Related Commercial Resources

CHAPTER 43

CENTRIFUGAL PUMPS

CENTRIFUGAL pumps provide the primary force to distribute and recirculate hot and chilled water in a variety of space-conditioning systems. The pump provides a predetermined flow of water to the space load terminal units or to a thermal storage chamber for release at peak loads. The effect of centrifugal pump performance on the application, control, and operation of various terminal units is discussed in Chapter 12. Other hydronic systems that use pumps include (1) condensing water circuits to cooling towers (Chapters 13 and 39), (2) water-source heat pumps (Chapter 8), (3) boiler feeds, and (4) condensate returns (Chapter 10). Boiler feed and condensate return pumps for steam boilers should be selected based on boiler manufacturer's requirements.

CONSTRUCTION FEATURES

The construction features of a typical centrifugal pump are shown in Figure 1. These features vary according to the manufacturer and the type of pump.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

Materials. Centrifugal pumps are generally available in bronze-fitted or iron-fitted construction. The choice of material depends on those parts in contact with the liquid being pumped. In bronze-fitted pumps, the impeller and wear rings (if used) are bronze, the shaft sleeve is stainless steel or bronze, and the casing is cast iron. All-bronze construction is often used in domestic water applications.

Seals. The **stuffing box** is that part of the pump where the rotating shaft enters the pump casing. To seal leaks at this point, a mechanical seal or packing is used. **Mechanical seals** are used predominantly in clean hydronic applications, either as unbalanced or balanced (for higher pressures) seals. Balanced seals are used for high-pressure applications, particulate-laden liquids, or for extended seal life at lower pressures. Inside seals operate inside the stuffing box, while outside seals have the rotating element outside the box. Pressure and temperature limitations vary with the liquid pumped and the style of seal. **Packing** is used where abrasive substances (that are not detrimental to operation) could damage mechanical seals. Some leakage at the packing gland is needed to lubricate and cool the area between the packing material and shaft. Some designs use a large seal cavity instead of a stuffing box.

Shaft sleeves protect the motor or pump shaft.

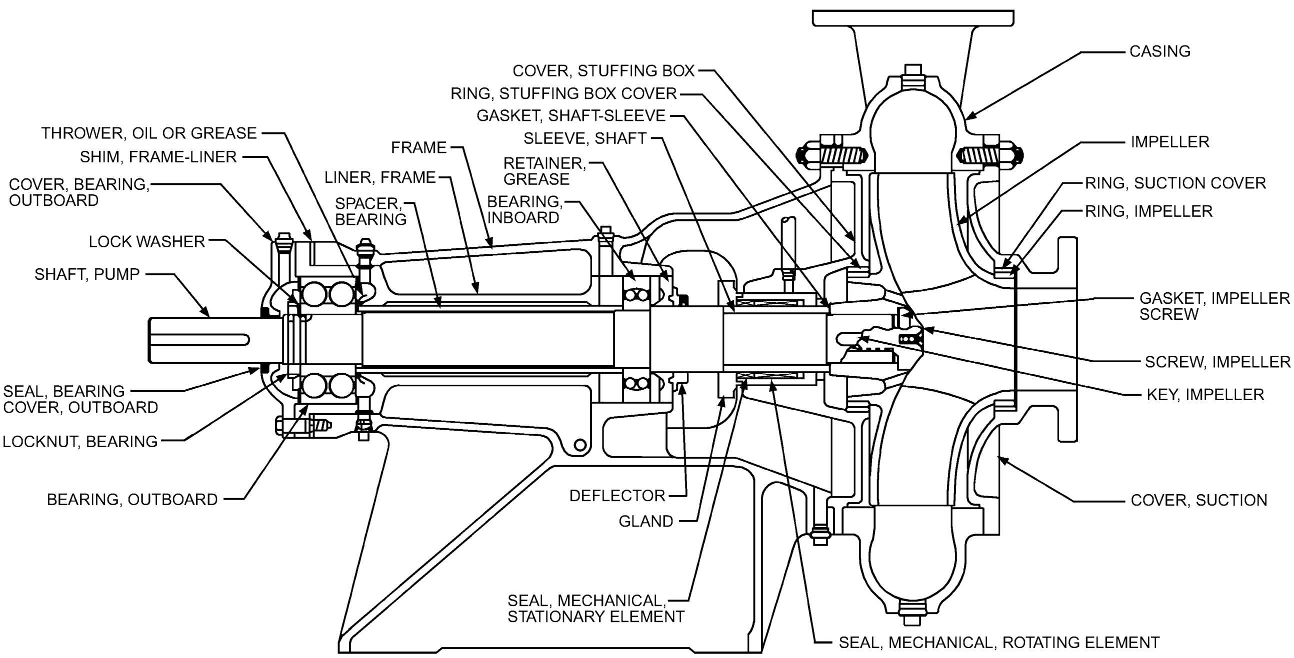


Fig. 1 Cross Section of Typical Overhung-Impeller End-Suction Pump

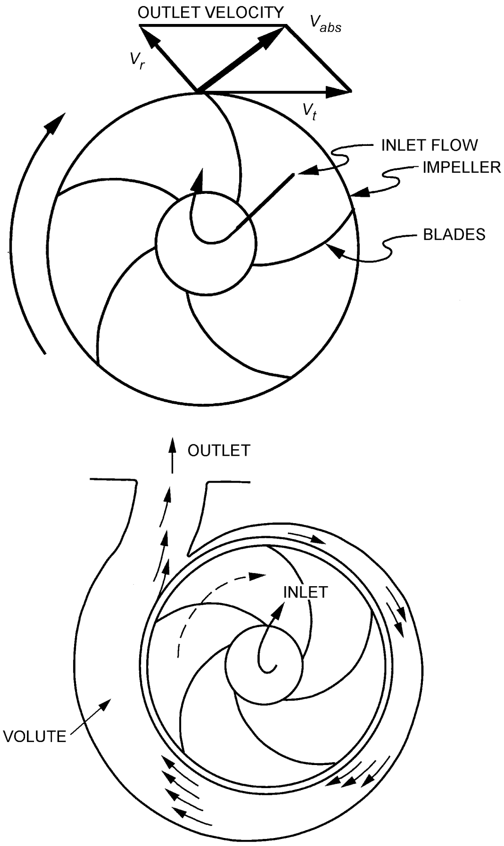


Fig. 2 Impeller and Volute Interaction

Wear rings prevent wear to the impeller and/or casing and are easily replaced when worn.

Ball bearings are most frequently used, except in low-pressure circulators, where motor and pump bearings are the sleeve type.

A **balance ring** placed on the back of a single-inlet enclosed impeller reduces the axial load, thereby decreasing the size of the thrust bearing and shaft. Double-inlet impellers are inherently axially balanced.

Normal operating speeds of motors may be selected from 600 to 3600 rpm. The pump manufacturer can help determine the optimum pump speed for a specific application by considering pump efficiency, the available pressure at the inlet to prevent cavitation, maintenance requirements, and operating cost.

PUMP OPERATION

In a centrifugal pump, an electric motor or other power source rotates the impeller at the motor's rated speed. Impeller rotation adds energy to the fluid after it is directed into the center or eye of the rotating impeller. The fluid is then acted upon by centrifugal force and rotational or tip speed force, as shown in the vector diagram in Figure 2. These two forces result in an increase in the velocity of the fluid. The pump casing is designed for the maximum conversion of velocity energy of the fluid into pressure energy, either by the uniformly increasing area of the volute or by diffuser guide vanes (when provided).

PUMP TYPES

Most centrifugal pumps used in hydronic systems are single-stage pumps with a single- or double-inlet impeller. Double-inlet pumps are generally used for high-flow applications, but either type is available with similar performance characteristics and efficiencies.

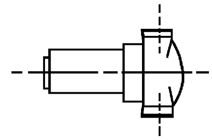


Fig. 3 Circulator Pump (Pipe-Mounted)

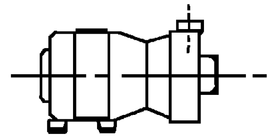


Fig. 4 Close-Coupled End-Suction Pump

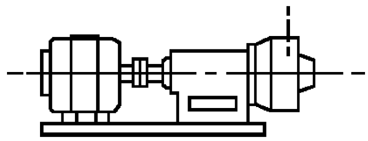


Fig. 5 Frame-Mounted End-Suction Pump on Base Plate

A centrifugal pump has either a volute or diffuser casing. Pumps with volute casings collect water from the impeller and discharge it perpendicular to the pump shaft. Casings with diffusers discharge water parallel to the pump shaft. All pumps described in this chapter have volute casings except the vertical turbine pump, which has a diffuser casing.

Pumps may be classified as close-coupled or flexible-coupled to the electric motor. The close-coupled pump has the impeller mounted on a motor shaft extension, and the flexible-coupled pump has an impeller shaft supported by a frame or bracket that is connected to the electric motor through a flexible coupling.

Pumps may also be classified by their mechanical features and installation arrangement. One-horsepower and larger pumps are available as close-coupled or base-mounted. Close-coupled pumps have an end-suction inlet for horizontal mounting or a vertical in-line inlet for direct installation in the piping. Base-mounted pumps are (1) end-suction, frame-mounted or (2) double-suction, horizontal or vertical split-case units. Double-suction pumps can also be arranged in a vertical position on a support frame with the motor vertically mounted on a bracket above the pump. Pumps are usually labeled by their mounting position as either horizontal or vertical.

Circulator Pump

Circulator is a generic term for a pipe-mounted, low-pressure, low-capacity pump (Figure 3). This pump may have a wet rotor or may be driven by a close-coupled or flexible-coupled motor. Circulator pumps are commonly used in residential and small commercial buildings to circulate source water and to recirculate the flow of terminal coils to enhance heat transfer and improve the control of large systems.

Close-Coupled, Single-Stage, End-Suction Pump

The close-coupled pump is mounted on a horizontal motor supported by the motor foot mountings (Figure 4). Mounting usually requires a solid concrete pad. The motor is close-coupled to the pump shaft. This compact pump has a single horizontal inlet and vertical discharge. It may have one or two impellers.

Frame-Mounted, End-Suction Pump on Base Plate

Typically, the motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figure 5). Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft

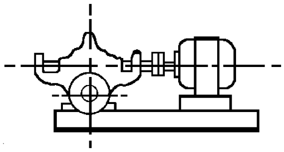


Fig. 6 Base-Mounted, Horizontal (Axial), Split-Case, Single-Stage, Double-Suction Pump

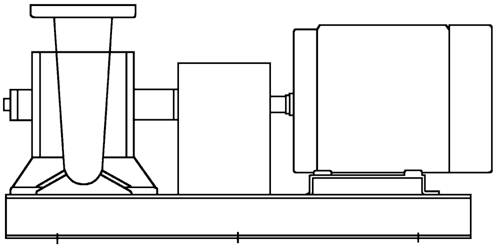


Fig. 7 Base-Mounted, Vertical, Split-Case, Single-Stage, Double-Suction Pump

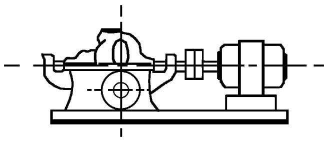


Fig. 8 Base-Mounted, Horizontal, Split-Case, Multistage Pump

and should have an OSHA-approved guard. For horizontal mounting, the piping is horizontal on the suction side and vertical on the discharge side. This pump has a single suction.

Base-Mounted, Horizontal (Axial) or Vertical, Split-Case, Single-Stage, Double-Suction Pump

The motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figures 6 and 7). Sometimes axial pumps are vertically mounted with a vertical pump casing and motor mounting bracket. Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft, and the coupling should have an OSHA-approved guard. A split case permits complete access to the impeller for maintenance. This pump may have one or two double suction impellers.

Base-Mounted, Horizontal, Split-Case, Multistage Pump

The motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figure 8). Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft, and the coupling should have an OSHA-approved guard. Piping is horizontal on both the suction and discharge sides. The split case permits complete access to the impellers for maintenance. This pump has a single suction and may have one or more impellers for multistage operation.

Vertical In-Line Pump

This close-coupled pump and motor are mounted on the pump casing (Figure 9). The unit is compact and depends on the connected piping for support. Mounting requires adequately spaced pipe hangers and, sometimes, a vertical casing support. The suction and discharge piping is horizontal. The pump has a single or double suction impeller.

Vertical Turbine, Single- or Multistage, Sump-Mounted Pump

Vertical turbine pumps have a motor mounted vertically on the pump discharge head for either wet-sump mounting or can-type

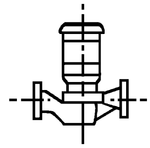


Fig. 9 Vertical In-Line Pump

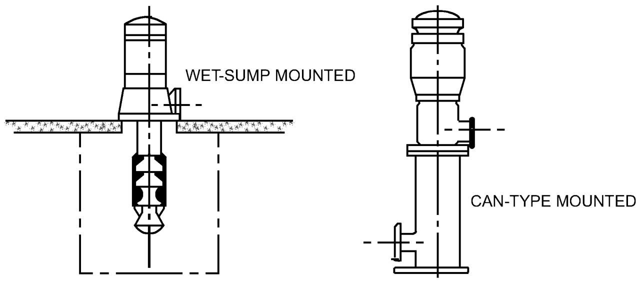


Fig. 10 Vertical Turbine Pumps

mounting (Figure 10). This single-suction pump may have one or multiple impellers for multistage operation. Mounting requires a solid concrete pad or steel sole plate above the wet pit with accessibility to the screens or trash rack on the suction side for maintenance. Can-type mounting requires a suction strainer. Piping is horizontal on the discharge side and on the suction side. The sump should be designed according to Hydraulic Institute (1994) recommendations.

PUMP PERFORMANCE CURVES

Performance of a centrifugal pump is commonly shown by a manufacturer's performance curve (Figure 11). The figure displays the pump power required for a liquid with a specific gravity of 1.0 (water) over a particular range of impeller diameters and flows. The curves are generated from a set of standard tests developed by the Hydraulic Institute (1994). The tests are performed by the manufacturer for a given pump volute or casing and several impeller diameters, normally from the maximum to the minimum allowable in that volute. The tests are conducted at a constant impeller speed for various flows.

Pump curves represent the average results from testing several pumps of identical design under the same conditions. The curve is sometimes called the head-capacity curve (H-Q) for the pump. Typically, the discharge head of the centrifugal pump, sometimes called the total dynamic head (TDH), is measured in feet of water flowing at a standard temperature and pressure. TDH represents the difference in total head between the suction side and the discharge side of the pump. This discharge head decreases as the flow increases (Figure 12). Motors are often selected to be non-overloading at a specified impeller size and maximum flow to ensure safe motor operation at all flow requirements.

The pump characteristic curve may be further described as flat or steep (Figure 13). Sometimes these curves are described as a normal rising curve (flat), a drooping curve (steep), or a steeply rising curve. The pump curve is considered flat if the pressure at shutoff is about 1.10 to 1.20 times the pressure at the best efficiency point. Flat characteristic pumps are usually installed in closed systems with modulating two-way control valves. Steep characteristic pumps are usually installed in open systems, such as cooling towers (see Chapter 13), where higher head and constant flow are usually desired.

Pump manufacturers may compile performance curves for a particular set of pump volutes in a series (Figure 14). The individual curves are shown in the form of an envelope consisting of the maximum and minimum impeller diameters and the ends of their curves. This set of curves is known as a family of curves. A family of curves

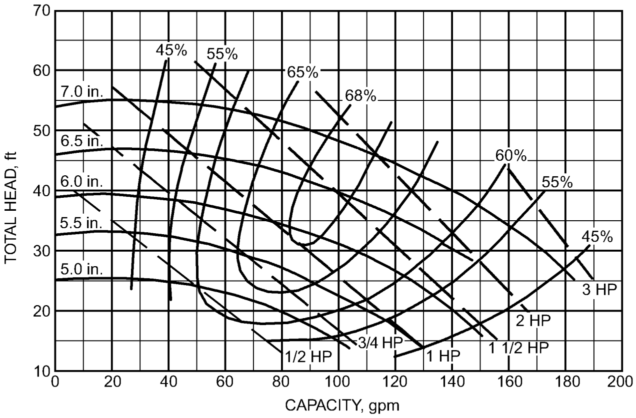


Fig. 11 Typical Pump Performance Curve

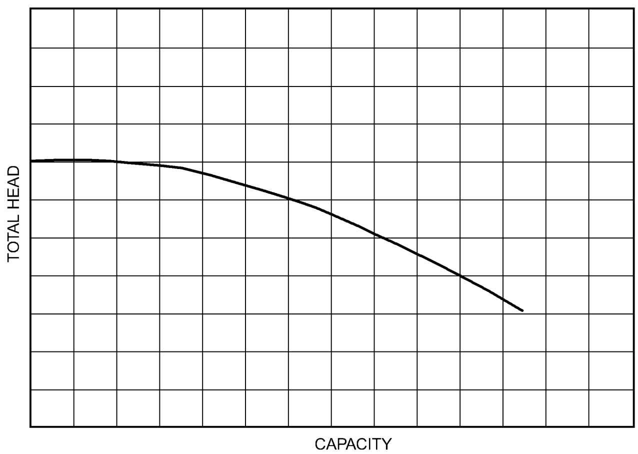


Fig. 12 Typical Pump Curve

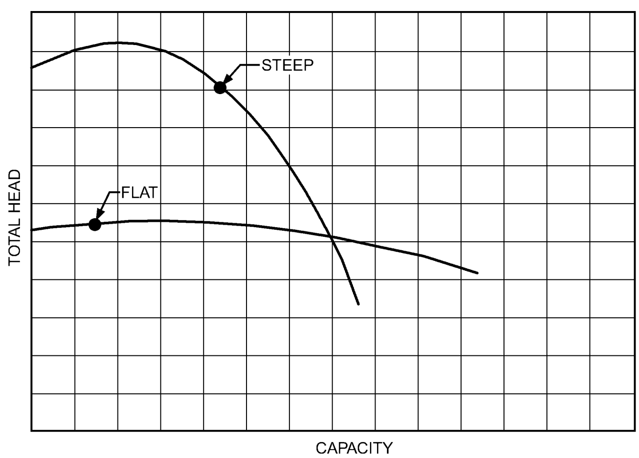


Fig. 13 Flat Versus Steep Performance Curves

is useful in determining the approximate size and model required, but the particular pump curve (Figure 11) must then be used to confirm an accurate selection.

Many pump manufacturers and HVAC software suppliers offer electronic versions for pump selection. Pump selection software typically allows the investigation of different types of pumps and operating parameters. Corrections for fluid specific gravity, temperature, and motor speeds are easily performed.

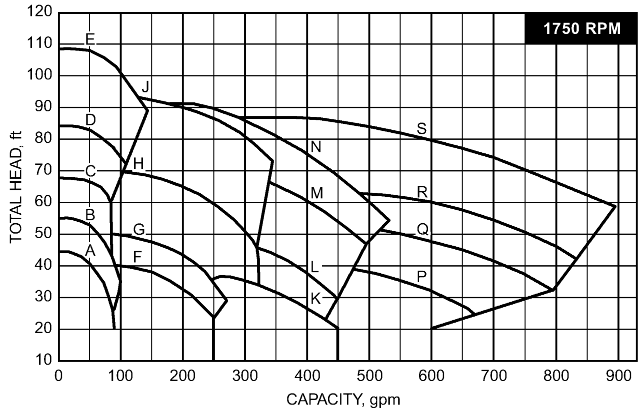


Fig. 14 Typical Pump Manufacturer's Performance Curve Series

HYDRONIC SYSTEM CURVES

Pressure drop caused by the friction of a fluid flowing in a pipe may be described by the Darcy-Weisbach equation:

$$\Delta p = f\frac{L}{D}\frac{\rho}{g}\frac{V^2}{2} \qquad (1)$$

Equation (1) shows that pressure drop in a hydronic system (pipe, fittings, and equipment) is proportional to the square of the flow (V^2 or Q^2 where Q is the flow). Experiments show that pressure drop is more nearly proportional to between $V^{1.85}$ and $V^{1.9}$, or a nearly parabolic curve as shown in Figures 15 and 18. The design of the system (including the number of terminals and flows, the fittings and valves, and the length of pipe mains and branches) affects the shape of this curve.

Equation (1) may also be expressed in head or specific energy form:

$$\Delta h = \frac{\Delta p}{\rho} = f\frac{L}{D}\frac{V^2}{2g} \qquad (2)$$

where

Δh = head loss through friction, ft (of fluid flowing)
Δp = pressure drop, lb/ft^2
ρ = fluid density, lb/ft^3
f = friction factor, dimensionless
L = pipe length, ft
D = inside diameter of pipe, ft
V = fluid average velocity, ft/s
g = gravitational acceleration, 32.2 ft/s^2

The system curve (Figure 15) defines the system head required to produce a given flow rate for a liquid and its characteristics in a piping system design. To produce a given flow, the system head must overcome pipe friction, inside pipe surface roughness, actual fitting losses, actual valve losses, resistance to flow due to fluid viscosity, and possible system effect losses. The general shape of this curve is parabolic since, according to the Darcy-Weisbach equation [Equation (2)], the head loss is proportional to the square of the flow.

If static pressure is present due to the height of the liquid in the system or the pressure in a compression tank, this head is sometimes referred to as **independent head** and is added to the system curve (Figure 16).

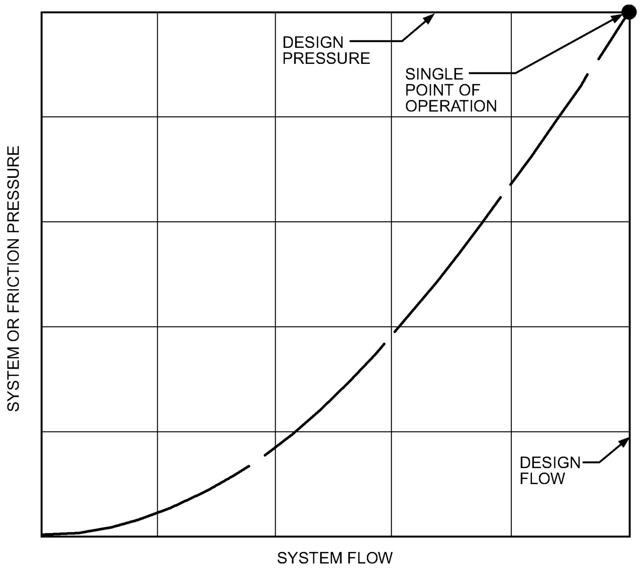


Fig. 15 Typical System Curve

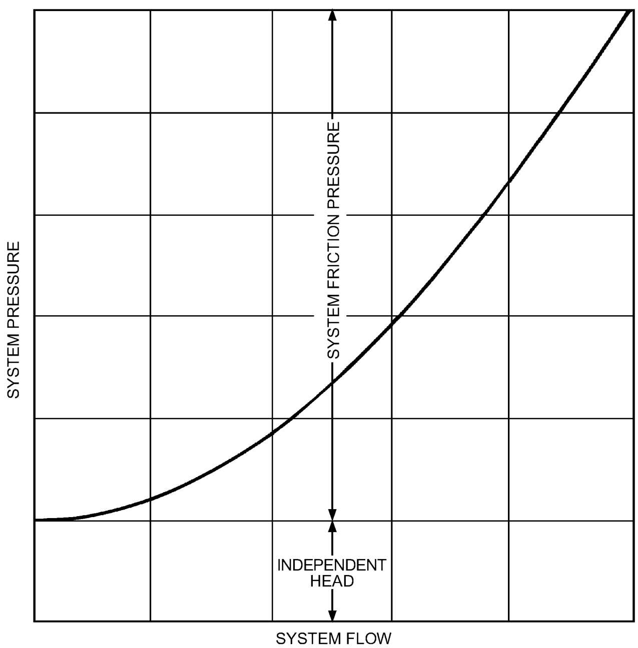


Fig. 16 Typical System Curve with Independent Head

PUMP AND HYDRONIC SYSTEM CURVES

The pump curve and the system curve can be plotted on the same graph. The intersection of the two curves (Figure 17) is the system **operating point**, where the pump's developed head matches the system's head loss.

In a typical hydronic system, a thermostat or controller varies the flow in a load terminal by positioning a two-way control valve to match the load. At full load the two-way valves are wide open, and the system follows curve A in Figure 18. As the load drops, the terminal valves begin closing to match the load (part load). This increases the friction and reduces the flow in the terminals. The system curve gradually changes to curve B.

The operating point of a pump should be considered when the system includes two-way control valves. Point 1 in Figure 19 shows

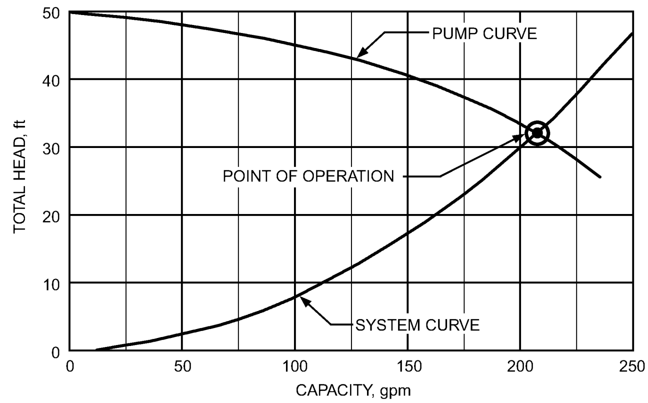


Fig. 17 System and Pump Curves

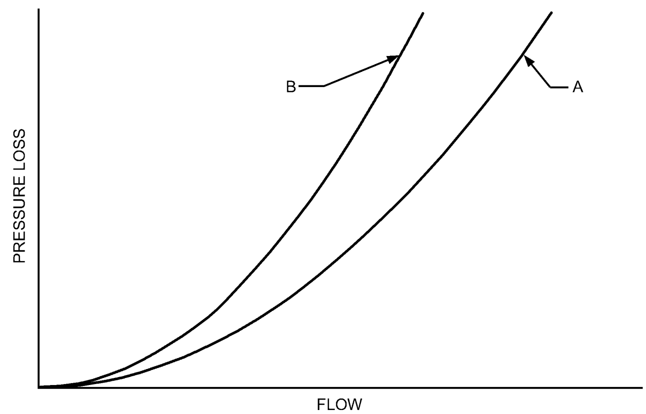


Fig. 18 System Curve Change due to Part-Load Flow

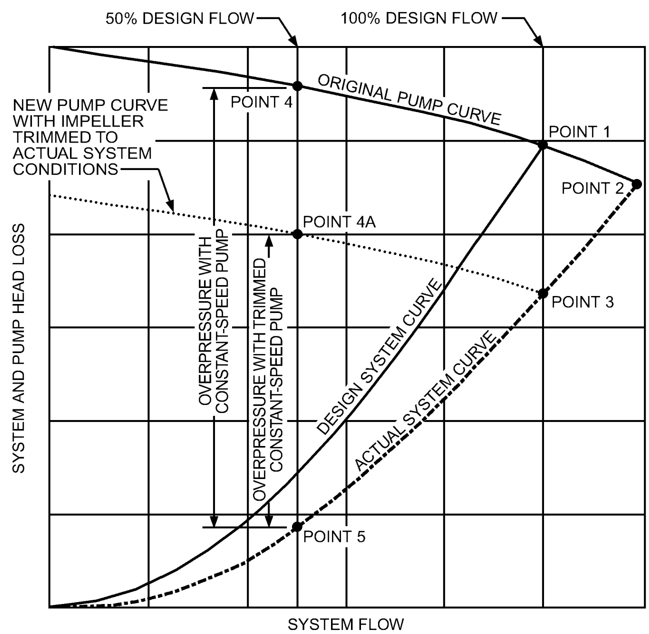


Fig. 19 Pump Operating Points

the pump operating at the design flow at the calculated design head loss of the system. But typically, the actual system curve is slightly different than the design curve. As a result, the pump operates at point 2 and produces a flow rate higher than design.

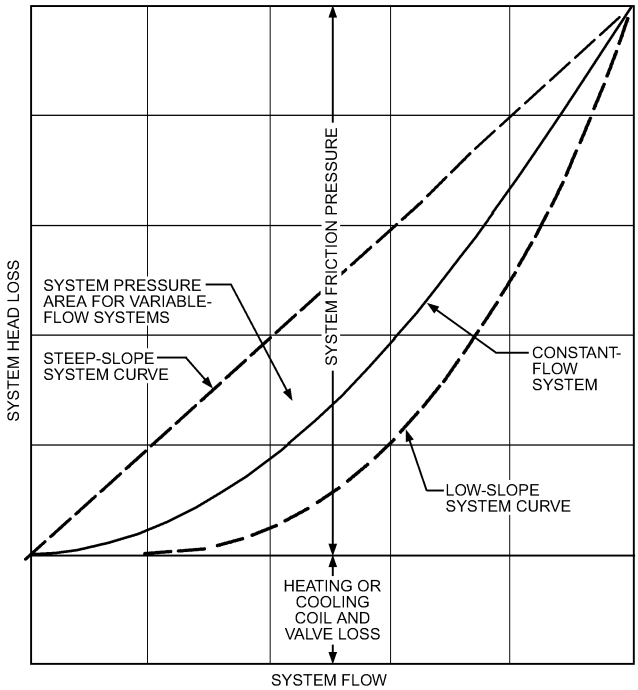


Fig. 20 System Curve, Constant and Variable Head Loss

To reduce the actual flow to the design flow at point 1, a balancing valve downstream from the pump can be adjusted while all the terminal valves are in a wide-open position. This pump discharge balancing valve imposes a pressure drop equal to the pressure difference between point 1 and point 3. The manufacturer's pump curve shows that the capacity may be reduced by substituting a new impeller with a smaller diameter or by trimming the existing pump impeller. After trimming, reopening the balancing valve in the pump discharge then eliminates the artificial drop and the pump operates at point 3. Points 3 and 4A demonstrate the effect a trimmed impeller has on reducing flow.

Figure 20 is an example of a system curve with both fixed head loss and variable head loss. Such a system might be an open piping circuit between a refrigerating plant condenser and its cooling tower. The elevation difference between the water level in the tower pan and the spray distribution pipe creates the fixed head loss. The fixed loss occurs at all flow rates and is, therefore, an independent head as shown.

Most variable-flow hydronic systems have individual two-way control valves on each terminal unit to permit full diversity (random loading from zero to full load). Regardless of the load required in most variable-flow systems, the designer establishes a minimum pressure difference to ensure that any terminal and its control valve receive the design flow at full demand. When graphing a system curve for a nonsymmetrically loaded variable-flow system (Figure 20), the Δh (minimum maintained pressure difference) is treated like a fixed pressure loss (independent head), and becomes the starting datum for the system curve.

The low slope and steep slope curves in Figure 20 represent the boundaries for operation of the system. The net vertical difference between the curves is the difference in friction loss developed by the distribution mains for the two extremes of possible loads. The area in which the system operates depends on the diverse loading or unloading imposed by the terminal units. This area represents the pumping energy that can be conserved with one-speed, two-speed, or variable-speed pumps after a review of the pump power, efficiency, and affinity relationships.

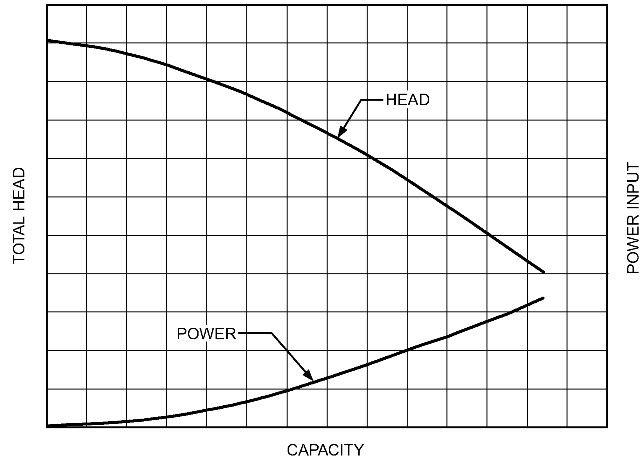


Fig. 21 Typical Pump Water Power Increase with Flow

PUMP POWER

The theoretical power to circulate water in a hydronic system is the **water horsepower** (whp) and is calculated as follows:

$$\text{whp} = \frac{\dot{m}\Delta h}{33{,}000} \tag{3}$$

where

$\dot{m}$ = mass flow of fluid, lb/min
Δh = total head, ft of fluid
33,000 = units conversion, ft·lb/min per hp

At 68°F, water has a density of 62.3 lb/ft^3, and Equation (3) becomes

$$\text{whp} = \frac{Q\Delta h}{3960} \tag{4}$$

where

Q = fluid flow rate, gpm
3960 = units conversion, ft·gpm per hp

Figure 21 shows how water power increases with flow. At other water temperatures or for other fluids, Equation (4) is corrected by multiplying by the specific gravity of the fluid.

The **brake horsepower** (bhp) required to operate the pump is determined by the manufacturer's test of an actual pump running under standard conditions to produce the required flow and head as shown in Figure 11.

PUMP EFFICIENCY

Pump efficiency is determined by comparing the output power to the input power:

$$\text{Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{\text{whp}}{\text{bhp}} \times 100\% \tag{5}$$

Figure 22 shows a typical efficiency versus flow curve.

The pump manufacturer usually plots the efficiencies for a given volute and impeller size on the pump curve to help the designer select the proper pump (Figure 23). The best efficiency point (BEP) is the optimum efficiency for this pump—operation above and below this point is less efficient. The locus of all the BEPs for each impeller size lies on a system curve that passes through the origin (Figure 24).

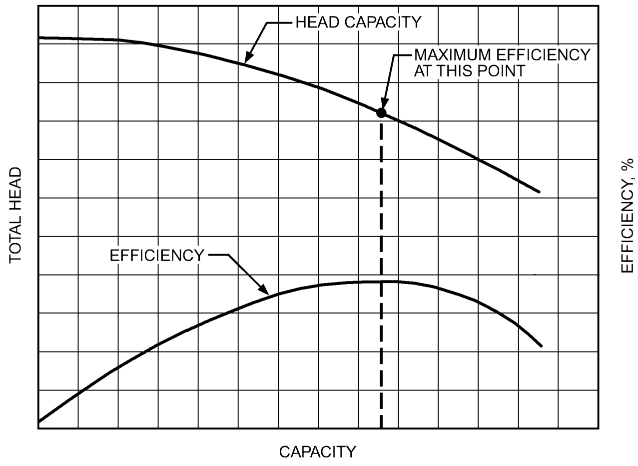


Fig. 22 Pump Efficiency Versus Flow

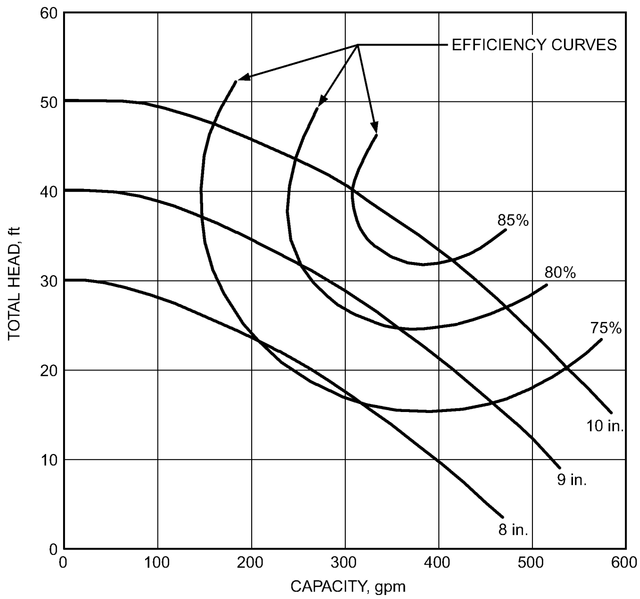


Fig. 23 Pump Efficiency Curves

AFFINITY LAWS

The centrifugal pump, which imparts a velocity to a fluid and converts the velocity energy to pressure energy, can be categorized by a set of relationships called **affinity laws** (Table 1). The laws can be described as similarity processes that follow these rules:

1. Flow (capacity) varies with rotating speed N (i.e., the peripheral velocity of the impeller).
2. Head varies as the square of the rotating speed.
3. Brake horsepower varies as the cube of the rotating speed.

The affinity laws are useful for estimating pump performance at different rotating speeds or impeller diameters D based on a pump with known characteristics. The following two variations can be analyzed by these relationships:

1. By changing speed and maintaining constant impeller diameter, pump efficiency remains the same, but head, capacity, and brake horsepower vary according to the affinity laws.
2. By changing impeller diameter and maintaining constant speed, the pump efficiency for a diffuser pump is not affected if the impeller diameter is changed by less than 5%. However, efficiency changes if the impeller size is reduced enough to affect the clearance between the casing and the periphery of the impeller.

Table 1 Pump Affinity Laws

Function	Speed Change	Impeller Diameter Change
Flow	$Q_2 = Q_1\left(\frac{N_2}{N_1}\right)$	$Q_2 = Q_1\left(\frac{D_2}{D_1}\right)$
Head	$h_2 = h_1\left(\frac{N_2}{N_1}\right)^2$	$h_2 = h_1\left(\frac{D_2}{D_1}\right)^2$
Horsepower	$\mathrm{bhp}_2 = \mathrm{bhp}_1\left(\frac{N_2}{N_1}\right)^3$	$\mathrm{bhp}_2 = \mathrm{bhp}_1\left(\frac{D_2}{D_1}\right)^3$

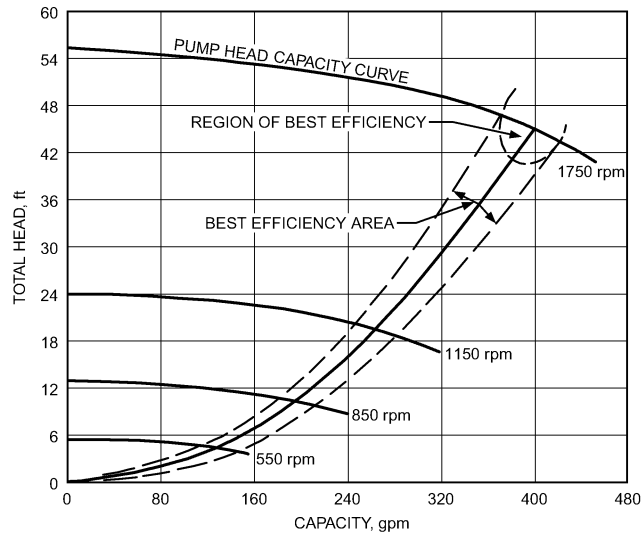


Fig. 24 Pump Best Efficiency Curves

The affinity laws assume that the system curve is known and that head varies as the square of flow. The operating point is the intersection of the total system curve and the pump curve (Figure 17). Because the affinity law is used to calculate a new condition due to a flow or head change (e.g., reduced pump speed or impeller diameter), this new condition also follows the same system curve. Figure 25 shows the relationship of flow, head, and power as expressed by the affinity laws.

The affinity laws can also be used to predict the BEP at other pump speeds. As discussed in the section on Pump Efficiency, the BEP follows a parabolic curve to zero as the pump speed is reduced (Figure 24).

Multiple-speed motors can be used to reduce system overpressure at reduced flow. Standard two-speed motors are available with speeds of 1750/1150 rpm, 1750/850 rpm, 1150/850 rpm, and 3500/1750 rpm. Figure 26 shows the performance of a system with a 1750/1150 multiple-speed pump. In the figure, curve A shows a system's response when the pump runs at 1750 rpm. When the pump runs at 1150 rpm, operation is at point 1 and not at point 2 as the affinity laws predict. If the system were designed to operate as shown in curve B, the pump would operate at shutoff and be damaged if run at 1150 rpm. This example demonstrates that the designer must analyze the system carefully to determine the pump limitation and the effect of lower speed on performance.

Variable-speed drives have a similar effect on pump curves. These drives normally have an infinitely variable speed range, so that the pump, with proper controls, follows the system curve without any overpressure. Figure 27 shows operation of the pump in Figure 19 at 100%, 80%, 64%, 48% and 32% of the speed.

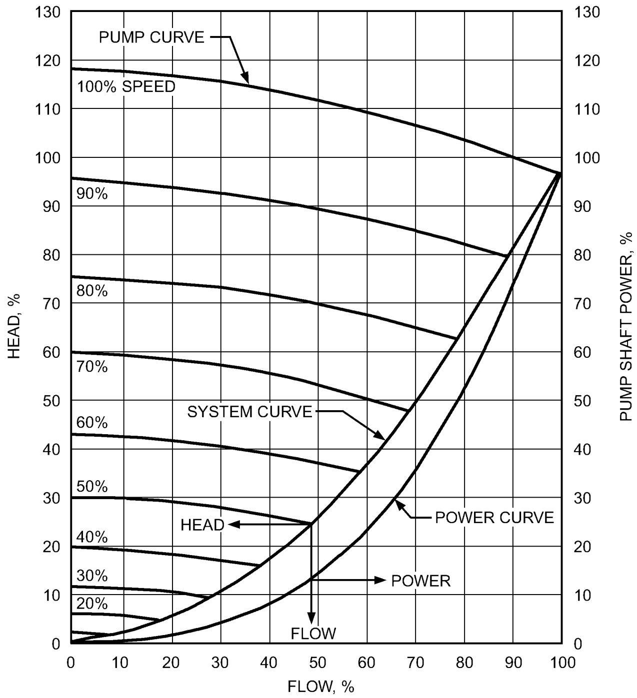


Fig. 25 Pumping Power, Head, and Flow Versus Pump Speed

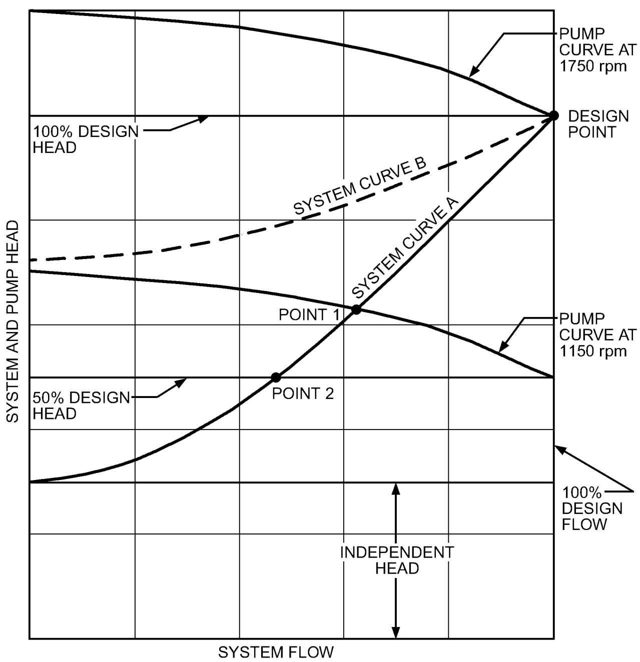


Fig. 26 Example Application of Affinity Law

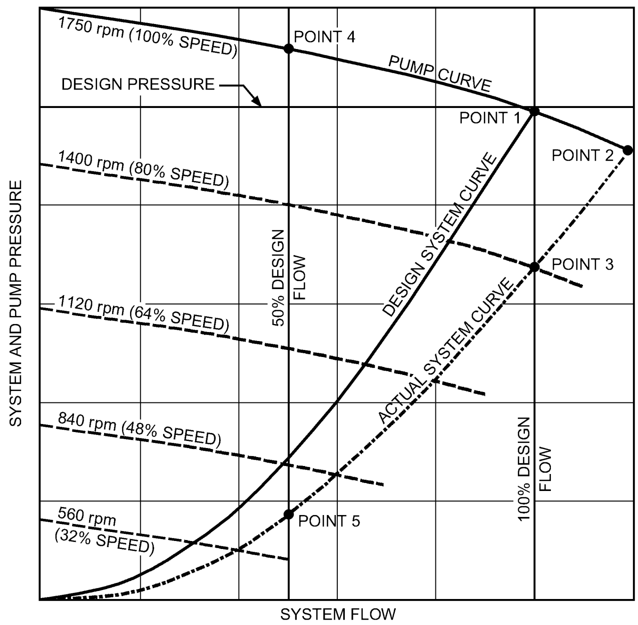


Fig. 27 Variable-Speed Pump Operating Points

Although the variable-speed motor in this example can correct overpressure conditions, about 20% of the operating range of the variable-speed motor is not used. The maximum speed required to provide design flow and pressure is 80% of full speed (1400 rpm) and the practical lower limit is 30% (525 rpm) due to the characteristics of the system curve as well as pump and motor limitations. The variable-speed drive and motor should be sized for actual balanced hydronic conditions.

The affinity laws can also be used to predict the effect of trimming the impeller to reduce overpressure. If, for example, the system shown in Figure 27 were allowed to operate at point 2, an excess flow of 15% to 25% would occur, depending on the shapes of the system and pump curves. The pump affinity laws (Table 1) show that pump capacity varies directly with impeller diameter. In Figure 27, the correct size impeller operates at point 3. A diameter ratio of 0.8 would reduce the overcapacity from 125% to 100%.

The second affinity law shows that head varies as the square of the impeller diameter. For the pump in Figure 27 and an impeller diameter ratio of 0.8, the delivered head of the pump is $(0.8)^2 = 0.64$ or 64% of the design pump head.

The third affinity law states that pump power varies as the cube of the impeller diameter. For the pump in Figure 27 and an impeller diameter ratio of 0.8, the power necessary to provide the design flow is $(0.8)^3 = 0.512$ or 51.2% of the original pump's power.

RADIAL THRUST

In a single-volute centrifugal pump, uniform or near-uniform pressures act on the impeller at design capacity, which coincides with the BEP. However, at other capacities, the pressures around the impeller are not uniform and there is a resultant radial reaction.

Figure 28 shows the typical change in radial thrust with changes in the pumping rate. Specifically, radial thrust decreases from shutoff to the design capacity (if chosen at the BEP) and then increases as flow increases. The reaction at overcapacity is roughly opposite that at partial capacity and is greatest at shutoff. The radial forces at extremely low flow can cause severe impeller shaft deflection and, ultimately, shaft breakage. This danger is even greater with high-pressure pumps.

NET POSITIVE SUCTION CHARACTERISTICS

Particular attention must be given to the pressure and temperature of the water as it enters the pump, especially in condenser towers, steam condensate returns, and steam boiler feeds. If the absolute pressure at the suction nozzle approaches the vapor pressure of the liquid, vapor pockets form in the impeller passages. The collapse of the vapor pockets (**cavitation**) is noisy and can be destructive to the pump impeller.

The amount of pressure in excess of the vapor pressure required to prevent vapor pockets from forming is known as the net

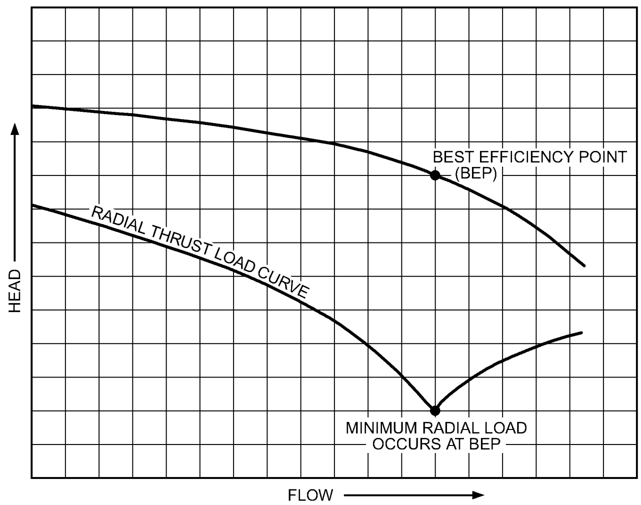


Fig. 28 Radial Thrust Versus Pumping Rate

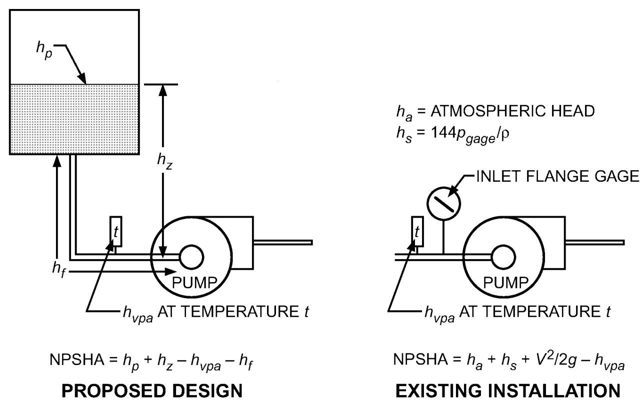


Fig. 29 Net Positive Suction Head Available

positive suction head required (NPSHR). NPSHR is a characteristic of a given pump and varies with pump speed and flow. It is determined by the manufacturer and is included on the pump performance curve.

NPSHR is particularly important when a pump is operating with hot liquids or is applied to a circuit having a suction lift. The vapor pressure increases with water temperature and reduces the net positive suction head available (NPSHA). Each pump has its NPSHR, and the installation has its NPSHA, which is the total useful energy above the vapor pressure at the pump inlet.

The following equation may be used to determine the NPSHA in a proposed design (see Figure 29):

$$NPSHA = h_p + h_z - h_{vpa} - h_f \tag{6}$$

where

h_p = absolute pressure on surface of liquid that enters pump, ft of head
h_z = static elevation of liquid above center line of pump (h_z is negative if liquid level is below pump center line), ft
h_{vpa} = absolute vapor pressure at pumping temperature, ft
h_f = friction and head losses in suction piping, ft

To determine the NPSHA in an existing installation, the following equation may be used (see Figure 29):

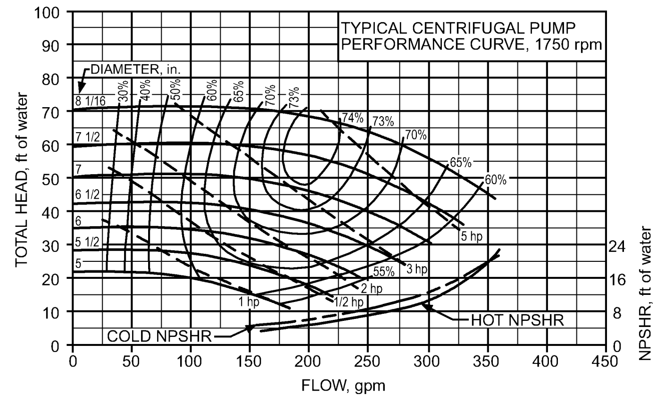


Fig. 30 Pump Performance and NPSHR Curves

$$NPSHA = h_a + h_s + \frac{V^2}{2g} - h_{vpa} \tag{7}$$

where

h_a = atmospheric head for elevation of installation, ft
h_s = head at inlet flange corrected to center line of pump (h_s is negative if below atmospheric pressure), ft
$V^2/2g$ = velocity head at point of measurement of h_s, ft

If the NPSHA is less than the pump's NPSHR, cavitation, noise, inadequate pumping, and mechanical problems will result. **For trouble-free design, the NPSHA must always be greater than the pump's NPSHR.** In closed hot and chilled water systems where sufficient system fill pressure is exerted on the pump suction, NPSHR is normally not a factor. Figure 30 shows pump curves and NPSHR curves. Cooling towers and other open systems require calculations of NPSHA.

SELECTION OF PUMPS

A substantial amount of data is required to ensure that an adequate, efficient, and reliable pump is selected for a particular system. The designer should review the following criteria:

- Design flow
- Pressure drop required for the most resistant loop
- Minimum system flow
- System pressure at maximum and minimum flows
- Type of control valve—two-way or three-way
- Continuous or variable flow
- Pump environment
- Number of pumps and standby
- Electric voltage and current
- Electric service and starting limitations
- Motor quality versus service life
- Water treatment, water conditions, and material selection

When a centrifugal pump is applied to a piping system, the operating point satisfies both the pump and system curves (Figure 17). As the load changes, control valves change the system curve and the operating point moves to a new point on the pump curve. Figure 31 shows the optimum regions to use when selecting a centrifugal pump. The areas bounded by lines AB and AC represent operating points that lie in the preferred pump selection range. But, because pumps are only manufactured in certain sizes, selection limits of 66% to 115% of flow at the BEP are suggested. The satisfactory range is that portion of a pump's performance curve where the combined effect of circulatory flow, turbulence, and friction losses are minimized. Where possible, pumps should be chosen to operate to the left of the BEP because the pressure in the actual system may be less than design due to overstated data for pipe friction and for other equipment. Otherwise, the pump operates at a higher flow and possibly in the turbulent region (Stethem 1988).

ARRANGEMENT OF PUMPS

In a large system, a single pump may not be able to satisfy the full design flow and yet provide both economical operation at partial loads and a system backup. The designer may need to consider the following alternative pumping arrangements and control scenarios:

- Multiple pumps in parallel or series
- Standby pump
- Pumps with two-speed motors
- Primary-secondary pumping
- Variable-speed pumping
- Distributed pumping

Parallel Pumping

When pumps are applied in parallel, each pump operates at the same head and provides its share of the system flow at that head (Figure 32). Generally, pumps of equal size are recommended, and the parallel pump curve is established by doubling the flow of the single pump curve.

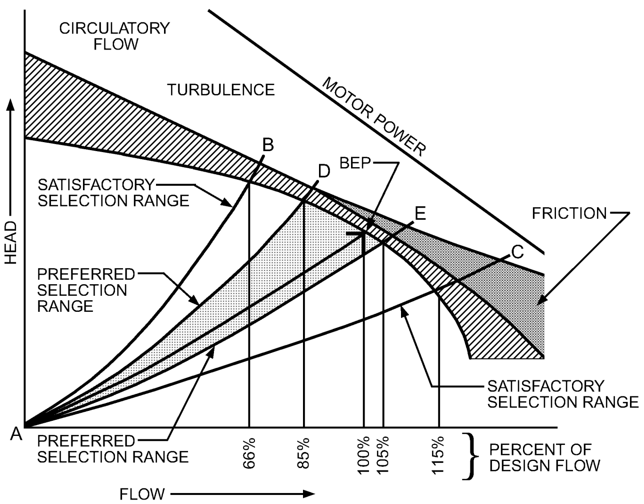


Fig. 31 Pump Selection Regions

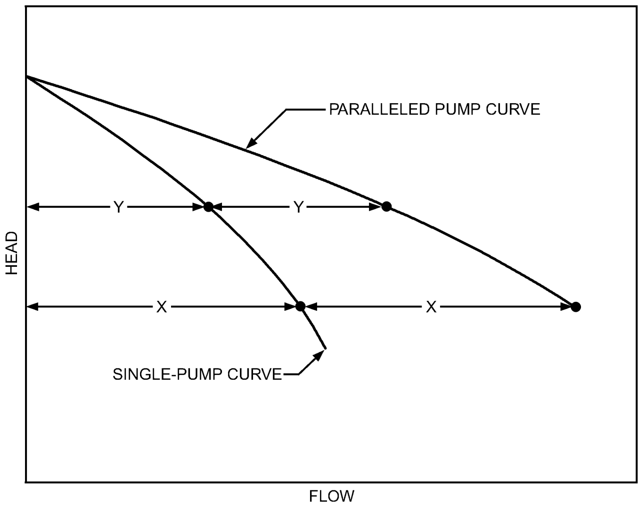


Fig. 32 Pump Curve Construction for Parallel Operation

Plotting a system curve across the parallel pump curve shows the operating points for both single and parallel pump operation (Figure 33). Note that single pump operation does not yield 50% flow. The system curve crosses the single pump curve considerably to the right of its operating point when both pumps are running. This leads to two important concerns: (1) the motor must be selected to prevent overloading during operation of a single pump and (2) a single pump can provide standby service for up to 80% of the design flow, the actual amount depending on the specific pump curve and system curve.

Construction of the composite curve for two dissimilar parallel pumps requires special care; for example, note the shoulder in the composite pump curve in Figure 34.

Operation. The piping of parallel pumps (Figure 35) should permit running either pump. A check valve is required in each pump's discharge to prevent backflow when one pump is shut down. Hand valves and a strainer allow one pump to be serviced while the other is operating. A strainer protects a pump by preventing foreign material from entering the pump. Gages or a common gage with a trumpet valve, which includes several valves as one unit, or pressure taps permits checking pump operation.

Flow can be determined (1) by measuring the pressure increase across the pump and using a factory pump curve to convert the pressure to flow, or (2) by use of a flow-measuring station or multipurpose valve. Parallel pumps are often used for hydronic heating and cooling. In this application, both pumps operate during the cooling season to provide maximum flow and pressure, but only one pump operates during the heating season.

Series Pumping

When pumps are applied in series, each pump operates at the same flow rate and provides its share of the total pressure at that

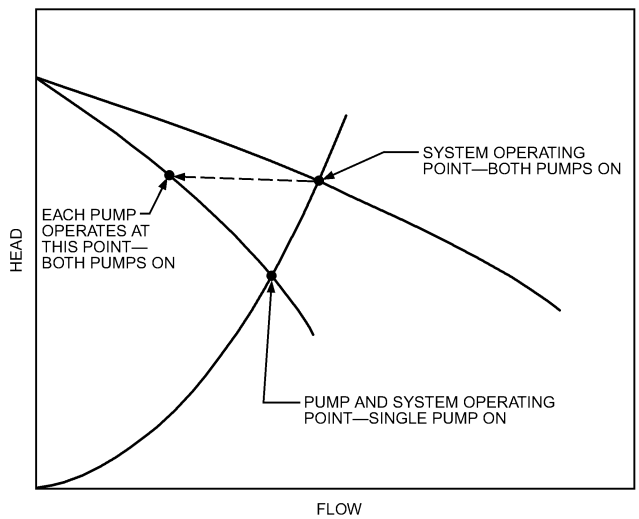


Fig. 33 Operating Conditions for Parallel Operation

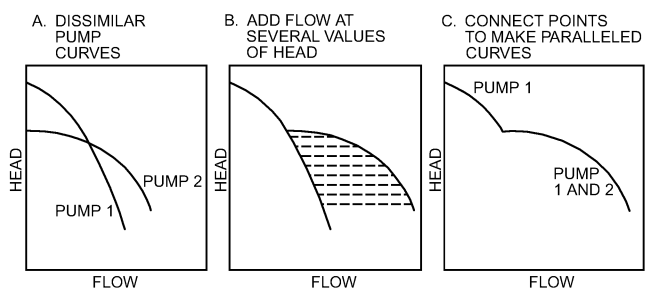


Fig. 34 Construction of Curve for Dissimilar Parallel Pumps

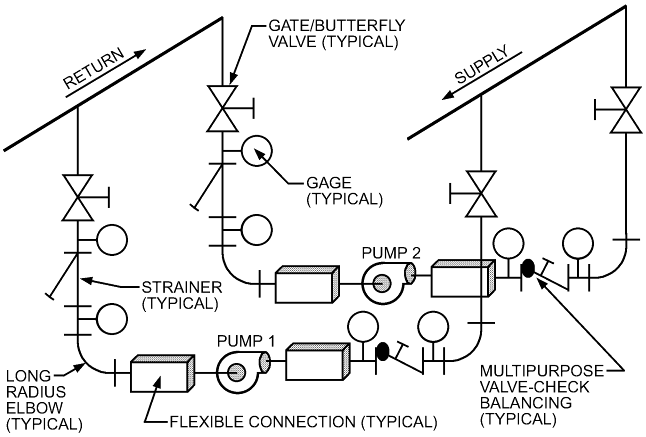


Fig. 35 Typical Piping for Parallel Pumps

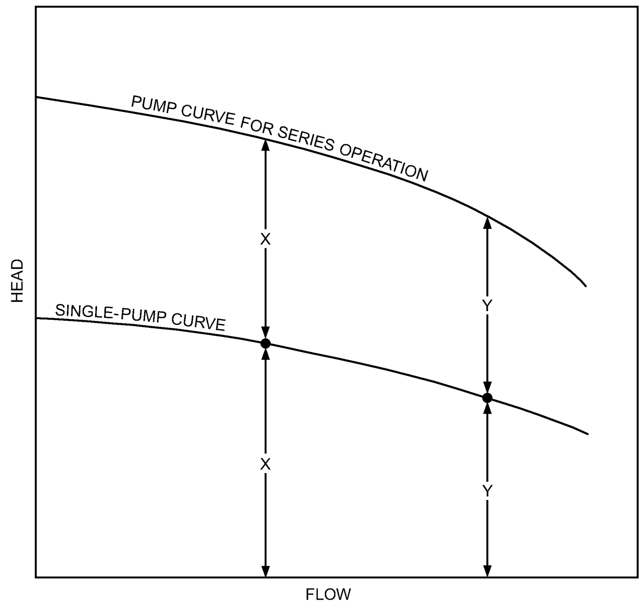


Fig. 36 Pump Curve Construction for Series Operation

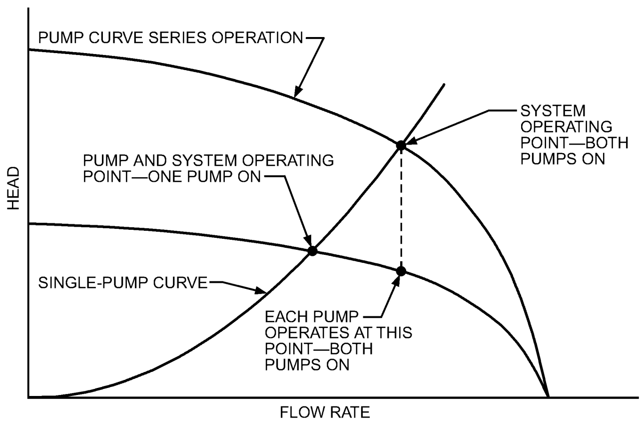


Fig. 37 Operating Conditions for Series Operation

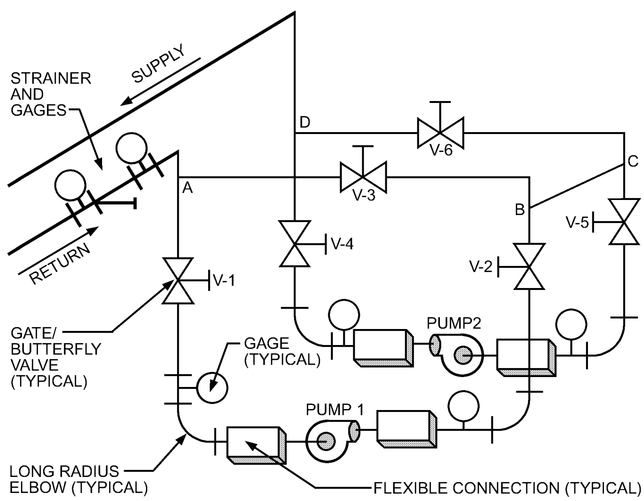


Fig. 38 Typical Piping for Series Pumps

flow (Figure 36). A system curve plot shows the operating points for both single and series pump operation (Figure 37). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

As with parallel pumps, piping for series pumps should permit running either pump (Figure 38). A bypass with a hand valve permits servicing one pump while the other is in operation. Operation and flow can be checked the same way as for parallel pumps. A strainer prevents foreign material from entering the pumps.

Note that both parallel and series pump applications require that the pump operating points be used to accurately determine the actual pumping points. The manufacturer's pump test curve should be consulted. Adding too great a safety factor for pressure, using improper pressure drop charts, or incorrectly calculating pressure drops may lead to a poor selection. In designing systems with multiple pumps, operation in either parallel or series must be fully understood and considered by both designer and operator.

Standby Pump

A backup or standby pump of equal capacity and pressure installed in parallel to the main pump is recommended to operate during an emergency or to ensure continuous operation when a pump is taken out of operation for routine service. A standby pump installed in parallel with the main pump is shown in Figure 35.

Pumps with Two-Speed Motors

A pump with a two-speed motor provides a simple means of reducing capacity. As discussed in the section on Affinity Laws, pump capacity varies directly with impeller speed. At 1150 rpm, the capacity of a pump with a 1750/1150 rpm motor is 1150/1750 = 0.657 or 66% of the capacity at 1750 rpm.

Figure 39 shows an example (Stethem 1988) with two parallel two-speed pumps providing flows of 2130 gpm at 75 ft of head, 1670 gpm at 50.5 ft, 1250 gpm at 33 ft, and 985 gpm at 26.2 ft of head. Points A, B, C, and D will move left along the pump curve as loading changes.

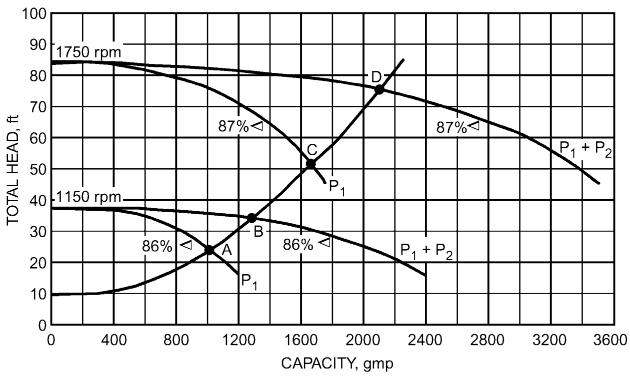
Primary-Secondary Pumping

In a primary-secondary or compound pumping arrangement, a secondary pump is selected to provide the design flow in the load coil from the common pipe between the supply and return distribution mains (Figure 40) (Coad 1985). The pressure drop in the common pipe should not exceed 1.5 ft (Carlson 1972).

In circuit A of the figure, a two-way valve permits a variable flow in the supply mains by reducing the source flow; a secondary pump provides a constant flow in the load coil. The source pump at the chiller or boiler is selected to circulate the source and mains, and the secondary pump is sized for the load coil. Three-way valves in circuits B and C provide a constant source flow regardless of the load.

Variable-Speed Pumping

In a variable-speed pumping arrangement, constant flow pump(s) recirculate the chiller or boiler source in a primary source loop, and a variable-speed distribution pump located at the source plant draws flow from the source loop and distributes to the load terminals as shown in Figure 41. The speed of the distribution pump is determined by a controller measuring differential pressure across the supply-return mains or across selected critical zones. Two-way control valves are installed in the load terminal return branch to vary the flow required in the load.



VARIABLE-VOLUME SYSTEM	POINT	FLOW, gpm	HEAD, ft	EFF., %	BHP
Two Equal-Sized Pumps $P_1 = P_2$	A	985	26.25	85	7.5
C/W Two-Speed Motors 1150/1750 rpm	B	1250	33	82	13
	C	1670	50.5	79	27.5
◁ = BEPs	D	2130	75	84	48

Fig. 39 Example of Two Parallel Pumps with Two-Speed Motors

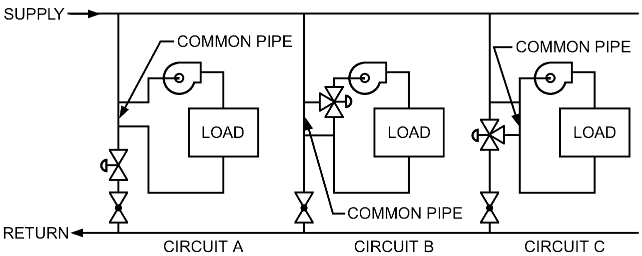


Fig. 40 Primary-Secondary Pumping

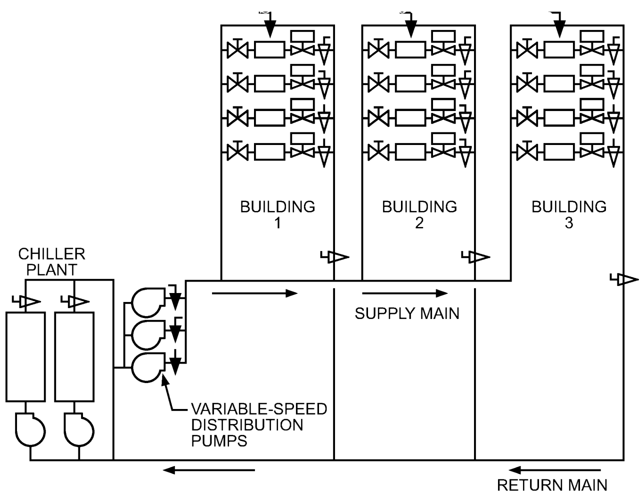


Fig. 41 Variable-Speed Source-Distributed Pumping

Distributed Pumping

In a variable-speed distributed pumping arrangement, constant flow pump(s) recirculate the chiller or boiler source in a primary source loop, and a variable-speed zone or building pump draws flow from the source loop and distributes to the zone load terminals as shown in Figure 42. The speed of the zone or building distribution pump is determined by a controller measuring zone differential pressure across supply-return mains or across selected critical zones. Two-way control valves in the load terminal return branch vary the flow required in the load.

MOTIVE POWER

Figure 43 demonstrates the improvement in efficiency of four-pole (1800 rpm) 25 to 125 hp motors from old, standard efficiency models to current standards and available premium efficiency models.

Electric motors drive most centrifugal pumps for hydronic systems. Internal combustion engines or steam turbines power some pumps, especially in central power plants for large installations. Electric motors for centrifugal pumps can be any of the horizontal or vertical electric motors described in Chapter 44. The sizing of electric

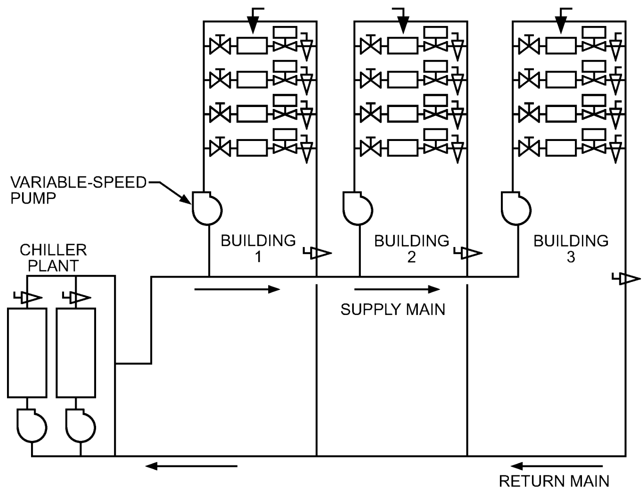


Fig. 42 Variable-Speed Distributed Pumping

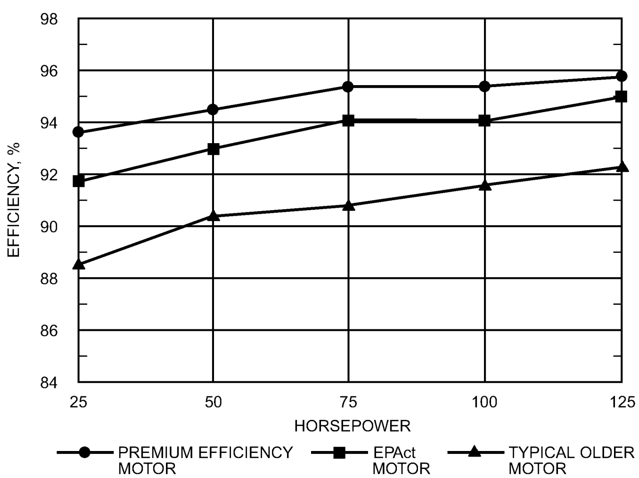


Fig. 43 Efficiency Comparison of Four-Pole Motors

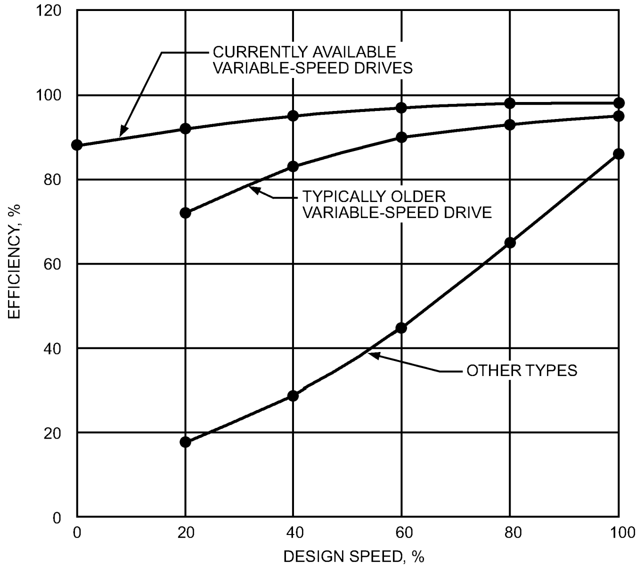


Fig. 44 Typical Efficiency Range of Variable-Speed Drives

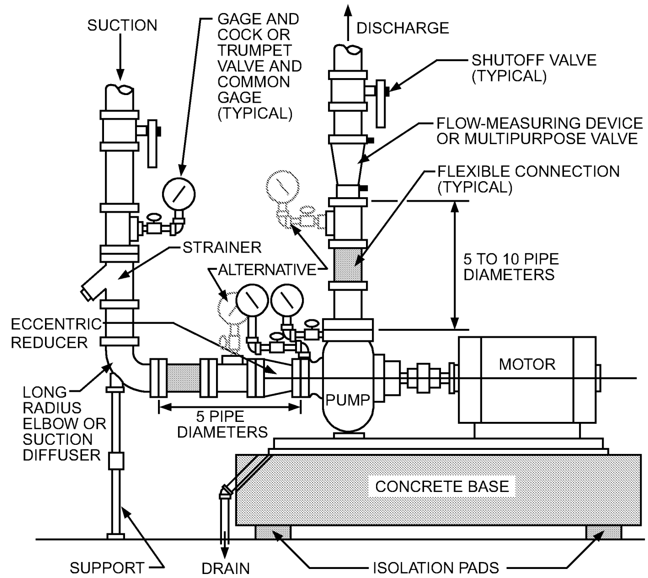


Fig. 45 Base Plate-Mounted Centrifugal Pump Installation

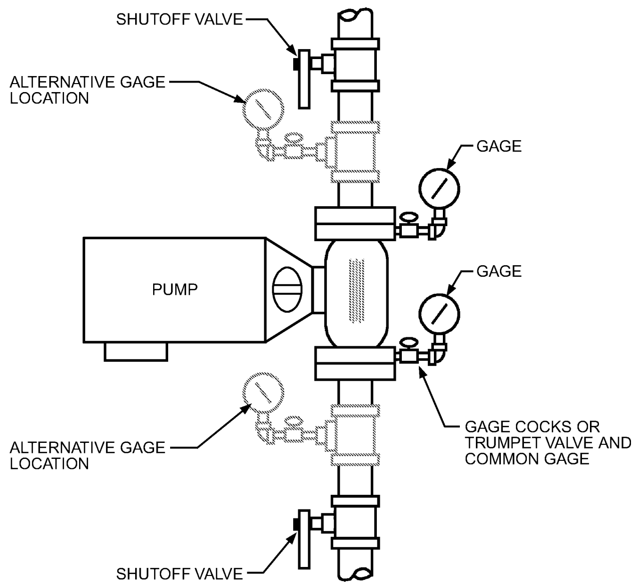


Fig. 46 In-Line Pump Installation

motors is critical because of the cost of electric power and the desire for improved efficiency. Non-overloading motors should be used; that is, the motor nameplate rating must exceed the pump brake horsepower (kW) at any point on the pump curve.

Many pumps for hydronic systems are close-coupled, with the pump impeller mounted on the motor shaft extension. Other pumps are flexible-coupled to the electric motor through a pump mounting bracket or frame. A pump on a hydronic system with variable flow has a broad range of power requirements, which results in reduced motor loading at low flow.

Many variable-speed drives (VSDs) are available for operating centrifugal pumps. Primarily, these include variable-frequency drives, and, occasionally, direct-current, wound rotor, and eddy current drives. Each drive has specific design features that should be evaluated for use with hydronic pumping systems. The efficiency range from minimum to maximum speed (as shown in Figure 44) should be investigated.

ENERGY CONSERVATION IN PUMPING

Pumps for heating and air conditioning consume appreciable amounts of energy. Economical use of energy depends on the efficiency of pumping equipment and drivers, as well as the use of the pumping energy required. Equipment efficiency (sometimes called the wire to water efficiency) shows how much energy applied to the pumping system results in useful energy distributing the water. For an electric-driven, constant speed pump, the equipment efficiency is

$$\eta_e = \eta_p \eta_m \tag{8}$$

where

η_e = equipment efficiency, 0 to 1
η_p = pump efficiency, 0 to 1
η_m = motor efficiency, 0 to 1

For a variable-speed pump, the variable-speed drive efficiency η_v (0 to 1) must be included in the equipment efficiency equation:

$$\eta_e = \eta_p \eta_m \eta_v \tag{9}$$

INSTALLATION, OPERATION, AND COMMISSIONING

1. Pumps may be base plate-mounted (Figure 45), either singly or in packaged sets, or installed in-line directly in the piping system (Figure 46). Packaged sets include multiple pumps, accessories, and electrical controls shipped to the job site on one frame. Packaged pump sets may reduce the requirements for multiple piping and field electrical connections and can be factory tested to ensure specified performance.
2. A concrete pad provides a secure mounting surface for anchoring the pump base plate and raises the pump off the floor to permit housekeeping. The minimum weight of concrete that should be used is 2.5 times the weight of the pump assembly. The pad should be at least 4 in. thick and 6 in. wider than the pump base plate on each side.
3. In applications where the pump bolts rigidly to the pad base, level the pad base, anchor it, and fill the space between pump base and the concrete with a non-shrink grout. Grout prevents the base from shifting and fills in irregularities. Pumps mounted on vibration isolation bases require special installation (see the section on Vibration Isolation and Control in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*).

Table 2 Pumping System Noise Analysis Guide

Complaint	Possible Cause	Recommended Action
Pump or system noise	Shaft misalignment	• Check and realign
	Worn coupling	• Replace and realign
	Worn pump/motor bearings	• Replace, check manufacturer's lubrication recommendations • Check and realign shafts
	Improper foundation or installations	• Check foundation bolting or proper grouting • Check possible shifting caused by piping expansion/contraction. • Realign shafts
	Pipe vibration and/or strain caused by pipe expansion/contraction	• Inspect, alter, or add hangers and expansion provision to eliminate strain on pump(s)
	Water velocity	• Check actual pump performance against specified, and reduce impeller diameter as required • Check for excessive throttling by balance valves or control valves
	Pump operating close to or beyond end point of performance curve	• Check actual pump performance against specified, and reduce impeller diameter as required
	Entrained air or low suction pressure	• Check expansion tank connection to system relative to pump suction • If pumping from cooling tower sump or reservoir, check line size • Check actual ability of pump against installation requirements • Check for vortex entraining air into suction line

Table 3 Pumping System Flow Analysis Guide

Complaint	Possible Cause	Recommended Action
Inadequate or no circulation	Pump running backward (3-phase)	• Reverse any two motor leads
	Broken pump coupling	• Replace and realign
	Improper motor speed	• Check motor nameplate wiring and voltage
	Pump (or impeller diameter) too small	• Check pump selection (impeller diameter) against specified requirements
	Clogged strainer(s)	• Inspect and clean screen
	Clogged impeller	• Inspect and clean
	System not completely filled	• Check setting of PRV fill valve • Vent terminal units and piping high points
	Balance valves or isolating valves improperly set	• Check setting and adjust as required
	Air-bound system	• Vent piping and terminal units • Check location of expansion tank connection line relative to pump suction • Review provisions to eliminate air
	Air entrainment	• Check pump suction inlet conditions to determine if air is being entrained from suction tanks or sumps
	Insufficient NPSHR	• Check NPSHR of pump • Inspect strainers and check pipe sizing and water temperature

4. Support in-line pumps independently from the piping so that pump flanges are not overstressed.
5. Once the pump has been mounted to the base, check the alignment of the motor to the pump. Align the pump shaft couplings properly and shim the motor base as required. Incorrect alignment may cause rapid coupling and bearing failure.
6. Pump suction piping should be direct and as smooth as possible. Install a strainer (coarse mesh) in the suction to remove foreign particles that can damage the pump. Use a straight section of piping at least 5 to 10 diameters long at the pump inlet and long radius elbows to ensure uniform flow distribution. Suction diffusers may be installed in lieu of the straight pipe requirement where spacing is a constraint. Eccentric reducers at the pump flange reduce the potential of air pockets forming in the suction line.
7. If a flow-measuring station (venturi, orifice plate, or balancing valve) is located in the pump discharge, allow 10 diameters of straight pipe between the pump discharge and the flow station for measurement accuracy.
8. Pipe flanges should match the size of pump flanges. Mate flat-face pump flanges with flat-face piping flanges and full-face gaskets. Install tapered reducers and increasers on suction and discharge lines to match the pipe size and pump flanges.
9. If fine mesh screen is used in the strainer at initial start-up to remove residual debris, replace it with normal size screen after commissioning to protect the pump and minimize the suction pressure drop.
10. Install shutoff valves in the suction and discharge piping near the pump to permit removing and servicing the pump and strainer without draining the system. Install a check valve in the pump discharge to prevent reverse flow in a non-running pump when multiple pumps are installed.
11. Install vibration isolators in the pump suction and discharge lines to reduce the transmission of vibration noise to building spaces (Figure 45). Properly located pipe hangers and supports can reduce the transmission of piping strains to the pump.
12. Various accessories need to be studied as alternates to conventional fittings. A suction diffuser in the pump inlet is an alternate to an eccentric reducer and it contains a strainer. Separate strainers can be specified with screen size. A multipurpose valve in the pump discharge is an alternate way to combine the functions of shutoff, check, and balancing valves.
13. Each pump installation should include pressure gages and a gage cock to verify system pressures and pressure drop. As a minimum, pressure taps with an isolation valve and common gage should be available at the suction and discharge of the pump. An additional pressure tap upstream of the strainer permits checking for pressure drop.

TROUBLESHOOTING

Table 2 lists possible causes and recommended solutions for pump or system noise. Table 3 lists possible causes and recommended solutions for inadequate circulation.

REFERENCES

Carlson, G.F. 1972. Central plant chilled water systems—Pumping & flow balance, Part I. *ASHRAE Journal* (February):27-34.

Coad, W.J. 1985. Variable flow in hydronic systems for improved stability, simplicity, and energy economics. *ASHRAE Transactions* 91(1B):224-237.

Hydraulic Institute. 1994. Centrifugal pumps. *Standards* HI 1.1 to 1.5. Parsippany, NJ.

Stethem, W.C. 1988. Application of constant speed pumps to variable volume systems. *ASHRAE Transactions* 94(2):1458-66.

BIBLIOGRAPHY

ASHRAE. 1985. Hydronic systems: Variable-speed pumping and chiller optimization. *ASHRAE Technical Data Bulletin* 1(7).
ASHRAE. 1991. Variable-flow pumping systems. *ASHRAE Technical Data Bulletin* 7(2).
ASHRAE. 1998. *Fundamentals of water system design*. Self-Directed Learning Course.
Beaty, F.E., Jr. 1987. *Sourcebook of HVAC details*. McGraw-Hill, New York.
Carrier. 1965. *Carrier handbook of air conditioning system design.* McGraw-Hill, New York.
Clifford, G. 1990. *Modern heating, ventilating and air conditioning*. Prentice Hall, New York.
Crane Co. 1988. Flow of fluids through valves, fittings, and pipe. *Technical Paper* 410. Joliet, IL.
Dufour, J.W. and W.E. Nelson. 1993. *Centrifugal pump sourcebook*. McGraw-Hill, New York.
Garay, P.N. 1990. *Pump application desk book.* Fairmont Press, Lilburn, GA.
Haines, R.W. and C.L. Wilson. 1994. *HVAC systems design handbook,* 2nd ed. McGraw-Hill, New York.
Hegberg, R.A. 1991. Converting constant-speed hydronic pumping systems to variable-speed pumping. *ASHRAE Transactions* 97(1):739-745.
Karassik, I.J. 1989. Centrifugal pump clinic. Marcel Dekker Inc., New York.
Karassik, I.J., W.C. Krutzsch, W.H. Fraser, and J.P. Messina, eds. 1986. *Pump handbook*. McGraw-Hill, New York.
Levenhagen, J. and D. Spethmann. 1993. *HVAC controls and systems*. McGraw-Hill, New York.
Lobanoff, V.S. and R.R. Ross. 1986. *Centrifugal pumps design and application*. Gulf Publishing, Houston.
Matley, J., ed. 1989. *Progress in pumps*. Chemical Engineering, McGraw-Hill, New York.
McQuiston, F.C. and J.D. Parker. 1988. *Heating, ventilating and air conditioning*. John Wiley & Sons, New York.
Monger, S. 1990. T*esting and balancing HVAC air and water systems.* Fairmont Press, Lilburn, GA.
Rishel, J.B. 1994. Distributed pumping for pumping for chilled- and hot-water systems. *ASHRAE Transactions* 100(1):1521-1527.
Trane Co. 1965. *Trane manual of air conditioning.* La Crosse, WI.