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Pump Selection
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Suction and Discharge Conditions

SUCTION CONDITIONS

More pumping installations fail because of poor suction conditions than from any other single cause. A centrifugal pump has no capability to "suck" from a lower level, such as a well, unless it is initially primed and all the air is removed. Whereas a reciprocating pump is capable of self priming, providing the plunger and valves are tight. A centrifugal pump is a kinetic energy machine designed to accelerate a volume of water from a low to a high velocity, and to convert this velocity into developed head at the pump discharge flange. The head developed depends upon the peripheral velocity of the impeller and is expressed as

$$H^{*} \text{ (approximately)} = \frac{U^2}{2g}$$

where: H is total head at zero flow developed by the pump impeller in feet (same as shutoff head [SOH]).
U is velocity at the periphery of the impeller in feet per second.
g is acceleration due to gravity, 32.2 ft per sec^2.

Net Positive Suction Head (NPSH)

For a centrifugal pump to operate, the liquid must enter the eye of the impeller under pressure, usually atmospheric pressure, referred to as Net Positive Suction Head (*NPSH*). It is important to realize that there are two values of NPSH: the *available NPSH* which depends on the location and design of the intake system and can be calculated by the Engineer; and the *required NPSH*, determined by manufacturers bench scale

* This value in actual practice can be higher or lower than calculated by the formula, depending upon the type of impeller used.

tests. The required *NPSH* is the suction head required at the inlet of the impeller to ensure that the liquid will not boil under the reduced pressure conditions and the impeller will operate smoothly without cavitation. It is essential that the *available NPSH* exceeds the *required NPSH* with a reasonable margin of safety, at least two to three feet and more if possible. See Figure 1 and Figure 2.

Available NPSH

$$NPSH_{(available)} = H_{abso} + H_s - H_f - H_{vp}$$

where: $NPSH_{(available)}$ is Net Positive Suction Head measured in feet.

H_{abso} is the absolute pressure on the surface of the liquid in the suction well measured in feet.

H_s is the static elevation of the liquid above the centreline of the pump (on vertical turbine pumps to the entrance eye of the first stage impeller) expressed in feet. If the liquid level is below the pump centre line, H_s is a minus quantity.

H_f is the friction head and entrance losses in the suction piping expressed in feet.

H_{vp} is the absolute vapour pressure of fluid at the pumping temperature expressed in feet of fluid.

A "standard atmosphere" at sea level is equivalent to

1 atmosphere = 14.7 psi
= 760 mm mercury
= 29.92 in. of mercury
= 33.93 ft of water

With changes in altitude the "standard atmosphere" is modified.

Storms will also cause the atmospheric pressure to drop and 26 in. mercury (29.5 ft water) at sea level during a storm is not uncommon. This is equivalent to only 87% of the standard atmosphere (for practical purposes, assume 85%).

The vapour pressure of water increases with temperature which also reduces the available pressure at the pump suction.

EXAMPLE

A 3000 USgpm vertical turbine pump is located 4000 ft above sea level and is pumping water at a maximum temperature of 90°F. The suction bell is 24 in. in diameter, reducing to 12 in. in diameter at the first bowl assembly. The water level is never less than 8 ft above the first stage impeller. What is the available *NPSH* under the worst conditions?

TABLE 1. *Atmospheric Pressure at Various Altitudes*[6]

Altitude in Feet	*Barometer Reading*		*Atmospheric Pressure*	
	in. Hg	*mm Hg*	*psia*	*ft of water*
−1000	31.0	788	15.2	35.2
sea level	29.9	760	14.7	33.9
+1000	28.9	734	14.2	32.8
2000	27.8	706	13.7	31.5
3000	26.8	681	13.2	30.4
4000	25.8	655	12.7	29.2
5000	24.9	633	12.2	28.2
6000	24.0	610	11.8	27.2
7000	23.1	587	11.3	26.2
8000	22.2	564	10.9	25.2
9000	21.4	544	10.5	24.3
10000	20.6	523	10.1	23.4

TABLE 2. *Properties of Water*[5][9]

Temperature °F	*Absolute Vapour Pressure*		*Specific Weight lb/cu ft*	*Specific Gravity**	*Absolute Viscosity (Centipoises)*
	psia	*ft of water*			
32	0.088	0.20	62.42	1.0016	1.79
40	0.122	0.28	62.43	1.0018	1.54
50	0.178	0.41	62.41	1.0015	1.31
60	0.256	0.59	62.37	1.0008	1.12
70	0.363	0.89	62.30	0.9998	0.98
80	0.507	1.2	62.22	0.9984	0.86
90	0.698	1.6	62.12	0.9968	0.81
100	0.949	2.2	62.00	0.9949	0.77
110	1.275	3.0	61.86	0.9927	0.62
120	1.693	3.9	61.71	0.9903	0.56
130	2.223	5.0	61.56	0.9878	0.51
140	2.889	6.8	61.38	0.9850	0.47
150	3.718	8.8	61.20	0.9821	0.43
160	4.741	11.2	61.01	0.9790	0.40
170	5.993	14.2	60.79	0.9755	0.37
180	7.511	17.8	60.57	0.9720	0.35
190	9.340	22.3	60.35	0.9684	0.32
200	11.526	27.6	60.13	0.9649	0.31
210	14.123	33.9	59.88	0.9609	0.29

* Refers to water at 68°F, weighing 62.318 lb/cu ft, and having specific gravity of 1.000.

$$NPSH_{(available)} = H_{abso} + H_s - H_f - H_{vp}$$

where: H_{abso} at 4000 ft elevation is 29.2 ft.
under storm conditions (29.2×0.85) is 24.8 ft.
H_s is 8.0 ft.

H_f for a suction bell $\left[0.1\left(\frac{V^2}{2g}\right)\right]$ is 0.115 ft.

H_{vp} at 90°F is 1.6 ft.

Therefore:

$$\begin{aligned} NPSH_{(available)} &= 24.8 + 8.0 - 0.115 - 1.6 \\ &= 31.1 \text{ ft} \end{aligned}$$

For this installation, the required *NPSH* of the selected pump should not exceed 28 ft. It should be noted that the required *NPSH* increases as the capacity of the pump increases beyond the normal operating range (see Figure 3).

Cavitation

Cavitation is defined as the formation of cavities beneath the back surface of an impeller vane and the liquid normally in contact with it.
It can be caused in a centrifugal pump—

1. By the impeller vane travelling faster, at higher rpm, than the liquid can keep up with it.
2. By a restricted suction. (Hence, never throttle the suction of a centrifugal pump.)
3. When the *required NPSH* is equal to or greater than the *available NPSH*.
4. When the specific speed is too high for optimum design parameters.
5. When the temperature of the liquid is too high for the suction conditions.

The cavity consists of a partial vacuum, gradually being filled with vapour as the liquid at the interface boils at the reduced pressure in the cavity. As the cavity moves along the underside of the vane towards the outer circumference of the impeller, the pressure in the surrounding liquid increases and the cavity collapses against the impeller vane with considerable force. A pump which is cavitating can usually be detected by the noise inside the casing, but this is not always the case. When a pump

impeller is examined, the evidence that cavitation has occurred will be deep pitting and general erosion on the underside of the vanes some distance from the impeller inlet. If a pump is cavitating due to a temporary upset in the system, the cavitation can sometimes be reduced by allowing a small amount of air to enter the pump suction or by throttling the pump discharge valve. These are temporary expedients and are used only until the problem can be finally eliminated.

Intakes

In most states, provinces, and territories of the North American continent, permission must be obtained from the appropriate authorities before an intake can be installed in a river or lake. Not only are the health authorities concerned, but also the departments responsible for navigable watercourses. In some cases, the procedures are complex, particularly if the waterway is extensively used for navigation. Special protection of the intake structures are often necessary to minimize the possibilities of damage from ships' keels and anchors. Whenever a plant design is contemplated, an early approach to the governing authorities will frequently save many frustrations and much wasted effort.

Intake Sump

Intake sump designs have changed considerably during recent years. Reference should be made to *Hydraulic Institute Standards.*[2]
Complicated baffled walled designs are no longer favoured since they tend to cause vortexing. However, it is important to ensure that one side of the suction bell is almost touching one wall of the pump chamber and that the bottom opening is reasonably close to the floor in accordance with the recommendations of the Hydraulic Institute. Additional side clearance is necessary for vertical turbine pumps, particularly if they have a deep setting and small diameter columns, since the lower extremities of the pump column will gyrate. If this movement is restricted by rubbing against the wall, bearing problems may develop. See Figure 4.

Protective Screens

Protective screens should always be provided whenever there is possibility of suspended or floating debris entering the pump suction. A ball of discarded electrician's insulation tape or a roll of plastic foil can be particularly damaging to the bowl assembly of a multistage vertical turbine. However, wire screens bolted or welded directly on to the suction bowl as protection devices are not recommended. They can cause serious suction problems if they become plugged, and there is always the possibility that they may corrode, fail, and be drawn into the pump suction, causing the damage they were designed to prevent.

Fish Screens

Fish screens are also mandatory in many states and provinces to protect the juvenile salmon and trout. Specific requirements relating to the watercourse must be obtained from the appropriate government departments. Stationary screens with clear openings of 0.1-in. or less are frequently required, with approach velocities not exceeding 0.1 ft per second. However, in certain cases velocities of 0.4 ft per second are acceptable, depending on the location of the intake and the fish population. Self-cleaning mechanical screens are frequently used wherever there is a possibility of continual obstruction from debris.

A level differential cell should be installed with sensing elements on either side of the screens to measure the head loss and to provide an alarm when abnormal differential head conditions occur due to plugged screens. A low-level cut-out switch should also be installed on the pump side of the screen to stop the pumps when there is insufficient water in the pump wet well to provide adequate *available NPSH*.

Discharge Conditions

The total discharge head of a pumping installation consists of a static head or static lift and a friction head or dynamic head.

Static Head

The static head is measured from the surface of the liquid in the suction well to the surface of the liquid at the discharge reservoir. (See Figure 5.) Variations in terminal levels both at the suction well and at the discharge reservoir must be considered when calculating the upper and lower limits of the static head.

Friction Head

This is the head lost in overcoming pipe friction and depends on the size of pipe, smoothness of the inside surface, the number and type of fittings, orifice plates and control valves, velocity of flow, and viscosity and density of the liquid. The most up-to-date correlation of these factors is expressed in the Colebrook equation.

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{k}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right)$$

where: f is friction coefficient $2gDi/V^2$
k is a linear measure of effective roughness
D is pipe diameter
Re is Reynolds number $(DV)/v$
V is velocity in feet per second
v is kinematic viscosity of the fluid
g is gravitational constant (32.2)
i is hydraulic gradient

This equation is too cumbersome and contains too many variables to be of practical use in the above format, however, the Institution of Water Engineers in their *Manual of British Water Engineering Practice*[6] has published a nomograph (see Figure 6) entitled "Universal Pipe Friction Diagram" based on the work of Prandtl, Von Karman, Nikuradse, and Colebrook. This nomograph is sufficiently accurate for most practical purposes and is superior to the Hazen-Williams equation. Greater accuracy can be achieved with the use of this nomograph if the velocity (V) is precalculated and plugged into the nomograph together with the internal pipe diameter (D) instead of using (Q) (quantity: thousands of imperial gallons per hour) and the pipe diameter (D), which has too short a "length of sight" for accurate alignment. (See Appendix 4 for velocity tables.)

The roughness coefficient K expressed in inches is a measure of the actual roughness of the pipe surface. Recommended roughness values of K in inches are given in Table 3.

Total Head

The total head developed by the pump can be expressed by one of the following equations (see Figure 5):

PUMP WITH SUCTION LIFT

$$H = h_d + h_s + f_d + f_s + (V^2/2g)$$

PUMP WITH SUCTION HEAD

$$H = h_d - h_s + f_d + f_s + (V^2/2g)$$

where: H is total head in feet of liquid pumped when operating at the desired capacity.

h_d is static discharge head in feet, equal to the vertical distance between the pump datum and the surface of liquid in the discharge reservoir. (The datum is taken from the shaft centre line of horizontal centrifugal pumps or the entrance eye of the first stage impeller of vertical turbine pumps.)

h_s is static suction head or lift in feet equal to the vertical distance from the water surface to the pump datum. (Notice that this value is positive when operating with a suction lift and negative when operating with a suction head.)

TABLE 3. *Recommended Roughness Values*[7]

Pipe Materials	Values of "K" in inches: Good	Normal	Poor
Class 1			
Smooth materials—drawn copper, aluminum, brass, plastic, glass, fiberglass		0.0005	
Class 2			
Asbestos cement		0.001	
Class 3			
Bitumen lined cast iron		0.0015	
Cement mortar lined steel		0.0015	
Uncoated steel	0.001	0.0015	0.003
Galvanized iron	0.003	0.008	0.015
Uncoated cast iron	0.007	0.015	0.030
Class 4			
Old tuberculated water mains, with the following degrees of attack:			
Slight	0.025	0.06	0.15
Moderate	0.06	0.15	0.25
Severe	0.60	1.5	2.5
Class 5			
Woodstave pipe	0.015	0.030	0.060
Class 6			
Smooth surface precast concrete pipe in lengths over 6 feet with spigot and socket joints internally pointed.	0.003	0.006	0.015
Precast pipes with mortar squeeze at the joints		0.15	0.30
Class 7			
Gravity sewer pipes (new)	0.030	0.060	0.15
Gravity sewer pipes (dirty)	0.25	0.50	1.0

f_d is friction head loss in the discharge piping measured in feet.

f_s is friction head loss in the suction piping measured in feet.

$(V^2/2g)$ is velocity head in feet. For vertical turbine and submersible pumps, the velocity head is measured at the discharge flange. However, for booster pumps and centrifugal pumps, as shown in Figure 5, the velocity head developed by the pump is the difference between the $V^2/2g$ at the discharge flange and the $V^2/2g$ at the suction flange. That is,

$$V^2/2g = (V^2d/2g) - (V^2s/2g)$$

where $V^2d/2g$ is velocity head at the discharge flange.
$V^2s/2g$ is velocity head at the suction flange.

Since the discharge flange is usually a size smaller than the suction flange, the difference in the velocity head is always positive. Usually, it is a small percentage of the total head and is frequently erroneously neglected.

System Head Curve

The total discharge head for a municipal water supply is known as a system head curve and is plotted for various conditions of flow. A typical system head curve with two pumps operating in parallel is shown in Figure 7. In this instance, the system head curve is very flat, since only a very small portion of the total head is due to friction, and the two pumps in parallel will deliver almost double the flow of a single pump.

If, however, the system head curve is steep due to high friction losses (see Figure 8), the second pump operating in parallel with the first pump will deliver considerably less than double the original flow. The friction head loss in a pipe system is approximately proportional to the velocity squared, i.e., if the velocity is doubled, the head loss will be approximately four times the original value. This is illustrated in Figure 8, where one pump will deliver 4400 USgpm against a system head of 320 ft. If, however, two pumps are in parallel they will deliver 6500 USgpm (48% increase) against a system head of 700 ft and will absorb approximately 1440 hp compared to 446 hp for one pump only. This represents a 50% increase in flow for over three times the horse power.

Pipe Diameter

The economical or optimum pipe diameter can be determined graphically as illustrated in Figure 9.

Where:

pipeline costs = annual amortization cost of a unit length of installed line (usually 1000 ft).

pumping costs = the annual power cost for a unit length of installed line (1000 ft). (See Figure 10.*)

total cost = summation of the pipeline cost and the pumping cost.

The optimum pipe diameter is the one that has the least total cost, (in Figure 9, it is 18 in.). If the power cost increases, a larger pipe diameter would be more economical, and if pipe prices increase, a smaller pipe would be better.

It should be noted that the economical pipe diameter is selected on the basis of friction losses only and is not influenced by static head.

Having selected the optimum pipe diameter, the system head curve should be plotted—head against flow. The static head is the system head curve datum at zero flow. If the suction well level is subjected to seasonal variations which will affect the system head curve, a parallel curve should be shown for the system head curve under each set of conditions. Likewise, if the pumps are to discharge into an elevated tank, with upper and lower level limitations, this too should be indicated on the system head curve to ensure that all conditions of pump service are presented.

Pump Discharge Head

The pump discharge head can be specified in different ways, depending on the particular design code.

The *American Standard for Vertical Turbine Pumps*[1] defines "pump total head (H) as the bowl assembly head." This does not include the column and discharge head losses, which can be determined from graphs published in the *American Standard*.

With horizontal centrifugal pumps, it is usual to define the *Total dynamic head* (*TDH*), as "the difference between the elevation corresponding to the pressure at the discharge flange of the pump and the elevation corresponding to the vacuum or pressure at the suction flange of the pump, corrected to the same datum plane, plus the velocity head at the discharge flange of the pum[illegible]nus the velocity head at the suction flange of the pump."[9]

As previously mentioned, the head developed by a centrifugal pump impeller at shutoff head (*SOH*) is expressed thus:

$$H \text{ (approximately)} = U^2/2g$$

where: H = total head at zero flow, developed by each pump impeller in ft of liquid.
U = velocity at the periphery of the impeller in ft per sec.
g = acceleration due to gravity 32.2 ft per sec^2.

For multistage pumps, the total head is equal to:

$$\text{Total head (approximately)} = H \times \text{number of stages}$$

In Figure 11, all three identical pumps on the left of the diagram are operating with the same size impellers and at the same shaft speed. Notice that the heads developed by the pumps are independent of the weight of the liquid and the discharge head in feet is the same whether the pumps are handling brine with a specific gravity (SG) of 1.2, water at SG = 1.0, or gasoline at SG = 0.7.

The pressure gauge reading would be in accordance with

$$\text{psi} = \frac{\text{head in feet} \times \text{SG}}{2.31}$$

If, however, all three pumps are delivering liquids at 50 psi, as shown on the right of Figure 11, because of the difference in specific gravity of the liquids, each pump develops a different head. Therefore, if the speed of all three pumps is the same, the pump handling gasoline would have the largest diameter impeller and the pump handling brine would have the smallest.

* Figure 10 shows typical power costs in specific areas of British Columbia and is included in this book only to illustrate a convenient format for presenting the usually complicated schedule of costs.

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Specifications

Shaft Speed

The developed head of a centrifugal pump is proportional to the peripheral speed of the impeller. A specific developed head can be obtained either by installing a larger diameter impeller and operating it at a lower speed or by installing a smaller diameter impeller and operating it at higher speeds, so that the circumferential velocity in both cases is the same.

The smaller impeller is housed in a smaller casing or bowl assembly and can therefore be built at a lower initial capital cost than the larger, slower unit. The faster the shaft revolves, the greater the wear in the bearings, wear rings, neck bushings, sleeves, and stuffing boxes. It is reported that the rate of wear and, therefore, the maintenance costs are proportional to the shaft speed squared, so that doubling the speed may result in four times the wear. Vibration becomes more pronounced as the wear increases the running clearances, and, since vibrations occur at greater frequency with higher shaft speeds, maintenance costs are higher than those of lower speed machines. If pumps are to be purchased on the basis of the lowest bidder, higher speed pumps will invariably be the result since they are less expensive to build (but more expensive to maintain).

Ball and roller bearings are commonly used on pumps and electric motors. They are relatively inexpensive but have a limited life which is highly dependent upon the degree of skill exercised in their initial installation. Their principal advantages are low cost and antifriction properties, since the rolling contacts between the stationary and rotating elements produce very little friction. They are, however, somewhat sensitive to damage by shock loads, changes in speed, ingress of foreign matter, and poor lubrication. The pinpoint or knife edge contacts are very highly stressed under load and can readily be damaged, resulting in increased wear and shaft vibration. Wherever longevity, high rotational speeds in excess of 1800 rpm, and large horsepowers are required, oil-lubricated sleeve bearings and Kingsbury thrusts are preferred. Worn bearings result in vibration and, wherever possible, the shaft displacement should be measured and the bearings replaced if the following tolerances are exceeded.(9)

900 rpm shaft displacement, peak to peak = 0.0050 in.
1200 rpm shaft displacement, peak to peak = 0.0042 in.
1800 rpm shaft displacement, peak to peak = 0.0035 in.
3600 rpm shaft displacement, peak to peak = 0.0020 in.

The shaft tolerances should be measured at the top bearing of vertical turbine pumps and on the bearing housing(s) of horizontal pumps.

The most troublesome maintenance item of a centrifugal pump is the gland: whether a soft-packed stuffing box or a mechanical seal, it will give far less trouble if it operates at a lower shaft speed.

This is obviously an optimum balance between lower capital and higher maintenance costs as opposed to higher capital and lower maintenance costs. No doubt the balance is influenced by the current taxation laws; however, with increasing inflation, maintenance costs are rising sharply and therefore must be considered when selecting high speed pumping equipment.

Specific Speed (N_s)

The various implications and aspects of this parameter are dealt with in detail in published texts.(2, 9, 10) Specific speed is a correlation of pump capacity, head, and speed, and is a number expressed as follows:

$$\text{Specific speed } (N_s) = \frac{\text{rpm}\sqrt{\text{USgpm}}}{H^{0.75}}$$

where: rpm is shaft speed in revolutions per minute.
H is head per stage in feet.

Specific speed places an upper limit on the shaft speed for any particular combination of total head, flow, and suction conditions. Impeller form and proportions vary with specific speed as shown in Figure 12. For double suction impellers, the total flow through the pump should be divided by two in calculating the specific speed; i.e., they should be considered to be two single-suction impellers operating in parallel.

The specific speed parameter (N_s) has many uses for the pump designer. For the pump applications engineer, it serves as a useful limitation in the design of suction condition. If a pump installation is within the specific speed limitations of the Hydraulic Institute,(2) the engineer can be reasonably certain that there will be fewer problems due to poor suction conditions.(9) Reference to the charts published in the *Hydraulic Institute Standards*(2) for the upper limits of specific speed (N_s) for any given type of impeller, first stage total head developed and suction lift or suction head conditions, will dictate the upper allowable shaft speed to comply with their recommendations. This upper limit should not be exceeded.

Suction Specific Speed (S)

Specific speed (N_s) is an index number indicative of pump type, whereas the parameter known as suction specific speed (S) is essentially an index number descriptive of the suction characteristics of a given impeller. It is defined as:

$$S = \frac{\text{rpm}\sqrt{\text{USgpm}}}{NPSH^{0.75}}$$

where: rpm is shaft speed in revolutions per minute
NPSH is the *required NPSH* for satisfactory operation in feet

Note that, for double suction impellers, the flow in USgpm should be taken as one-half the total flow.

The upper limits of specific speed (N_s) and suction specific speed (S) are given in Figure 54–57 of the *Hydraulic Institute Standards*.(2)

WR^2

The expression WR^2 is used for defining the moment of inertia of a symmetrical body about a given axis, i.e., pump impellers, shafts, and rotors of electric motors. Actually, the moment of inertia is defined as $(WR^2)/g$, but is usually reduced to WR^2, where W is weight of the rotating body (in pounds); R is radius of gyration or the distance from the centre of rotation to a point at which the whole mass of the body is considered to be concentrated (in feet); and g is 32.2.

The WR^2 of a pump impeller is determined experimentally by the pump manufacturers and is given for water or other liquid. This parameter of pump design is used to calculate the required starting torque of the motor and to ensure that it is capable of accelerating the rotating mass up to synchronous speed without stalling.

Variable Speeds

Variable speed drives are becoming increasingly popular. Pump characteristics influenced by shaft speed are flow (gpm) varies directly with (rpm); head (ft) varies as the $(rpm)^2$; and horsepower (bhp) varies as the $(rpm)^3$.

To be able to reduce the rpm is analogous to having an impeller of variable diameter.

Initially, the only variable speed drives available were engines, steam and water turbines, or wound rotor motors. Later, induction motors with electromagnetic drives became available and, in the last few years, solid state variable speed controls for standard squirrel cage induction motors have been developed. The later devices are limited in their range and reduce the full load speed by approximately 20%. However, in most cases, 20% is all that is required to reduce the speed of a pump motor, since the developed head is proportional to speed squared. A reduction in rpm of 20% would result in a 36% reduction in head, which is usually all that is required to reduce the developed head to below the system head curve, with the result that the pump capacity, in relation to the system head curve, is reduced to zero.

The wound rotor motor, however, still has a place, particularly where engine-drive generators are installed to provide power to operate the pumps during periods of failure of the normal power supply. The starting current of a standard induction motor can be up to six times the normal full load current. Even with low inrush current motors and reduced voltage starting, the starting current may be as much as three times the normal full load current. However, a wound rotor motor can be started at reduced speeds below full load current; consequently, the capacity of the engine driven generator can be reduced to a nominal full load capacity machine, eliminating the necessity to purchase additional horsepower for the sole purpose of providing starting current.

Affinity Laws

The relationships between flow (gpm), head (ft), horsepower (bhp), and shaft speed (rpm), defined under the heading "Variable Speeds," are referred to as the "affinity laws" and are shown graphically in Figure 13.

Performance Curves

A typical performance curve for a vertical turbine pump is shown in Figure 3. The characteristic shapes of the curves for centrifugal, vertical turbine, and axial-flow pumps are shown in Figure 12.

It is important to select pumps with characteristic curves to suit the specific application. For example, if two centrifugal pumps are required to operate together in parallel, it is important that the head capacity curve has a uniform downward slope, as shown for the vertical turbine pumps in Figure 12. If, however, the curve is humped in the middle, as shown for the centrifugal pump, two pumps will not operate smoothly in parallel at this point on the curve, since there are two capacity values for the same head and they will continually "hunt" from one horsepower condition to another. It is also important to notice that the horsepower characteristics of a radial-vane impeller continue to increase as the head is reduced. With this type of impeller, the "run-out horsepower" will exceed the normal operating horsepower, and the motors, unless they are oversized for normal duty, will trip out on overload at low discharge heads. The vertical turbine impellers, also known as Francis-Screw impellers, have head-capacity (H-Q) curves with uniform downward slopes. They are able to run together smoothly in parallel without surging. Their horsepower reaches a peak near to the point of maximum efficiency and then slopes downward at reduced heads with the result that the motors operate at lower horsepowers as the flow increases. Axial flow impellers have their point of maximum efficiency at lower heads and have higher capacities than the other two impeller types. These pumps are frequently used on low head irrigation schemes.

When selecting the most suitable pump from a number of supplier quotations, it is advisable to plot all the curves on the same scale. Figure 14 shows three typical H-Q curves from three different manufacturers. All three pumps are in accordance with the *Standards of the National Fire Protection Association.*[8] Curve A has preferred characteristics to B or C, since it has a lower shut-off head and generally better performance characteristics to the right of the rated point of 231 ft at 1500 USgpm. This pump was selected and, when it was factory tested, the curve shown as AA in Figure 14 was the actual certified performance curve.

It must be emphasized that published characteristic performance curves are at the best only good approximations, and a manufacturer will guarantee only one or two points on the H-Q curve generally in the range of maximum efficiency.

It is almost impossible to test a pump on site in complete accordance with the requirements of the *Hydraulic Institute Standards*,[2] and any tests below the requirements of this standard are not generally regarded as acceptable. Therefore, the only way in which the owner can be certain that he is getting the performance he specified is to witness the factory test and make certain that the pumps are tested in accordance with the requirements of the Hydraulic Institute and are capable of producing the performance characteristics quoted in the specifications. If the witnessed factory tests do not come up to the standards anticipated, it is usually possible to polish the impellers or the diffuser vanes in the casing, until the required results are obtained. If the published characteristics cannot be obtained by modifications to the impeller or casings, the manufacturer should reimburse the owner accordingly. If it is inconvenient for the owner or his representative to witness the test, the owner should insist on a signed certified performance curve giving the necessary data obtained under test bed conditions.

Efficiency

The efficiency of a large installation can be a critical factor when choosing the optimum pumping units from a number of competitive quotations. This can be illustrated by the following example.

A municipal pumping station is to house three vertical turbine pumps, each capable of pumping 3000 USgpm against a 200-ft head. Three pumps will operate together in parallel for 50% of the year, and two pumps in parallel for the remaining time. The average power cost for the whole station can be taken at approximately one cent per kilowatt hour. Calculate the monetary value of one percentage point of efficiency.

From a brief look at a number of catalogues, it would appear that most manufacturers have pumps capable of this performance at efficiencies ranging from 83% to 87%. Using 85% as an efficiency datum:

$$\text{Horsepower (hp)} = \frac{3000 \times 8.35 \times 200}{33{,}000 \times 0.85} = 178.5$$

Efficiency of a 200 hp induction motor will be approximately 92% plus a further 2% for transformer and switchgear losses, so that the overall electrical efficiency will be approximately 90%.

$$\text{Kilowatts per pumping unit} = \frac{178.5 \times 0.746}{0.9} = 148 \text{ kw}$$

Total kilowatt hours per year (365 days = 8760 hours)

Three pumps in operation for 50% of the time:

$$3 \times 8760 \times 0.50 \times 148 = 1.94 \times 10^6 \text{ kwh}$$

Two pumps in operation for 50% of the time:

$$2 \times 8760 \times 0.50 \times 148 = 1.295 \times 10^6 \text{ kwh}$$

Total power consumption:

$$(1.94 \times 10^6) + (1.295 \times 10^6) = 3.235 \times 10^6 \text{ kwh per year}$$

Annual power costs at one cent per kwh = $32,350 per year

From a similar calculation, but using a pump efficiency of 84% instead of 85%, the annual power cost would amount to $32,800, an increase of $450 per year which is approximately equivalent to a capital expenditure now of $4500 based on an interest rate of 8% for 20 years (present worth value). This sum divided by three is equivalent to $1500 per pump per one percentage point of efficiency.

The comparative values of five competitive quotations are tabulated as follows:

Quotation Number	*Basic Unit Price (dollars)*	*Pump Efficiency (percent)*	*Bonus or Penalty (dollars)*	*Comparative Price (dollars)*
1	18,500	85	...	18,500
2	17,200	84	+1,500	18,700
3	19,000	86	−1,500	17,500
4	16,500	83½	+2,250	18,750
5	20,000	87	−3,000	17,000

On the basis of comparative price, Quotation 5 is the "best buy." Provided the pumps are to be tested in accordance with the *Hydraulic Institute Standards*[2] and competently witnessed, there should be no difficulty in ensuring that the pumps are capable of their specified test bed performance. However, the question as to whether or not they will continue to operate at the specified efficiency with the minimum of maintenance depends on shaft speed, pump design, workmanship, materials of construction, column diameter, lubrication, bearings, and other features. Therefore, it is essential to ensure that, if a premium price has been paid for a high efficiency pump, the efficiency can be maintained at its peak performance throughout its economic life. Otherwise, it would be better to recommend Quotation 3 or even Quotation 1. It should be noted that, in the example shown, the lowest basic unit price has the highest comparative price, assuming that, apart from power costs, all other annual operating costs are equal for the five quotations.

Stuffing Boxes

The stuffing box requires more attention during the operating life of a pump than any other single item, and a small defect can prevent the pump from performing satisfactorily. The function of the stuffing box and gland is to provide a liquid-tight seal between a rotating element (shaft) and a nonrotating element (casing). The packing consists of a soft semiplastic material which is cut in rings and fits snugly around the shaft or shaft sleeve. There is usually a slotted metal lantern ring approximately halfway down the stuffing box to permit leakage of fluid through the inner rings to provide lubrication and to reduce pressure on the outer rings of packing. Alternately, where gritty liquids are handled, clean water under pressure is forced into the stuffing box through the lantern ring to keep the abrasive grit from getting between the packing and the shaft.

Stuffing boxes always need to be lubricated, either internally by allowing water to pass up through the packing and leak to atmosphere or by forcing water, oil, or grease to the packing through the lantern ring and, at the same time, allowing a small external leakage. A gland must never be tightened to the extent that there is no leakage; otherwise the packing will quickly overheat, shrink, and fail. Excessive leakage will result in having to unpack the stuffing box and remove all the old packing. To facilitate repacking the stuffing box, it is important to ensure that the manufacturer has provided adequate clearances. In some cases, it is possible to remove the gland and use the casing pressure to push out the old packing; this greatly simplifies the task of removing the old packing. It is essential to ensure that all the old packing has been removed and there are no broken pieces jammed in between the bottom neck bushing and the shaft, since old packing can be severely abrasive to the pump shaft or sleeve.

When starting a new pump for the first time, the gland nuts should be loosened to allow an adequate leakage of water to lubricate the packings. The nuts should be tightened only by an experienced pump mechanic when the pump is running. Never tighten the gland nuts when the shaft is stopped. If the stuffing box leakage cannot be controlled to a reasonable rate after completely repacking, the problem may be due to loose or worn bearings, misalignment, or an out-of-balance impeller. If the pump has been in service for some time, the problem may be due to a worn shaft or sleeve. In spite of their obvious disadvantages, stuffing boxes are still preferred for certain installations. Pumps approved for fire protection service by the Canadian Underwriters Association are all fitted with stuffing boxes rather than mechanical seals.

Pumps installed in isolated pump stations are often fitted with packed stuffing boxes since they rarely fail completely without warning.

In many instances, however, mechanical seals have been used to considerable advantage over the old packed stuffing boxes, and, in the pumping of certain nonlubricating fluids, only mechanical seals can be used.

Mechanical Seals

Mechanical seals are available in many different designs and with many features, and the choice of the most suitable seal for a particular pump on a specific service is one for the specialist. The right mechanical seal in the right application can be an excellent investment and will frequently give trouble-free operation for many years. Most pumps can be supplied with either a stuffing box or a mechanical seal, and the choice as to which to install is usually left to the engineer. If the pump is to handle water or sewage, either system of sealing the shaft against leakage is applicable. Seals are more expensive in initial cost, but, provided the pump bearings are good and the rotating parts are well balanced and maintained, the maintenance cost of the seal will be negligible. If, however, the pumps will be handling gritty fluids and the bearings are likely to become worn with little or no preventative maintenance, then stuffing boxes are preferred. Never install mechanical seals on an old pump in the belief that they will be less troublesome than the original stuffing boxes unless the pumps have been completely overhauled with new bearings, the shaft reground if necessary, and rotating parts balanced and completely realigned. In one pumping installation, a new vertical turbine pump with a 75hp motor operating at 1800rpm, fitted with a mechanical seal valued at over $1000, failed after three days' operation. A new seal was fitted and this also failed after running for a few days. It was eventually discovered that the third line-shaft bearing down the pump column had excessive clearance. When this fault was corrected, the pump and seal worked well together for many years. Usually, when a mechanical seal fails, it is due to excessive vibration or malalignment of the pump shaft, rather than an actual seal failure. However, when a seal does fail, there is a considerable amount of high pressure leakage and care must be taken to ensure that no vulnerable items of electrical equipment are in the line of fire. It