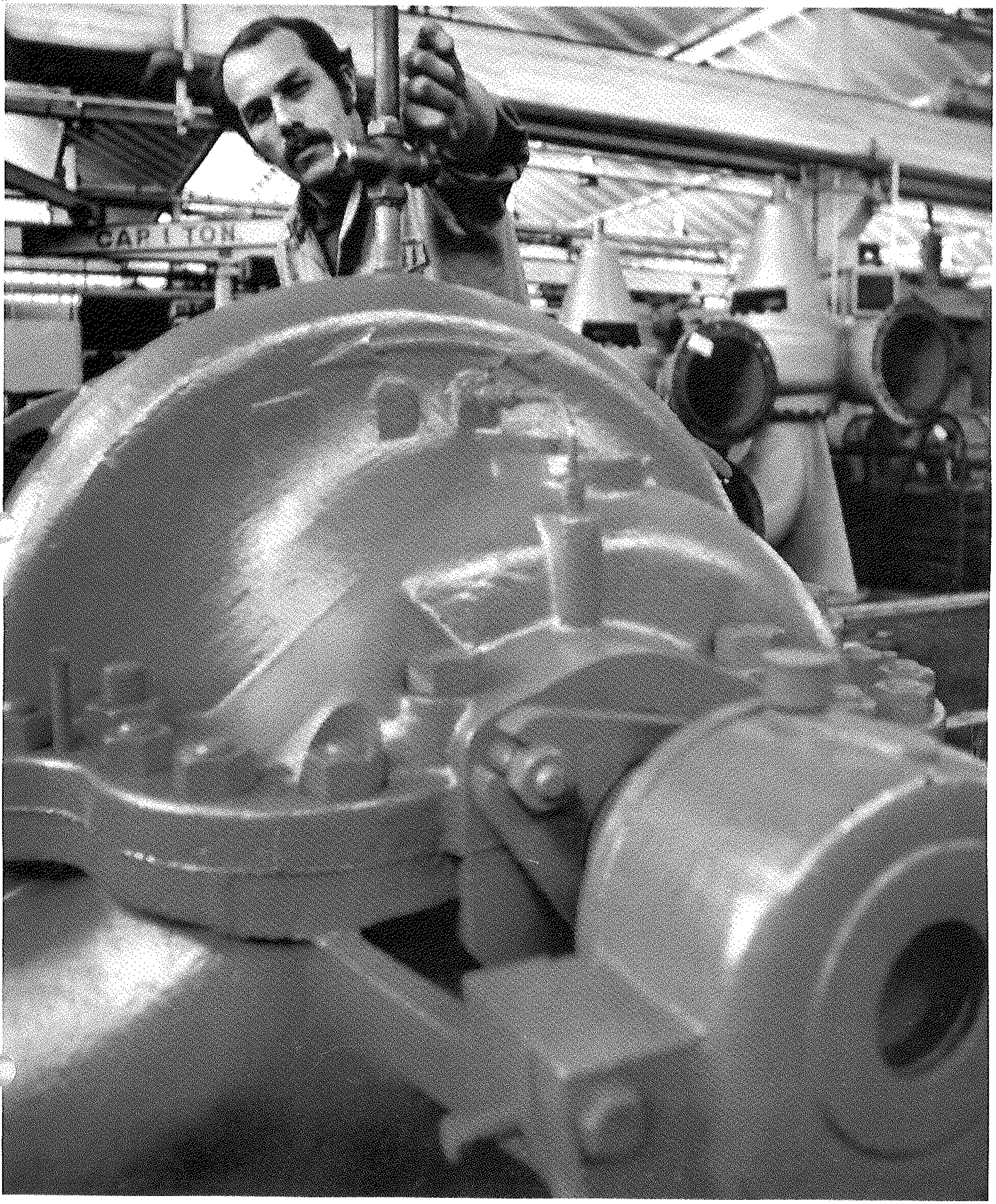
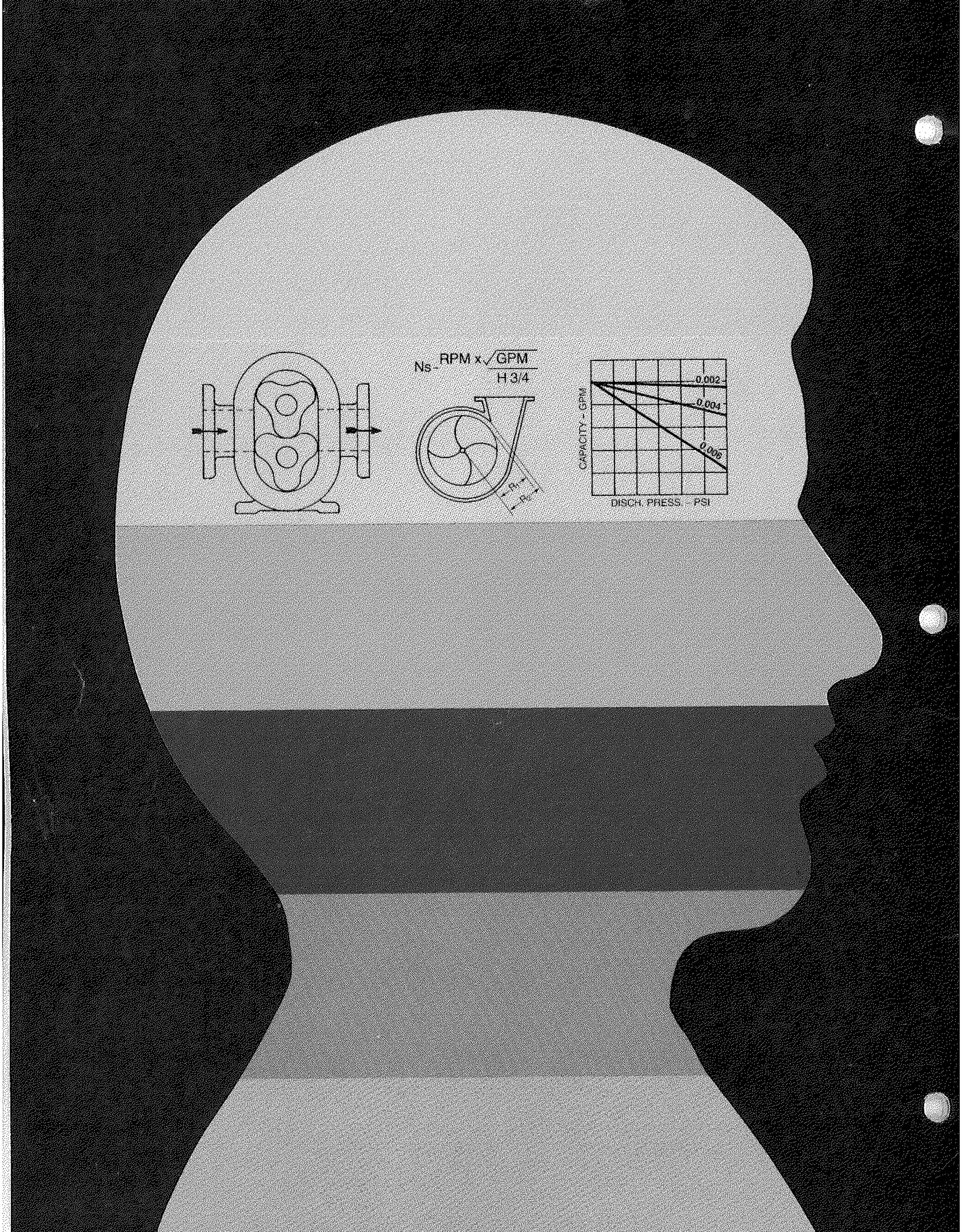
pumpworld




Ns- RPM x √GPM / H 3/4
0.002
0.004
0.006
CAPACITY - GPM
DISCH. PRESS. - PSI
R1
R2

Like all scientific knowledge, pump technology is a living and changing thing. So, although many of us work with pumps every day of our lives, there always seems to be a point or two in need of re-thinking and review, or perhaps one worth passing on to that new fellow down the hall. To accommodate these needs, this issue of *Pumpworld* presents the first installment of a new series reviewing pump basics.

"Pump Refresher" represents an updating of a series of presentations on pumping basics developed over the years by Worthington Pump designers and application experts, and polished in the course of dozens of seminars. It has been well received by audiences representing every degree of fluid-system expertise — and we hope *you* will find it useful, too.

When it comes to change in the field of fluid handling, Worthington has always been in the forefront. In his article "NPSH Guidelines for Centrifugal Pumps," W. C. Krutzsch continues that tradition by presenting a new alternative to current Hydraulic Institute Standards for determining NPSH requirements.

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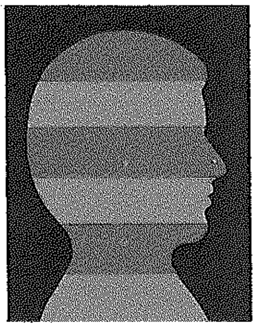

pump refresher

Centrifugal pump operating principles, performance and system curves

A pump converts the energy provided by a prime mover, such as an electric motor, steam turbine, or gas engine, to energy within the liquid being pumped. This energy within the liquid is present as velocity energy, pressure energy, static elevation energy, or some combination of these.

Rotation imparts two motions.
The rotating element of a centrifugal pump, which is turned by the prime mover, is called the *impeller*. The liquid being pumped surrounds the impeller, and as the impeller rotates, its motion imparts a rotating motion to the liquid (Figure 1).

There are two components to this motion. One component is motion in a radial direction, outward from the center of the impeller, caused by centrifugal force. As the liquid leaves the impeller, it also tends to move in a direction tangential to the outside diameter of the impeller. The actual direction the liquid will take is the resultant of the two components.

Velocity.
The energy added to the liquid by the rotating impeller is related to the velocity with which the liquid moves. Energy, expressed as pressure energy, is proportional to the square of the resultant exit velocity:

$$H = K\frac{V^2}{G}$$

Where H = energy, ft. of liquid; K = a proportionality factor; V = velocity, fps; G = acceleration due to gravity in ft/sec^2.

From these facts we can predict two things. First, we can say that anything that increases the tip velocity of the impeller will also increase the energy imparted to the liquid. Second, we can say that changing the vane tip velocity will result in a change in the energy imparted to the liquid which is proportional to the square of the change in tip velocity.

For example, doubling the rotative speed of the impeller would double the tip speed, which in turn would quadruple the energy imparted to the liquid expressed in terms of pressure.

Doubling the impeller diameter would also double the tip speed, which again would quadruple the energy imparted to the liquid. These facts can be used in evaluating and predicting the performance of an individual pump.

Liquid collected in volute.
What happens to the liquid which is being discharged from the tip of the impeller? Taking a volute pump as typical of centrifugals in general, the liquid is discharged from all points around the circumference of the impeller, and moves in a direction which is generally outward from the impeller. At the same time it is moving around with the impeller.
It is the function of the pump casing to gather up this liquid and direct it through the discharge nozzle or opening of the pump. The casing is designed so that, at one point, its wall is very close to the outer diameter of the impeller. This point is called the *tongue* of the casing (Figure 2).

Between the tongue and a point slightly to the left, a certain amount of liquid has been discharged from the impeller. This liquid must rotate with the impeller until it is finally discharged through the outlet nozzle of the pump. Additional liquid is discharged from the impeller at every point around the casing, and this must also travel with the impeller and be discharged through the outlet nozzle. As we continue around the casing, more and more liquid accumulates, which must be carried around between the wall of the casing and the outer edge of the impeller. In order to keep the velocity fairly constant, even though the volume of liquid increases, the area between the tip of the impeller and the casing wall is gradually increased from the casing tongue around to the beginning of the discharge nozzle.

Diffuser.
At a point just before the tongue, all the liquid discharged from the impeller has been collected. This liquid must now be led out into the discharge pipe. However, in most cases this liquid possesses a high velocity, which would mean a high friction loss in the discharge piping. Therefore, velocity is usually decreased through the diffuser, by increasing the area for flow. In this way, some of the high velocity energy is changed into pressure energy.

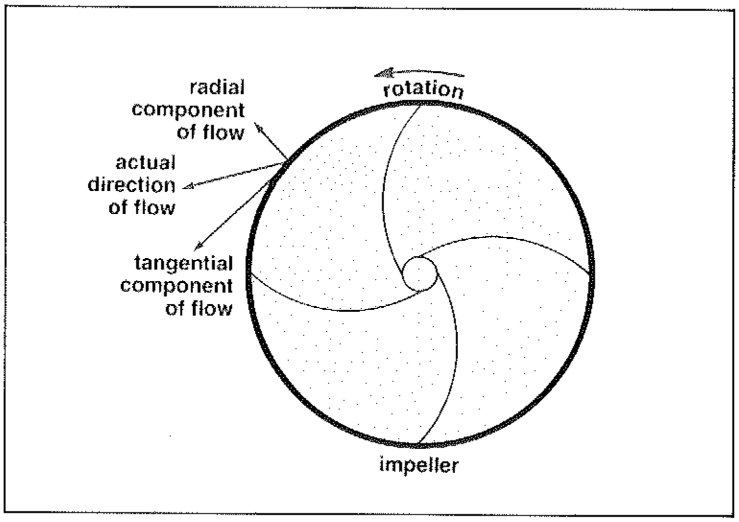

Figure 1

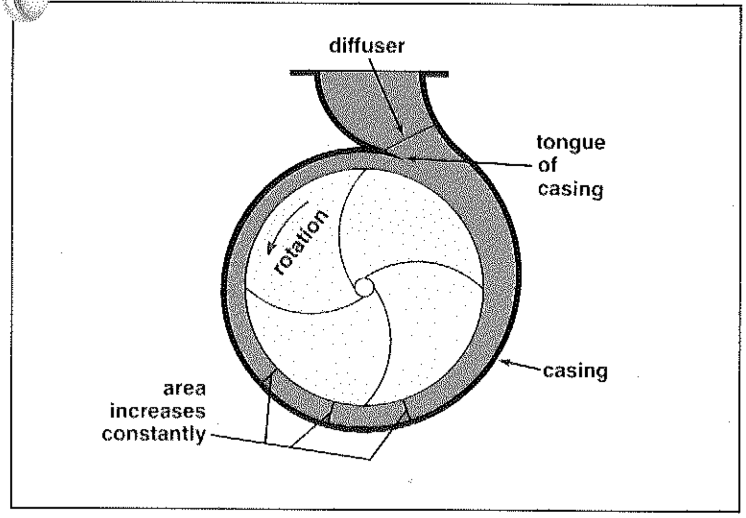

Figure 2

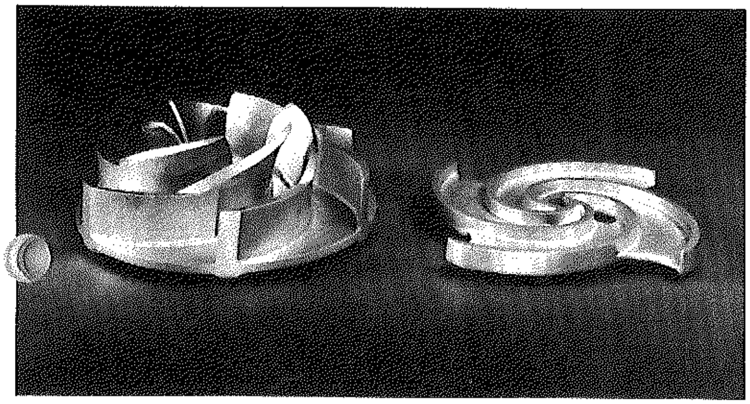

Figure 3

Capacity dictates impeller width.
In an actual centrifugal pump, the impeller is provided with vanes which act to guide the liquid. Furthermore, the impeller has a certain axial width, depending on the capacity it is intended to handle. As an example, Fig. 3 shows two impellers of approximately the same diameter, but varying widths. Although their diameters and rotative speeds are identical, the impeller on the left has a flow capability many times that of the impeller on the right.

Pump performance curves.
For any centrifugal pump, curves can be developed to show various relationships. When all these characteristics are plotted on one coordinate system, the capabilities of the pump are completely defined.

Note that every performance curve is based on a particular speed, impeller diameter, and viscosity (this is usually understood as the viscosity of water). While these figures are omitted for simplicity in this discussion, they must be present on any "real life" performance curve.

Head/capacity.
This curve indicates the relationship between the head, or pressure, developed by the pump, and the flow through the pump (Figure 4). As capacity increases, the total head which the pump is capable of developing is reduced. In general, the highest head occurs at a point where there is no flow through the pump; that is, with the discharge valve completely closed and the pump running.

Bhp/capacity.
In order for the centrifugal pump to deliver the desired capacity, a certain horsepower must be supplied. We can plot a curve representing the relationship between capacity and brake horsepower (Figure 5).

Efficiency/capacity.
Head/capacity and bhp/capacity are determined by testing an actual pump. The efficiency with which the pump operates cannot be measured directly, but must be calculated. The formula for efficiency is:

$$\text{efficiency} = \frac{\text{head} \times \text{capacity} \times \text{sp. gr.}}{3{,}960 \times \text{hp}}$$

From this formula, pump efficiency at given capacity can be determined and the efficiency curve can be plotted (Figure 6).

Npsh/capacity.
One other important characteristic of a centrifugal pump is always given on the pump performance curve: the relationship between the capacity the pump will deliver, and the npsh (net positive suction head) required for proper operation of the pump at that capacity (Figure 7). This data is obtained from actual tests.

Overall rating.
By plotting all these characteristics on one coordinate system (Figure 8), the capabilities and limitations of the pump are completely defined for a particular speed, impeller diameter and viscosity. Of course the performance curve also includes actual figures for the coordinates of head, gpm, npsh and bhp (Figure 9).

System curves.
After considering the capability of the pump, it is necessary to consider the requirements of the system in which it will work.

A very simple system is shown in Figure 10. Points A and B are on the same level. They are connected by a line through which the liquid is to flow. In this line we can assume heat exchangers, valves, and other equipment which add to the total friction loss. Friction loss is proportional to the square of the capacity (or velocity).

Figure 11 plots a curve for such a system. Friction loss is expressed as a number of feet of head. At O capacity, of course, since there is no flow, there is no friction loss.

Figure 12 shows the same system, with the complication that point B is higher than point A. Because of the difference in height, it is necessary to add more energy to the liquid to move it from point A to point B. The amount of energy we must add, expressed in feet, is exactly equal to the difference of elevation between point B and point A. Of course, we must still overcome the friction loss between point A and point B, as we did in the previous system.

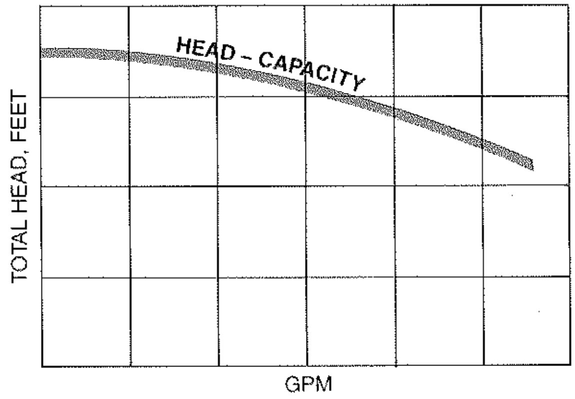

Figure 4

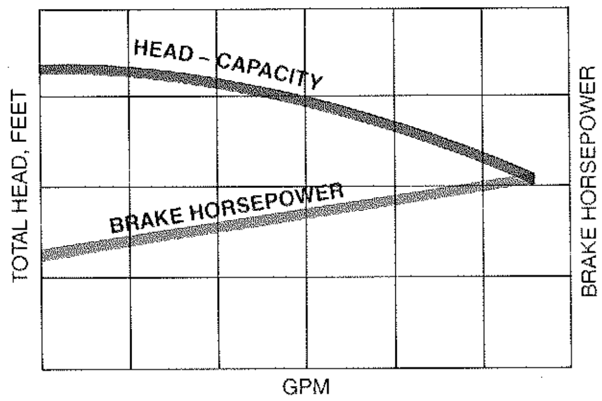

Figure 5

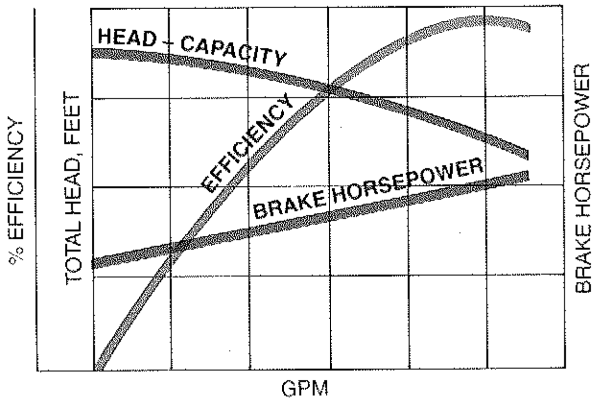

Figure 6

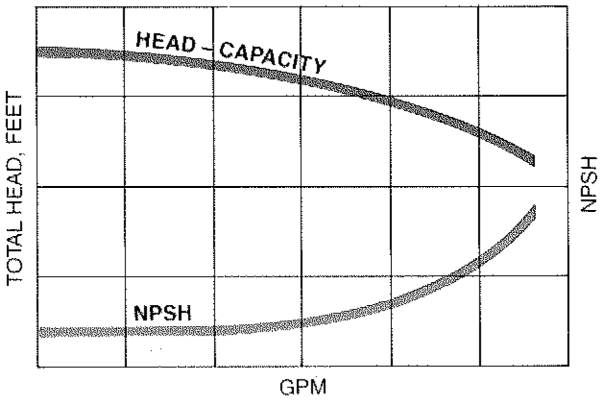

Figure 7

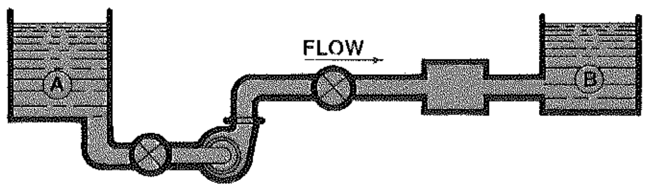

Figure 10

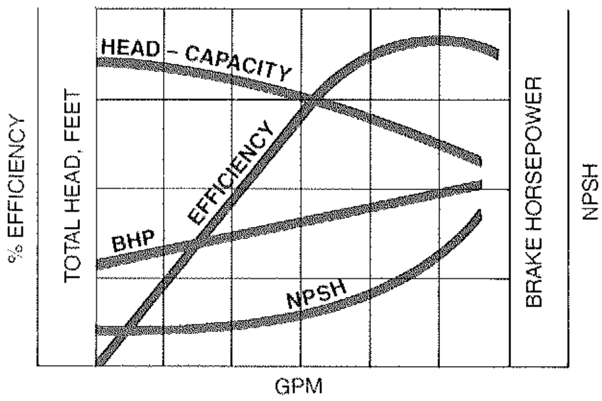

Figure 8

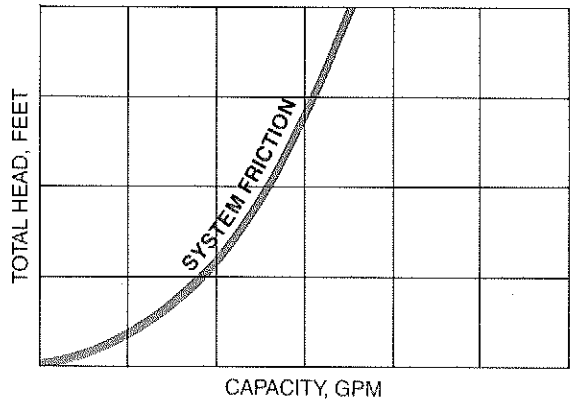

Figure 11

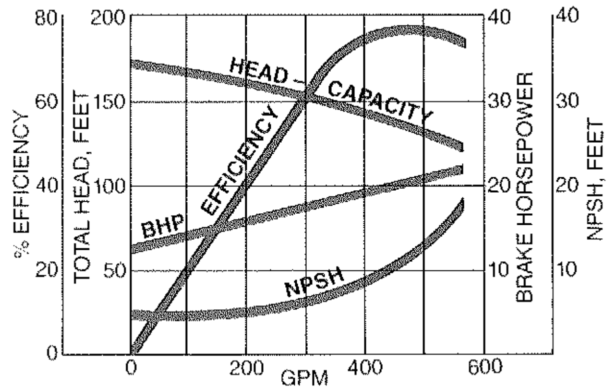

Figure 9

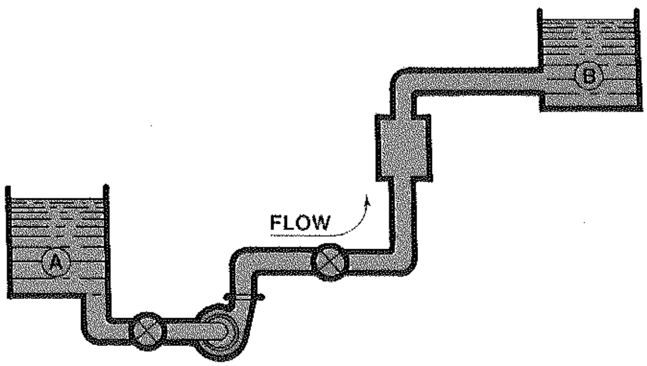

Figure 12

The new system can be expressed by the curve shown in Figure 13. The friction curve is exactly the same, because the friction loss between point A and B is the same. But in addition, we must also add a constant amount of head at any capacity — the static difference — just to get the liquid from the elevation at point A to the elevation at point B.

On this curve we were concerned with two different elevations. Instead, the pressure at point A might have been different from the pressure at point B. As an example, if we were taking suction from an open tank at point A and discharging into a closed tank under pressure at point B, it would be necessary to overcome the pressure differential between points A and B. If pressure in the vessel at point B is 10 lb. higher than pressure in the vessel at point A, enough energy must be added to overcome this 10 lb., in addition to the friction loss between points A and B.

Many such system variations can exist. In every case, as we are attempting to move the liquid from A to B, there is a friction loss between the two points, and there may also be an elevation or pressure differential, or both.

In a closed system we may take a liquid from point A, move it through a system of piping, and end up at point A again. In such a system there can be no pressure differential between points A and B, since these points are the same. The only loss in the system is the friction loss. The curve for this system, like Figure 11, represents only the friction loss in the system.

An actual system curve.
After analyzing the system to determine which type it is, it is good practice to plot the curve representing its requirements for any flow rate. Let us assume, in Figure 14, the static difference between A and B is 20 ft. and friction loss in the line between A and B is 25 ft. at 200 gpm. If we double the capacity, the friction loss will quadruple, to 100 ft. at 400 gpm. We can now plot a curve representing the requirement of the system at any capacity between 0 and 400 gpm, knowing that the friction loss will be 0 at 0 gpm.

Matching pump to system.
With the system defined, it is necessary to select a centrifugal pump to deliver whatever capacity we require. Let us assume that we need to pump 425 gpm through this system.

Checking the system head curve in Figure 14, we find the pump must deliver 132 ft. head at 425 gpm. The pump performance curve developed before (Figure 9) basically meets those requirements, delivering 140 ft. head at 425 gpm. Let us now superimpose this pump's performance curve on the system head curve (Figure 14), producing Figure 15.

The point of intersection between the pump's performance curve and the system head curve represents the capacity at which the pump will operate. Since the pump does operate at this capacity, brake horsepower required to handle the capacity and npsh required for proper operation can also be read from the curve. Note that brake horsepower is based on specific gravity of cold water, 1.0. If liquid with higher or lower specific gravity is being handled, determine brake horsepower by multiplying the brake horsepower required on water by the actual specific gravity.

Variable friction loss.
Next we must consider the effect of a variable friction loss in the system. This could be introduced, for example, by a valve which may be manipulated during the pumping cycle. Variables can also be shown on the system head curve, with the system curve moving left with increased, right with decreased friction loss.

Variable static head.
Static head may also vary, for example, when the suction receiver level or discharge receiver level change. In this case, the whole system friction curve moves up or down as the pumping cycle progresses.

In the next issue, we will see the effect of speed and impeller diameter changes on pump performance.

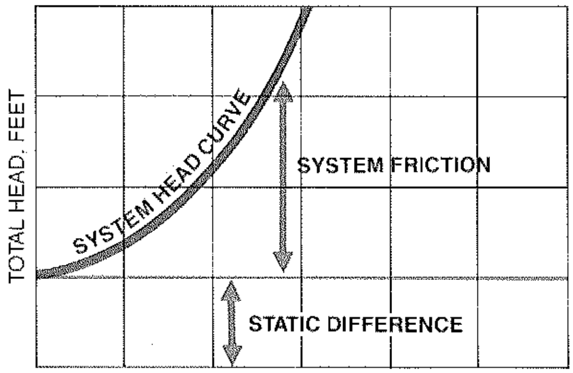

Figure 13

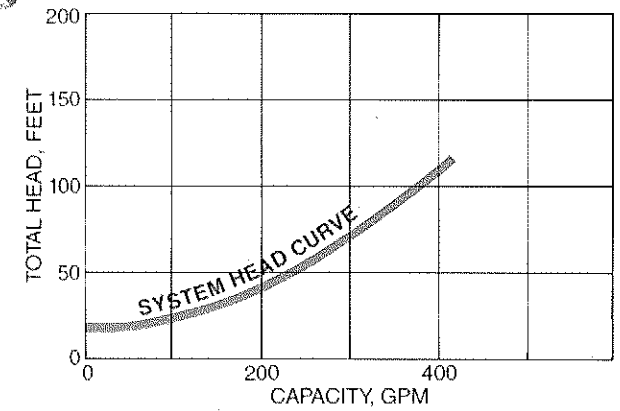

Figure 14

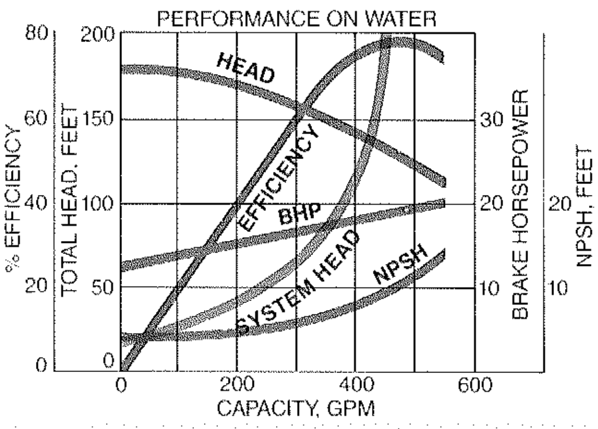

Figure 15

Hydraulic terms and basic formulas

HYDRAULICS is the study of fluids at rest or in motion.

DENSITY, sometimes referred to as specific weight, is the weight per unit volume of a substance. The density of water is 62.3 pounds per cubic foot (at sea level at 60 F).

SPECIFIC GRAVITY of a substance is the ratio of its density or specific weight to that of some standard substance. For liquids, the standard is water, usually at 60 F. Specific gravity is a pure, dimensionless number, whereas with density units must be given.

Most pump performance characteristics are determined using water, based on a specific gravity of 1.0. When pumping a different liquid, it is extremely important to know its specific gravity so the proper corrections can be made. This may be determined with a floating hydrometer. Several industries have their own scales for specific gravity, with arbitrary graduations.

In the petroleum industry, API gravity is used; 10 degrees API corresponds to a specific gravity of 1.

$$\text{Sp. gr. (relative to water at 60 F)} = \frac{141.5}{131.5 + \text{degrees API}}$$

In the chemical industry, degrees Baume are commonly used. Two scales are used, one for liquids lighter than water, one for those heavier.

$$\text{For liquids lighter than water, specific gravity} = \frac{140}{130 + \text{degrees Baume}}$$

$$\text{For liquids heavier than water, specific gravity} = \frac{145}{145 - \text{degrees Baume}}$$

PRESSURE is the force exerted per unit area of a fluid. The most common unit for designating pressure is pounds per square inch (psi). According to Pascal's principle, pressure applied to the surface of a fluid is transmitted undiminished in all directions.

ATMOSPHERIC PRESSURE (psi) is the force exerted on a unit area by the weight of the atmosphere. The pressure at sea level due to the atmosphere is 14.7 psi.

GAGE PRESSURE (psig) is a corrected pressure and is the difference between a given pressure and that of the atmosphere.

ABSOLUTE PRESSURE (psia) is the sum of gage pressure and atmospheric pressure. Psia in a perfect vacuum is 0; psia of the atmosphere at sea level is 14.7 psi (0 psig). psi + psig = psia

1 atmosphere = 14.7 psi = 34 ft. water
conversion factor = 34/14.7 = 2.31

$$\text{psi} = \frac{\text{head in feet}}{2.31} \times \text{sp. gr.}$$

VACUUM refers to pressures below atmospheric. Due to the common use of a column of mercury to measure vacuum, units are expressed in inches of mercury (14.7 psi = 30 in. Hg).

VAPOR PRESSURE of a liquid at a specified temperature is the pressure at which the liquid is in equilibrium with the atmosphere or with its vapor in a closed container. At pressures below vapor pressure at a given temperature, the liquid will start to vaporize due to the reduction in pressure at the surface of the liquid. (At 60 F, vapor pressure of water is 0.256 psi. At 212 F, it is 14.7 psi.)

Npsh guidelines for centrifugal pumps.

By W. C. Krutzsch

An intelligent determination of the npsh likely to be required by a centrifugal pump should consider not only the conditions which will exist in the system, but certain design characteristics of the pump itself. Unfortunately, the user is not ordinarily able to anticipate many of those characteristics, and the information which has been generally available to him on the subject of npsh requirements has therefore tended to limit consideration of this part of the problem to variations based only on sub-division of all pumps into a few basic types. Present day pumps cover such a wide range of capabilities that it becomes increasingly desirable to consider an approach which takes into account more of the variables likely to be encountered in both the pump and the system, provided this can be done without requiring the user to work with information which is not available to him. This article provides such a method.

Because of the seriousness of the problems which arise from inadequate npsh, and considering the frequency with which they still seem to occur, one might surmise that it would be expedient to make certain that the system provides an adequate margin over the npsh required by the pump. In many cases, fortunately, this is easily done, but in others it is not so simple. Complications which may arise include calculation of the margin required to cover transient conditions, and determination of the economic impact of increasing the amount of npsh to be provided by the system. This can be substantial whenever the source of the liquid is in a closed vessel at vapor pressure, under which circumstances npsh can be increased only by raising the liquid level or lowering the pump level, and either of these actions is likely to increase construction costs.

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Conversely, of course, failure to insure that the system will provide at least the npsh required by the pump under all conditions of sustained operation is very likely to lead to abnormal wear or premature failure of the pump, with increased cost of maintenance and downtime.

However well all of this may be understood, and however skilled the plant designer may be, an optimum solution to this economic and operational problem is obviously dependent entirely on the reliability of the information available on the npsh requirements of the pump. During the evaluation stage of the equipment purchasing process this information is provided by the manufacturer, who is therefore responsible for its accuracy. In the plant design stage, however, one must either obtain preliminary values from probable suppliers, or resort to some other source for preliminary information. Perhaps the best known present day source of such information is the Hydraulic Institute Standards. The material they contain on this subject has been in use for many years, and within the limits where it is applicable, has proven quite reliable. To that extent, those standards have been used as a principal reference for the material which follows.

In fact, a format emulating them has been used in this article, for the simple reason that some of this material was originally prepared as a proposed revision to those standards. Where necessary for added explanation, however, or for justification, we have inserted connecting paragraphs freely throughout the text.

With all of the foregoing serving essentially as a preamble, we can now proceed to the essence of the matter, beginning with a description of suction terms, which is

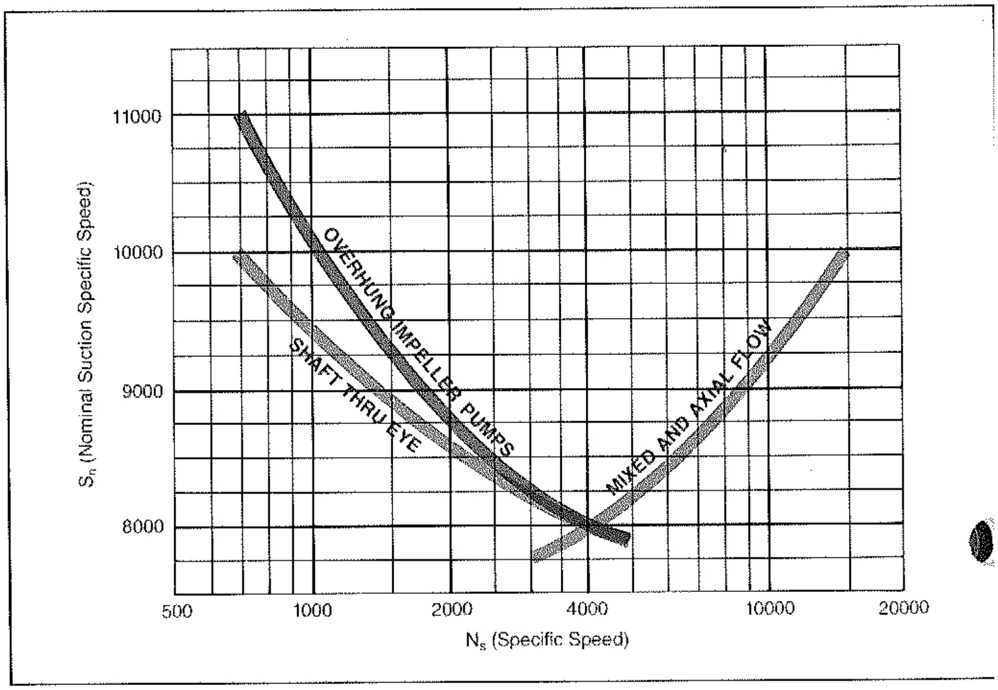


Figure 1—Relationship between specific speed and suction specific speed.

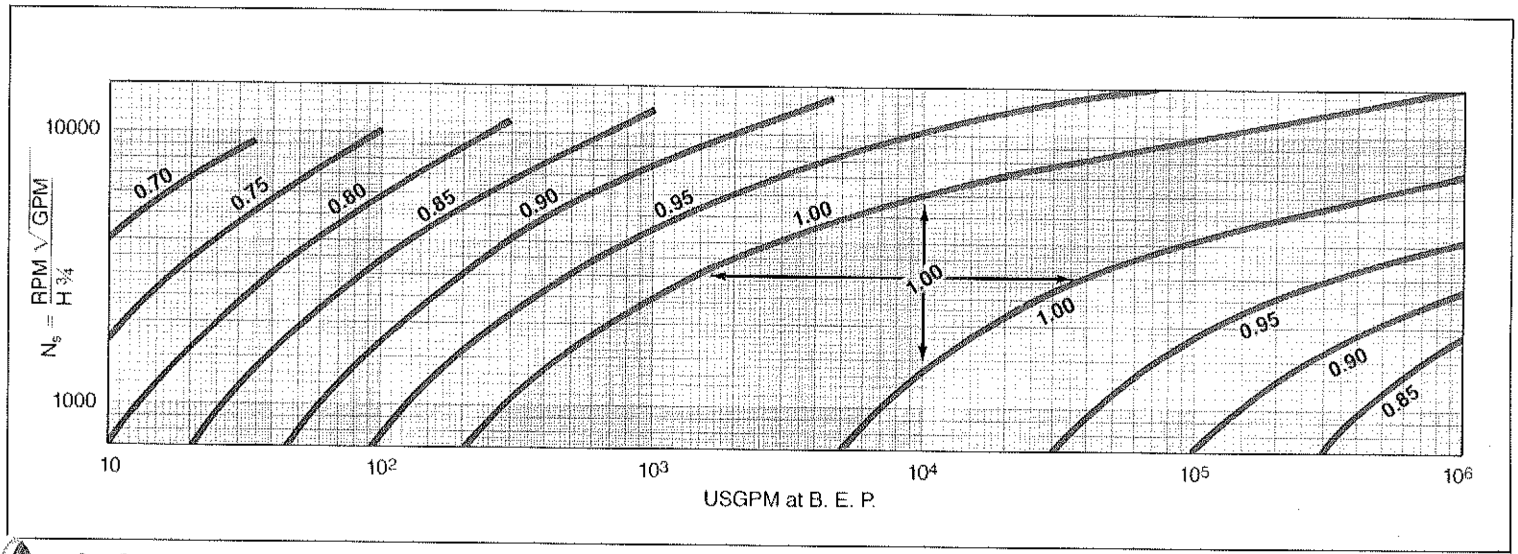


Figure 2—Correction factor K_1.

provided as a ready reference and to avoid misunderstanding in the material which follows.

Suction condition terms.

NET POSITIVE SUCTION HEAD (NPSH) is a general term for the description of suction conditions. It may be used to describe those conditions under any circumstances, and has been selected as the standard term for this purpose.

It is defined as the absolute pressure plus the velocity head, determined at the suction nozzle and corrected to datum, less the vapor pressure, all expressed in feet of liquid. (Datum elevation, for horizontal shaft pumps, is normally taken at the shaft centerline. For vertical single suction pumps it is at the first stage impeller eye, and for vertical double suction pumps at the horizontal plane thru the center of the impeller discharge.) Since the individual elements of this definition are all related to the system rather than the pump, the quantity thus defined is generally referred to as net positive suction head available.

NET POSITIVE SUCTION HEAD AVAILABLE (NPSHA) is a function of the geometry of the system, the rate of flow, and the condition of the liquid which will exist during operation of the pump. This is the term used by the purchaser to convey to the manufacturer the npsh under which the pump is expected to operate.

NET POSITIVE SUCTION HEAD REQUIRED (NPSHR) is determined by the design of the pump and the speed and capacity at which it will operate. This is the term used by the manufacturer to convey to the purchaser the npsh under which the pump is able to operate. Npshr must be equal to or less than npsha for satisfactory operation.

The foregoing terms, properly used, can fully define the essential relationships between the pump inlet and the system, and are entirely sufficient for that purpose. There are, however, a variety of other terms which are often used in connection with pump inlet arrangements, and some of the more common of these should also be defined, primarily to allow us to indicate the limits which need to be imposed on their usage.

TOTAL SUCTION HEAD is equivalent to the gauge pressure at the pump suction, in feet of liquid, referred to datum elevation, plus the velocity head at the point of gauge attachment. If the total suction head is negative, it is generally referred to as total suction lift.

TOTAL SUCTION HEAD and TOTAL SUCTION LIFT are terms commonly used when the liquid pumped is cold water and when the system is uncomplicated by extraneous factors such as an artificially produced vacuum. Since these terms are referred to atmosphere, rather than to absolute zero, barometric pressure variations, including those due to elevation above or below sea level, must be taken into account when converting them to equivalent values of npsh.

Submergence is a term used to indicate the depth of liquid below a free surface to the highest point of an intake pipe or the inlet to a wet pit pump. Since it is a static dimension it cannot be used in place of npsh, which includes dynamic components.

The differences between some of these terms may be further clarified by indicating mathematically how they are related to each other, as follows:

$$npsh = h_s + h_a - h_{vp}$$

where

h_s = total suction head
h_a = atmospheric pressure
h_{vp} = vapor pressure,

and all terms are expressed in feet of liquid.

Numerical examples for calculation of npsh may be found in the Hydraulic Institute Standards, and since the inclusion of similar examples would not add to the understanding of the new material presented here, none are provided.

Suction specific speed.
The suction, or inlet, characteristics of centrifugal pumps are generally correlated by an index number, known as the suction specific speed, and defined in the case of single suction pumps as

$$S = \frac{\text{rpm}\sqrt{\text{gpm}}}{(\text{npshr})^{3/4}}$$

or for double suction pumps, or multistage pumps with double suction first stage impellers,

$$S = \frac{\text{rpm}\sqrt{\text{gpm}/2}}{(\text{npshr})^{3/4}}$$

Numerical values of suction specific speed can range from a low of approximately 5,000 for very small pumps to a high near 15,000 for pumps having essentially conventional impellers, and to extreme values as high as 60,000 for pumps with inducers or very specialized impeller designs. This presentation is limited to conventional values of suction specific speed, which are provided to enable the user to determine pump operating speed where values of npsh are known, or vice versa.

Determination of suction specific speed.
Achievable values of S are affected by both pump design and pump application. Recommended limits are given by Figures 1 thru 4, which take into account major design and application variables, but which have been constructed in such a manner that detailed information on the pump design is not required to establish recommended limiting values of S.

Values obtained from these figures are valid only for operation of the pump at or near its point of best efficiency, and should be readily obtainable in well designed pumps. They are not intended to be, and must not be construed as, theoretical limits. Many successful applications have been made at S values higher than those indicated by these figures, and where the characteristics of the pump are based on the manufacturer's experience and test data, the limits shown may be exceeded.

All values of S obtained from Figures 1 thru 4, and corresponding values of npsh determined from Figure 5, are for clear, cold water, and freedom from cavitation damage when the pump is operated at or near its best efficiency point. Operation of a pump at a capacity far removed from the best efficiency point, and particularly at very low rates of flow, may result in cavitation damage almost regardless of the level of npsh available and such operation should generally be prohibited.

For pumps which will handle hot water or hydrocarbon liquids, values of npshr may be determined from information contained in this article, and subsequently modified as provided in the Hydraulic Institute Standards, 13th Edition, or in the Pump Handbook, published by McGraw-Hill.

Since with this method nominal values of suction specific speed are plotted against pump specific speed, it is necessary to start by first establishing the latter. This is calculated from the equation

$$N_s = \frac{\text{rpm}\sqrt{\text{gpm}}}{H^{3/4}}$$

where H is the total head in feet of liquid in the case of single stage pumps, or the total head of the first stage for multistage pumps. Note that for double suction pumps, or multistage pumps with double suction first stage impellers, specific speed is calculated using the full pump capacity, not one-half the capacity as in the computation for suction specific speed.

From the value of specific speed thus obtained, determine a nominal value S_N of suction specific speed from Figure 1, and a corrected value S determined from the expression

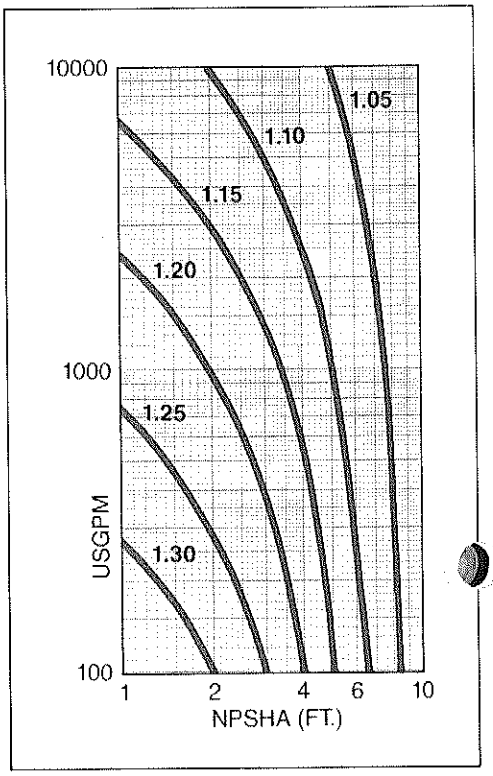


Figure 3—Correction factor K_2.

$$S = S_N \times K_1 \times K_2 \times K_3,$$

in which K_1, K_2, and K_3 are obtained from Figures 2, 3, and 4 respectively.

Determination of npshr.
With the S value determined by means of Figures 1 thru 4, and the capacity to be handled by the pump, it is possible by use of Figure 5, which is simply a graphical representation of the equation for suction specific speed, to establish suitable combinations of npshr and rpm. Before proceeding with examples of this procedure, however, we would like to offer some additional comments on these figures.

For many pumps, perhaps even a majority, Figure 1 alone is sufficient to establish the relationship between specific speed N_s and suction specific speed S. Obviously, this is the case whenever the value of all three correction factors is unity, and this situation will frequently be encountered. Thus, the relationships given in this figure must be based on the best available information, and we have tried to insure that they are.

For the most part, Figure 1 has been constructed from information contained in the Hydraulic Institute Standards, 13th Edition, Figures 57 thru 60, replotted in the format used here. The resulting curves were then modified slightly on the basis of information obtained by the Institute from several of its member companies, and finally, were checked against criteria established by Worthington over a period of years and through analysis of literally hundreds of tests, in order to determine that the S value shown can be obtained without compromising other essential operating characteristics.

Particular attention was paid to the limitations of S value which must be considered in order to avoid cavitation damage due to internal recirculation at the impeller inlet when the pump is operated at less than design capacity.* This aspect of impeller inlet design has become increasingly important in recent years as the limits of total head per stage and brake horsepower per stage have been pushed to ever increasing values, particularly on systems where pump capacity is subject to substantial variation, as in the case of nuclear reactor feed pumps.

It was also necessary in the construction of Figure 1 to anticipate the effect of the curves for correction factors, the absence of which in the past tended to depress published values of achievable suction specific speed, since any detrimental effects of unusual configurations or applications had to be "built-in" to values which could not be systematically altered to reflect these unusual conditions. For this reason, the values of S_N shown in Figure 1 are generally a bit higher than the comparable S values which could be determined from existing standards.

Figure 2 is essentially a chart of scale factor, and like all of the others, is an attempt to show, in conjunction with Figure 1, of course, values most likely to be encountered with present day pumps readily available as regular commercial products. This is perhaps a good place to re-emphasize that what we are presenting here is not an expression of theoretically, or even practically, achievable limits, but an evaluated determination of what is reasonably to be expected. Thus, the relatively low values of K_1 shown in Figure 2 for low capacities are not intended to imply that small pumps cannot be designed for the same S values as moderately large ones, but rather that they are not designed that way because there is ordinarily no need for them to be, since their npsh requirements are quite low in any event. Conversely, the relatively low values at high capacities reflect the tendency on the part of manufacturers to be a bit conservative in the design of very large units because of the increased economic seriousness of cavitation in these big pumps should it occur.

Figure 3 clearly is intended to show that when a pump is to operate at very low levels of npsha, particularly if it is relatively small, it can safely be designed for considerably higher S values than would otherwise be the case. The reason for this is that the energy available for release from the liquid during cavitation is directly proportional to the npsha, and at these very low levels, is insufficient to cause physical damage to most pump materials even when cavitation does occur. Because of this, the capacity of small condensate pumps can be automatically regulated by "submergence control" and many systems employing this concept have been built. These pumps are frequently provided with special first stage impellers which are designed for considerably higher S values than could be tolerated for npsh levels above 10 feet.

Finally, we have tried to provide in Figure 4 a means of taking into account the detrimental effect on achievable S values

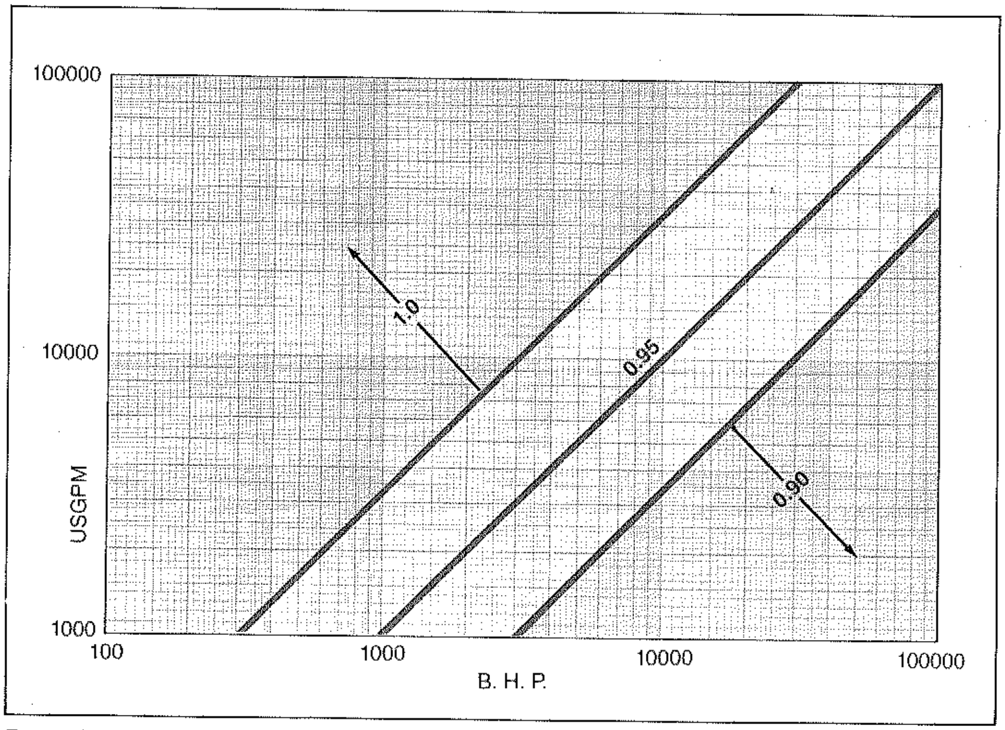


Figure 4—Correction factor K_3.

For additional information on internal recirculation see "Coping with Pump Progress," Bush, Fraser, and Karassik, Pump World, Vol. 1, No. 1, 1975, and Vol. 2, No. 1, 1976.

of having to provide an abnormally large shaft through the impeller eye, ordinarily for the purpose of carrying high shaft horsepower. We have chosen to plot this correction factor as a function of horsepower rather than shaft diameter since the former value is reasonably determinable by the user, whereas the latter can only be established by the pump designer.

Having now indicated the purposes for the inclusion of each of these figures, we can proceed to show how they are used in two specific examples.

Example No. 1.
A double suction pump is required for a rated capacity of 5,000 gpm, 200 feet total head, and is to be installed in a cold water system which will provide 15 feet npsha. What is the maximum speed at which this pump should be operated?

Since pump speed must be known to compute specific speed, which is the first step in the solution of this problem, it is necessary to start by assuming some value. A good basis for such an assumption would be a review of manufacturers' catalogs which would probably show pumps for this head and capacity rated at speeds of approximately 1200 rpm and 1800 rpm, and possibly at equivalent 50 Hz speeds of 1000 rpm and 1500 rpm. Since the slower speeds would represent more conservative (and probably more costly) pump selections, the higher available speeds would normally be tried first. In this case, 1800 rpm may be assumed.

Then, specific speed

$$N_s = \frac{1800\sqrt{5000}}{(200)^{3/4}} = 2393$$

and from the curve on Figure 1 for pumps with a shaft thru the impeller eye, the nominal suction specific speed S_N is determined as approximately 8500.

Checking the correction factors, K_1 is found to be 1.0, the same as K_2. To determine K_3, we must know the approximate pump efficiency in order to calculate the brake horsepower. Again, by reference to manufacturers' catalogs we find that 85% is a reasonable value, from which

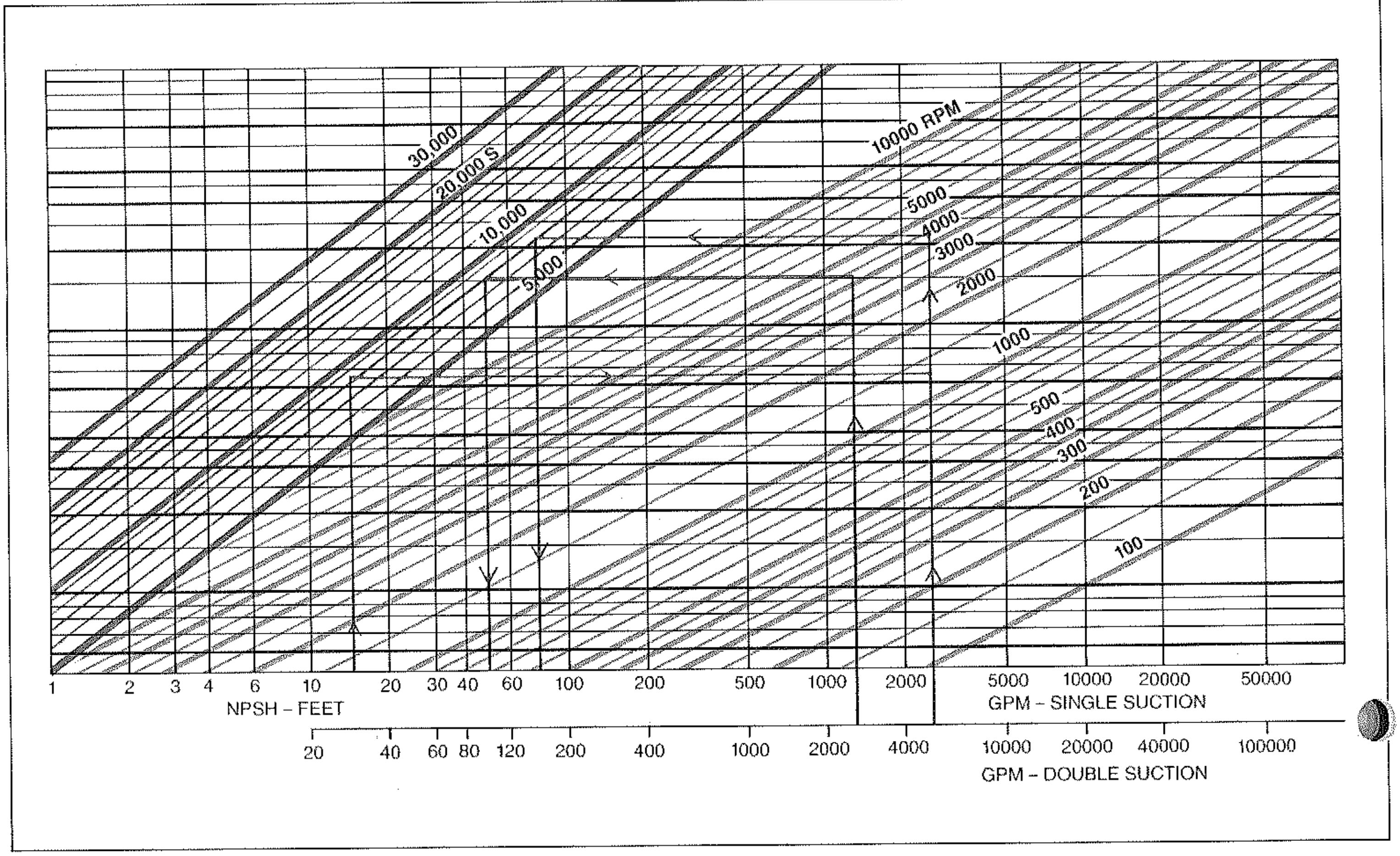


Figure 5—Suction specific speed chart.

$$\text{bhp} = \frac{\text{gpm} \times \text{Head} \times \text{Sp Gr}}{3960 \times \text{Efficiency}}$$

$$= \frac{5000 \times 200 \times 1.0}{3960 \times 0.85} = 297$$

From Figure 4, K_3 is also 1.0, and

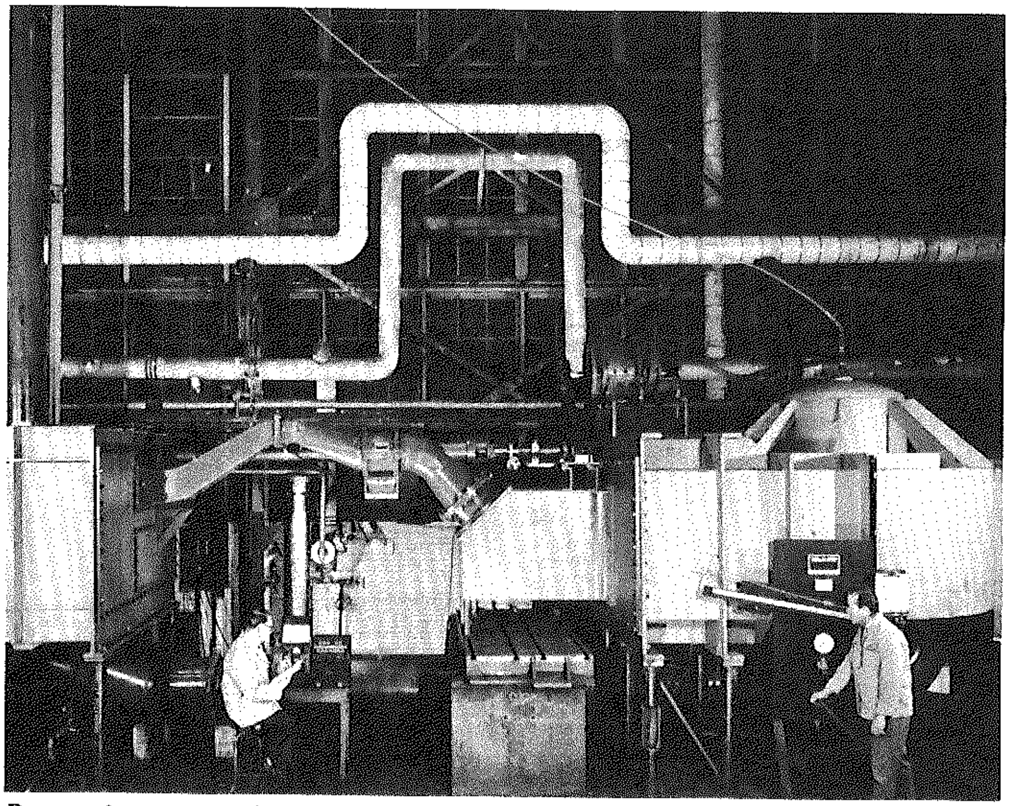
$$S = S_N = 8500$$

Now enter Figure 5 at 15 feet on the npsh scale and proceed upward to the diagonal line (interpolate) representing 8500 S. At this intersection draw, or visualize, a horizontal line. Now enter Figure 5 at 5000 gpm on the double suction scale and proceed upward to an intersection with the previously established horizontal line. This intersection falls on the system of sloping lines at a point corresponding to 1300 rpm, which is the maximum speed recommended for this application.

Had the lowest reasonable value of pump speed, 1000 rpm, been assumed, the results would have been only slightly different. Specifically, N_S would have been 1329, S_N and S would have been 9150, and the rpm determined would have been 1400. Thus, with an initial difference of 80% between the low and high speeds assumed, there is a difference of less than 8% in the result. Should even this small difference happen to fall in a critical area, such as straddling a normal motor speed, a final check could be made using an intermediate speed, but since the entire procedure is approximate at best, this degree of refinement is seldom warranted. The limiting rpm could reasonably be established in this case at any value from 1300 to 1400.

Example No. 2.
A multistage pump is required to handle 2500 gpm of cold water at a total head of 4000 feet, and is expected to be driven by a steam turbine at 4500 rpm. What is the npshr for pumps with single suction and double suction first stage impellers?

Reference to manufacturers' catalogs indicates that in this case the possible pump selections range from 6 stage to 8 stage units. For calculations a compromise of 7 stages may be used, resulting in a total head per stage of 571 feet, and a specific speed



Pump undergoing manufacturer's tests, including npsh.

$$N_s = \frac{4500\sqrt{2500}}{(571)^{3/4}} = 1923.$$

From Figure 1, $S_N = 8700$, K_1 is determined from Figure 2 to be 1.0, K_2 may be assumed as 1.0 subject to check later, and K_3 must be determined. Catalog values of pump efficiency are in the range of 80%, from which we determine

$$\text{bhp} = \frac{2500 \times 4000 \times 1.0}{3960 \times 0.80} = 3157.$$

From Figure 4 we find $K_3 = 0.94$ and

$$S = 0.94\ (8700) = 8178.$$

For a single suction first stage impeller, using Figure 5 in a manner similar to that indicated in the previous example but proceeding in order from capacity to rpm to S value to npsh, we find npshr would be approximately 80 feet. For a double suction first stage impeller, the solution becomes approximately 48 feet.

Confirmation that these values are proper is obtained by referring back to Figure 3 for verification that correction factor K_2 is 1.0, as assumed. If at this stage K_2 had been other than 1.0, it would have been necessary to establish a new value of S, with which a new solution of npshr could be obtained from Figure 5.

Conclusion.
While this presentation has been offered here primarily for the benefit of pump users, we recognize the possibility that it may also be of some use to pump designers as well. For that reason we wish to invite comment from all readers, and to that end this issue of *Pumpworld* includes a reader response card for your convenience. We would sincerely appreciate your critique of this article.

The economics of variable-speed pumping with speed-changing drives.

Part 1

By J. R. Bower

There are times when it is beneficial to be able to vary the speed of a centrifugal pump. You might be forced to specify the pump before system conditions can be finalized, or you might anticipate a system change at a later date. In some systems, the conditions of service are continually changing, as in water supply, process, or some automatic-response situations. If you need variable speed, there are many ways to achieve it — some very costly. It's important to define your needs carefully, and equally important to take advantage of the special characteristics of a centrifugal pump, in order to select the most cost-effective answer to your own situation.

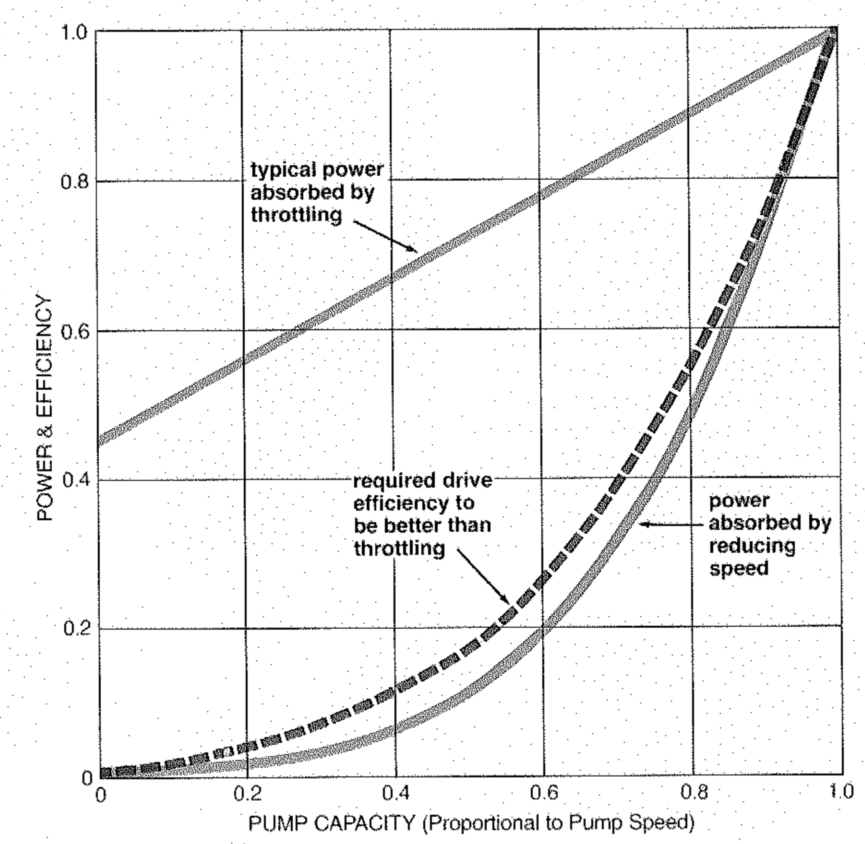


Figure 1—Comparison of capacity control by speed variation and throttling for friction head loss system.

The price per horsepower of a pump set is generally low compared to that for most other industrial equipment, with the driver representing a large proportion of the total. Variable speed control makes the drive arrangement even more costly. A fixed change of speed, such as provided by a vee-belt, increases the combined pump and driver cost by a factor ranging from 1.2 to 2.4; variable speed can increase it by a factor from 2 to 11! Obviously, specifying more than you need can be costly, so a careful definition of drive requirements should be made to determine the real need for variable drive and the most cost effective method of achieving it.

If a need for variable speed can be firmly established, however, it is helpful to remember that operating costs usually exceed the purchase price of a standard pump and motor in about six months.

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Added efficiency will in many cases justify the use of variable-speed control, as opposed to throttling the flow (see Design and Operate Your Fluid System for Improved Efficiency, Pump World, Volume I, Number I, 1975).

When do you need a speed-changing drive? Applications fall into two categories: those where a fixed speed change will do, and those where pump speed must be continuously and infinitely variable.

Fixed speed change.

A fixed speed change may be required to meet conditions of service at some point between 2- and 4-pole motor speeds, or to change speed to suit an existing, non-standard-speed prime mover. This requirement is rare; a pump can usually be found to handle the duty directly, without need for speed change. More frequently, a speed change of this nature would be used to increase the head or capacity of a given pump beyond that which it could achieve at 2-pole motor speed. There are, of course, limitations on the amount by which speed can be increased, resulting from the limits to which stress levels can be raised. A pump designed to take any substantial overspeed may be unrealistically overdesigned for standard speed applications.

Where head or capacity requirements can't be accurately figured, or if a subsequent system change is anticipated, the necessary adjustments in service conditions can often be accommodated by specifying pumps with impellers somewhat below full diameter. This allows for future replacements of a larger or smaller diameter. If the range of variation is too great, or a number of future changes are likely, then some form of speed changing drive may be necessary or desirable. In either event, the prime mover must be adequately sized to handle the maximum horsepower which will be needed. Finally, a single speed change might be selected to hold down overall noise level. In this case, a low-speed prime mover may be used with a speed-increasing drive to the pump.

Infinitely variable speed.

Infinite variation is needed on systems where conditions of service are frequently or continuously changing. Applications might include maintaining a constant pressure at a variable flow for water supply; process installations where variation of flow or pressure is required to suit the output demand; installations with automatic control in response to other variables.

The use of variable speed for capacity control can save considerable power over the use of a throttle valve. This is illustrated by Figure 1, in which the two solid line curves represent pump brake horsepower at constant speed (throttled discharge), and at variable speed for a system in which the resistance to flow is due entirely to friction loss. In such a system, pump brake-horsepower is proportional to the cube of pump speed, while capacity is directly proportional to speed.

The minimum efficiency required in a variable speed drive would be that which results in the same input power to the prime mover as for constant speed drive, and which can thus be represented as the ratio between the power at variable speed and the power at constant speed. This ratio, taken at various capacities, results in the dotted line in Figure 1. Even high loss variable speed drives are considerably more efficient than this curve requires.

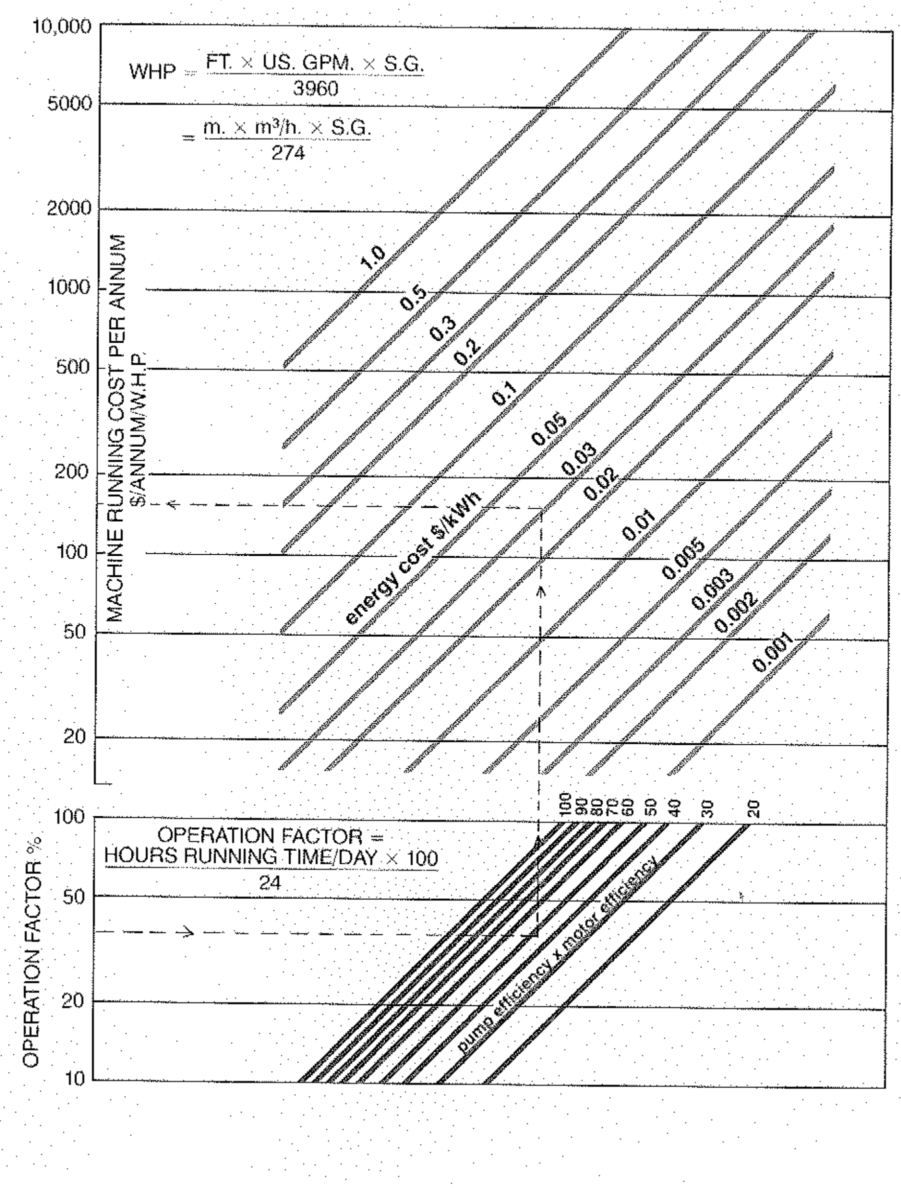

Figure 2—Pump running costs.

For an optimum selection, drives must be considered in relation to the specific characteristics of the centrifugal pump involved, and of the system to which it is being applied. High specific speed pumps, for example, may require constant or even increasing horsepower as capacity is reduced by throttling, thus creating a wider disparity between the two forms of power curves of Figure 1. On the other hand, most pump system head requirements include static as well as friction components, under which circumstances the variable speed

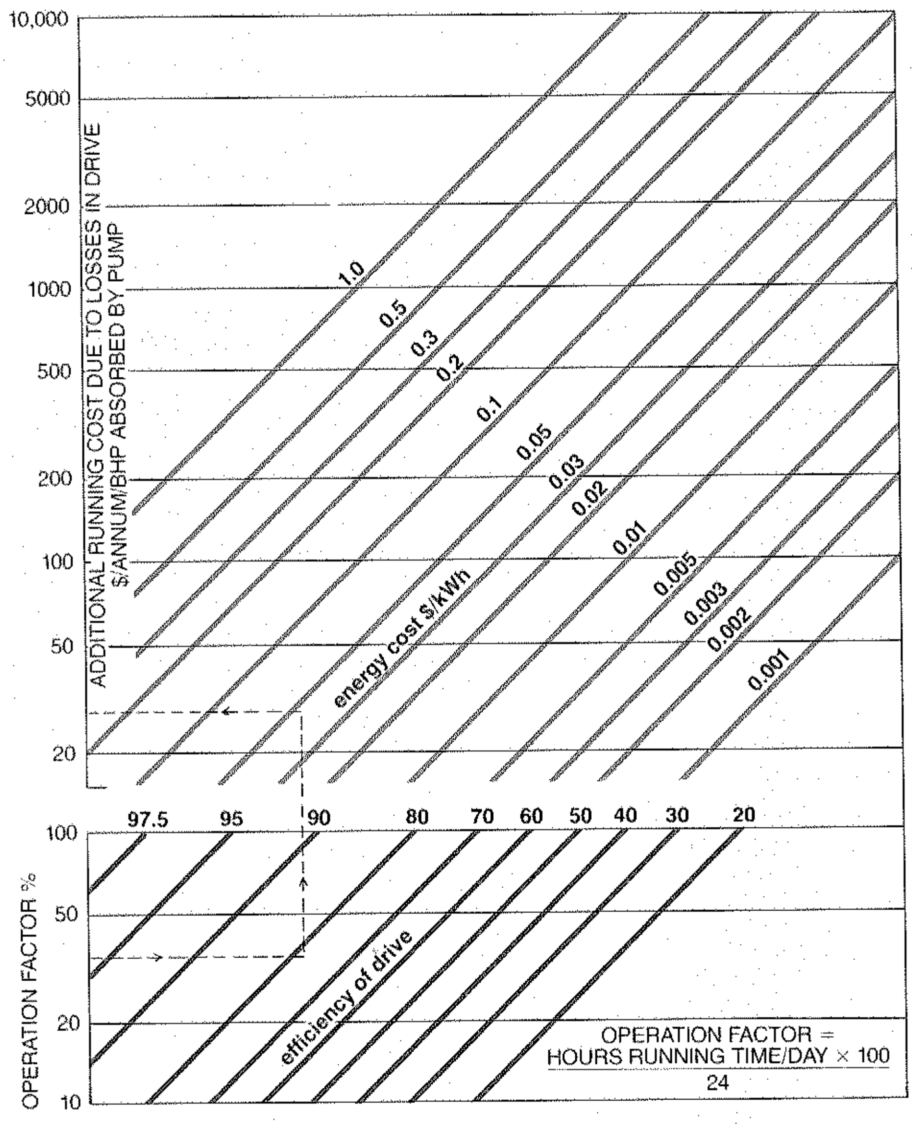

Figure 3—The additional running cost of an inefficient drive.

horsepower curve does not come down to zero at zero capacity, and the spread between the power curves is therefore diminished. Variations of this nature will affect the requirements for the variable speed drive.

Pay extra for efficiency.
Centrifugal pumps often run continuously, so operating costs are very significant. Figure 2 shows the operating cost of a motor driven pump per year per water horsepower. For example, a pump delivering 280 USGPM at 140 feet head (that is, 10 water horsepower), running 12 hours a day with 85% motor efficiency, 75% pump efficiency, and energy costing 3¢ per KWH, costs about $1,500 a year to run. This is probably considerably more than the purchase price of a pump of this size.

Figure 3 looks specifically at drive efficiency and shows the increased running costs due to losses in a drive. For example, an 80% efficiency variable-speed drive used on a 10 hp unit running 9 hours a day on energy costing 5¢ per KWH results in about an extra $300 a year operating cost.

Relating efficiency to capital expenditure, and taking the example of a pump operating 16 hours per day on 2.0¢ per KWH energy, the cost of the losses comes to about $85 per horsepower a year. Assuming machine life of 10 years and allowing for interest payment of 12% per annum throughout the period, you might justify up to about $400 additional initial cost per horsepower saved to buy a more efficient drive.

Know your power/speed relationship.
A centrifugal pump has a cubic power/speed relationship, unlike most other types of industrial equipment. This characteristic can be used to reduce the size or complexity of the drive. For a centrifugal pump, torque is proportional to speed squared; power is proportional to speed cubed. The cubic falloff in power with speed is particularly beneficial in variable-speed motors, in which the cooling from a direct-coupled fan only falls off approximately as the square of speed; thus the need for forced cooling of the variable speed drive may be eliminated. The power characteristic also means that high drive efficiency at reduced speeds is less important for a pump than for many other machines. If running time at reduced speed is only a small proportion of the total, power loss at reduced speed is minimal and low drive efficiency at low speed is not important. Nevertheless, it is important to know the operating speed or

speeds you'll most often need in order to select a drive with the optimum efficiency/speed relationship. For example, a pump running at half capacity (half speed) with only occasional demands for full speed will have a motor eight times oversized for its usual service. Even if a drive of 100% efficiency is installed between a pump and induction motor, the overall efficiency of the set will fall at reduced speed because of the decrease in efficiency of the drive motor when operated at reduced loads.

Speed ratio.
Because of the head/speed characteristic of a pump and the likely applications, a speed ratio of 5:1 will usually be the maximum required. This suits standard mechanical speed changers and is well within the range of electrical methods.

Power factor.
The importance of power factor of a motor or drive depends on whether your plant has facilities for power factor correction. If not, this should be included in the operating cost consideration.

High and low loss drive systems.
In the "low loss" variable-speed drive, efficiency remains practically constant at all speed ratios and input power falls off with output power absorbed. This category covers drives such as thyristor-controlled dc motors and mechanical variators.

The "high loss" systems such as eddy current couplings and fluid couplings control speed by allowing a degree of slip between the input and output shafts while maintaining input shaft torque equal to output shaft torque. Because of the speed difference of the input and output shaft, theoretical maximum efficiency is equal to the speed ratio: for example, 50% at half speed.

The pump's cubic power/speed relationship helps in reducing losses. Figure 4 shows the pump power absorbed and corresponding input to the coupling for a "high loss" drive. The losses, dissipated as heat in the coupling, are a maximum at .67 speed ratio of a value of 15% of full load power. At speed ratios approaching 1, losses are quite small. In practice, of course, friction or resistance losses further reduce efficiency. Speed variation of induction motors by voltage control is essentially the same principal as introducing slip to reduce the speed. The resulting efficiency is similar to eddy current or fluid couplings.

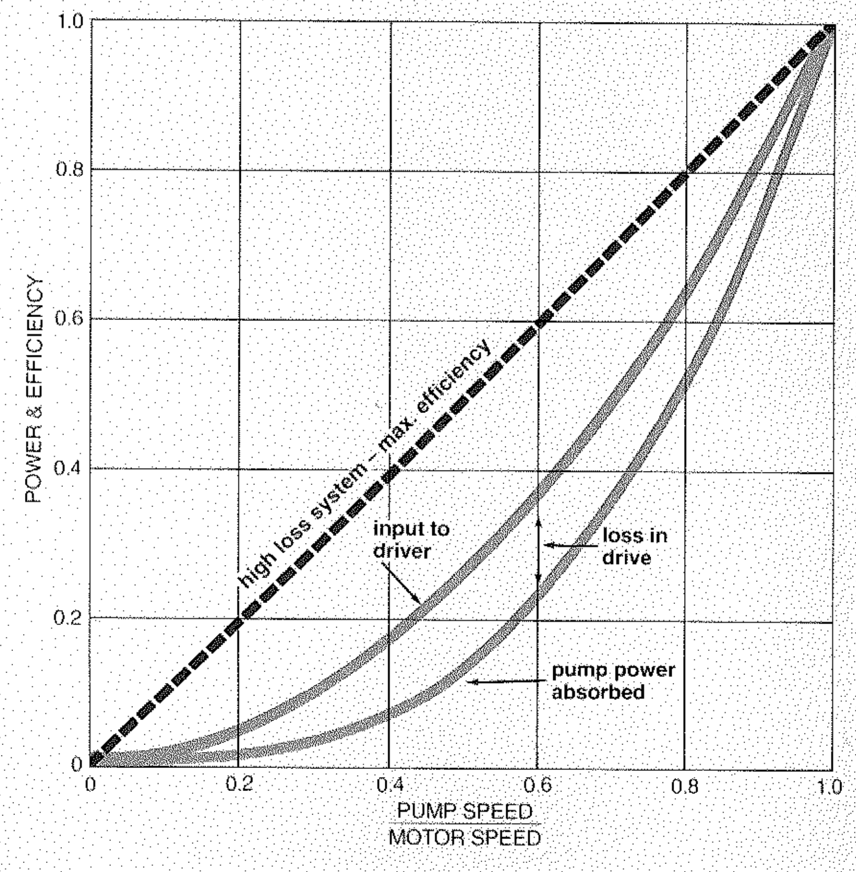

Figure 4—High loss drive characteristics related to centrifugal pumps.

Purchase price.
Pump costs per horsepower required tend to be low compared with costs of other industrial equipment, and as a result the cost of a variable-speed drive is high in relation to the overall purchase price. In addition, the use of a variable speed drive will in most cases prevent the use of the lowest cost — close-coupled — configuration, and often results in extra costs for base plate and guards. A variable speed drive can increase the pumping unit cost by a factor of 11 over that of a direct-connected pump and motor.

In the next issue, Part II will look at the relative prices and merits of many of the available drives.

Next issue: Part II *Types of drives, ranges and characteristics.*

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