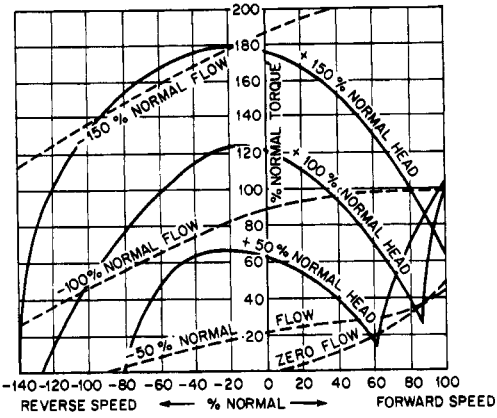


FIGURE 22 Typical reverse-speed-torque characteristics of a low-specific-speed, radial-flow, double-suction pump

little or no torque resistance, the pump will reach higher than normal forward speed in the reverse direction. This runaway speed will increase with specific speed and system head. Shown in Figures 22 and 23 are speed, torque, head, and flow, all expressed as a percentage of the pump design conditions for the normal forward speed. When a pump is running in reverse as a turbine under no load, the head on the pump will be static head (or head from other pumps) minus head loss as a result of friction due to the reverse flow.

If an attempt is made to start the pump while it is running in reverse, an electric motor must apply positive torque to the pump while the motor is initially running in a negative direction. Figures 22 and 23 show, for the two pumps, the torques required to decelerate, momentarily stop, and then accelerate the pump to normal speed. If either of these pumps were pumping into an all-static-head system, starting it in reverse would require overcoming 100% normal head, and it can be seen that a torque in excess of normal would be required by the driver while the driver is running in reverse. In addition to overcoming positive head, the driver must add additional torque to the pump to change the direction of the liquid. This could result in a prolonged starting time under higher than normal current demand. Characteristics of the motor, pump, and system must be analyzed together

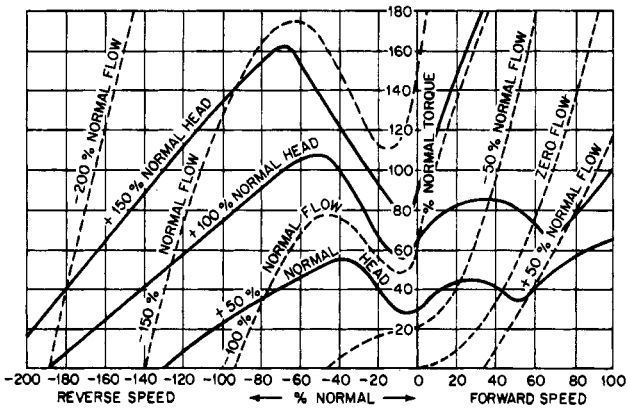


FIGURE 23 Typical reverse-speed-torque characteristics of a high-specific-speed, axial-flow, diffuser pump

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to determine actual operating conditions during this transient period. Starting torques requiring running in reverse become less severe when the system head is partly or all friction head (See Abnormal operation and 4-quadrant characteristics in Sections 2.1.3, 11.3, and 12.14.)

Inertial Head If the system contains an appreciable amount of liquid, the inertia of the liquid mass could offer a significant resistance to any sudden change in velocity. Upon starting a primed pump and system without a valve, all the liquid in the system must accelerate from rest to a final condition of steady flow. Figure 24 illustrates a typical system head resistance that could be produced by a propeller pump pumping through a friction system when accelerated from rest to full speed. If the pump were accelerated very slowly, it would produce an all-friction resistance with zero liquid acceleration varying with flow approaching curve *OABCDEF*. Individual points on this curve represent system resistance at various constant pump speeds. If the pump were accelerated very rapidly, it would produce a system resistance approaching curve *OGHIJKL*. This is a shutoff condition that cannot be realized unless infinite driver torque is available. Individual points on this curve represent a system resistance at various constant speeds with no flow, which is the same as operating with a closed discharge valve. Curve *LF* represents the maximum total head the pump can produce as a result of both system friction and inertia at 100% speed. Head variation from *L* to *F* is a result of flow through the system, increasing at a decreasing rate of acceleration and increasing friction to the normal operating point *F*, where all the head is frictional. The actual total system-head resistance curve for any flow condition will be, therefore, the sum of the frictional resistance in feet (meters) for that steady-flow rate plus the inertial resistance, also expressed in feet (meters). The inertial resistance at any flow is dependent on the mass of liquid and the instantaneous rate of change of velocity at the flow condition. A typical total system resistance for a motor-driven propeller pump is shown as *OMNPQRF* in Figure 24. For a particular pump and system, different driver-speed-torque characteristics will result in a family of curves in the area *OLF*.

The added inertial system head produced momentarily when high-specific-speed pumps are started is important when considering the duration of high driver torques and currents, the pressure rise in the system, and the effect on the pump of operation at high heads and low flows. The approximate acceleration head h_a required to change the velocity of a mass of liquid at a uniform rate and cross section is

$$h_a = \frac{L\Delta V}{g\Delta t} \tag{10}$$

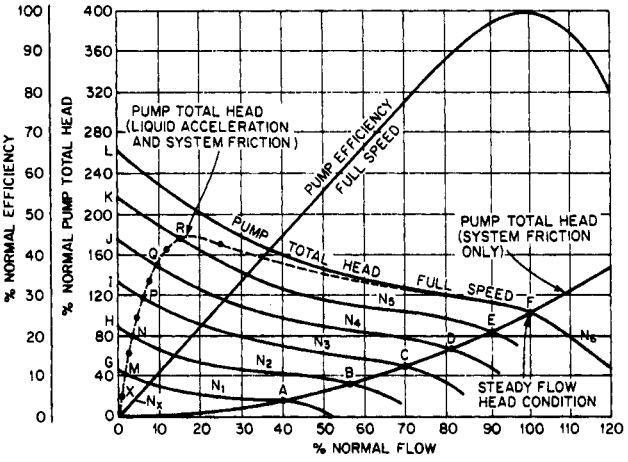


FIGURE 24 Transient system head as a result of liquid acceleration and system friction when a propeller pump is started with an open valve

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where L = length of constant-cross-section conduit, ft (m)
ΔV = velocity change, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)
Δt = time interval, s

To calculate the time to accelerate a centrifugal pump from rest or from some initial speed to a final speed, and to estimate the pump head variation during this interim, a trial-and-error solution may be used. Divide the speed change into several increments of equal no-flow heads, such as *OGHIKL* in Figure 24. For the first incremental speed change, point O to point N_1, estimate the total system head, point M, between G and A. Next estimate the total system head, point X, at the average speed for this first incremental speed change. These points are shown in Figure 24. Calculate the time in seconds for this incremental speed change to take place using the equation

in USCS units
$$\Delta t = \frac{\Sigma WK^2 \Delta N}{307(T_D - T_P)} \tag{11a}$$

where ΣWK^2 = total pump and motor rotor weight moment of inertia, W = weight (force), K = radius of gyration, lb · ft^2
ΔN = incremental speed change, rpm
T_D = motor torque at average speed, ft · lb
T_p = pump torque at average speed, ft · lb

in SI units
$$\Delta t = \frac{\Sigma MK^2 \Delta N}{9.55(T_D - T_P)} \tag{11b}$$

where ΣMK^2 = total pump and motor rotor mass moment of inertia, M = weight (mass), K = radius of gyration, kg · m^2
ΔN = incremental speed change, rpm
T_D = motor torque at average speed, N · m
T_P = pump torque at average speed, N · m

In SI units, when diameter of gyration D is used rather than radius of gyration K, and $MD^2 = 4MK^2$, then

$$\Delta t = \frac{\Sigma MD^2 \Delta N}{38.2(T_D - T_P)} \tag{11c}$$

Calculate the acceleration head required to change the flow in the system from point O to point M using Eq. 10 and time from Eq. 11. Add acceleration head to frictional head at the assumed average flow, and if this value is correct, it will fall on the average pump head-capacity curve, point X. Adjust points M and X until these assumed flows result in the total acceleration and frictional heads agreeing with flow X at the average speed. Repeat this procedure for other increments of speed change, adding incremental times to get total accelerating time to bring the pump up to its final speed. Plot system head versus average flow for each incremental speed change during this transient period, as shown in Figure 24.

Figures 25 and 26 illustrate how driver and pump torques can be determined from their respective speed-torque curves. Figure 25 is a family of curves that represents the torques required to produce flow against different heads without acceleration of the liquid or pump for the various speeds selected. Pump torque for any reduced speed can be calculated from the full-speed curve using the relation that torque varies as the second power and flow varies as the first power of the speed ratio. Point X is the torque at the average speed and the trial average flow during the first incremental speed change,

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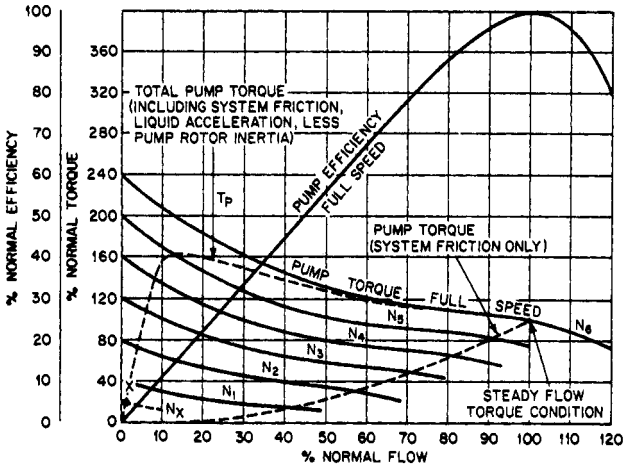


FIGURE 25 Transient total pump torque as a result of liquid acceleration and system friction when a propeller pump is started with an open valve. Pump head characteristics are shown in Figure 24.

which is adjusted for different assumed conditions. Figure 26 shows a typical squirrel-cage induction motor speed-torque curve, and, for the purpose of illustration, it has been selected to have the same torque rating as the pump requires at full speed (approximately 97% synchronous speed). Point X in this figure is the motor torque at the average speed during the first incremental speed change. In Figure 26, the developed torque curve for the different speeds shown in Figure 25 is redrawn as the total frictional and inertial pump torque T_p. For the conditions used in this example, and as shown in Figure 26, after approximately 88% synchronous speed, very little excess torque $(T_D - T_p)$ is available for

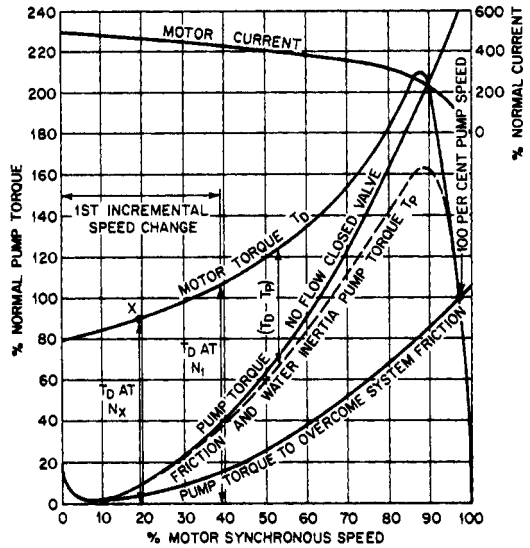


FIGURE 26 Propeller pump and motor speed-torque curves, showing effect of accelerating the liquid in the system during the starting period

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accelerating the pump and motor rotor inertia. However, as long as an induction motor has adequate torque to drive the pump at the normal condition, full speed will be reached providing the time-current demand can be tolerated. A synchronous motor, on the other hand, may not have sufficient torque to pull into step. Liquid acceleration decreases as motor torque decreases, adjusting to the available excess motor torque. The pump speed changes very slowly during the final period. Figure 26 also shows the motor current for the different speeds and torques.

A motor having a higher than normal pump torque rating or a motor having high starting torque characteristics will reduce the starting time, but higher heads will be produced during the acceleration period. Note, however, that heads produced cannot exceed shutoff at any speed. The total torque input to the pump shaft is equal to the sum of the torques required to overcome system friction, liquid inertia, and pump rotor inertia during acceleration. Torque at the pump shaft will therefore follow the motor speed-torque curve less the torque required to overcome the motor's rotor inertia. To reduce the starting inertial pump head to an acceptable amount, if desired, other alternative starting schemes can be used. A short bypass line from the pump discharge back to the suction can be provided to divert flow from the main system. The bypass valve is closed slowly after the motor reaches full speed. A variable-speed or a two-speed motor will reduce the inertial head by controlling motor torque and speed, thereby increasing the accelerating time.

This procedure for developing the actual system head and pump torque, including liquid inertia, becomes more complex if the pump must employ a discharge valve. To avoid high pump starting heads and torques, the discharge valve must be partially open on starting and then opened a sufficient amount before full speed is reached. The valve resistance must be added to the system friction and inertia curves if an exact solution is required.

Siphon Head Between any two points having the same elevation in a pumping system, no head is lost because of piping elevation changes because the net change in elevation is zero. If the net change in elevation between two points is not zero, additional pump head is required if there is an increase in elevation and less head is required if there is a decrease in elevation.

When piping is laid over and under obstacles with no net change in elevation, no pumping head is required to sustain flow other than that needed to overcome frictional and minor losses. As the piping rises, the liquid pressure head is transformed to elevation head, and the reverse takes place as the piping falls. A pipe or other closed conduit that rises and falls is called a *siphon,* and one that falls and rises is called an *inverted siphon.* The siphon principle is valid provided the conduit flows full and free of liquid vapor and air so the densities of the liquid columns are alike. It is this requirement that determines the limiting height of a siphon for complete recovery because the liquid can vaporize under certain conditions.

Pressure in a siphon is minimum at the summit, or just downstream from it, and Bernoulli's equation can be used to determine if the liquid pressure is above or below vapor pressure. Referring to Figure 27, observe the following. The absolute pressure head H_s in feet (meters) at the top of the siphon is

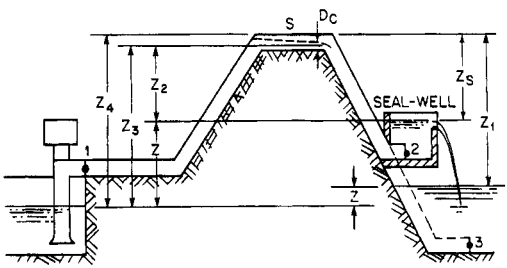


FIGURE 27 Pumping system using a siphon for head recovery

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$$H_S = H_B - Z_S + h_{f(S-2)} - \frac{V_S^2}{2g} \tag{12}$$

where H_B = barometric pressure head of liquid pumped, ft (m)

Z_S = siphon height to top of conduit (= Z_1 if no seal well is used), ft (m)

$h_{f(s-2)}$ = frictional and minor losses from S to 2 (or 3 if no seal well is used), including exit velocity head loss at 2 (or 3), ft (m)

$V^2_S/2g$ = velocity head at summit, ft (m)

The absolute pressure head at the summit can also be calculated using conditions in the up leg by adding the barometric pressure head to the pump head (TH) and deducting the distance from suction level to the top of the conduit (Z_4), the frictional loss in the up leg ($h_{f(1-S)}$), and the velocity head at the summit. If the suction level is higher than the discharge level and flow is by gravity, the absolute pressure head at the summit is found as above and $TH = 0$.

Whenever Z_1 in Figure 27 is so high that it exceeds the maximum siphon capability, a seal well is necessary to increase the pressure at the top of the siphon above vapor pressure. Note $Z_1 - Z_S$ represents an unrecoverable head and increases the pumping head. Water has a vapor pressure of 0.77 ft (23.5 cm) at 68°F (20°C) and theoretically a 33.23-ft-high (10.13-cm) siphon is possible with a 34-ft (10.36-m) water barometer. In practice, higher water temperatures and lower barometric pressures limit the height of siphons used in condenser cooling water systems to 26 to 28 ft (8 to 8.5 m). The siphon height can be found by using Eq. 12 and letting H_S equal the vapor pressure in feet (meters).

In addition to recovering head in systems such as condenser cooling water, thermal dilution, and levees, siphons are also used to prevent reverse flow after pumping is stopped by use of an automatic vacuum breaker located in the summit. Often siphons are used solely to eliminate the need for valves or flap gates.

In open-ended pumping systems, siphons can be primed by external means of air removal. Unless the siphon is primed initially upon starting, a pump must fill the system and provide a minimum flow to induce siphon action. During this filling period and until the siphon is primed, the siphon head curve must include this additional siphon filling head, which must be provided by the pump. Pumps in siphon systems are usually low-head, and they may not be capable of filling the system to the top of the siphon or of filling it with adequate flow. Low-head pumps are high-specific-speed and require more power at reduced flows than during normal pumping. Figure 28 illustrates the performance of a typical propeller pump when priming a siphon system and during normal operation.

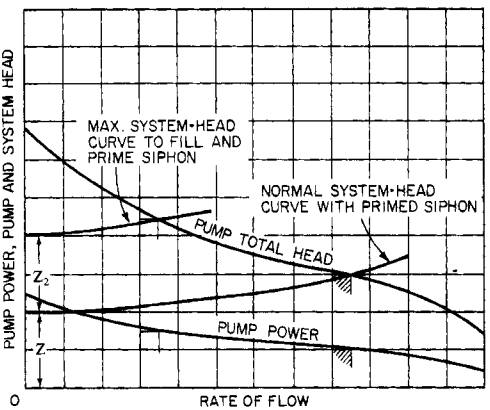


FIGURE 28 Transient system total head priming a siphon

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When a pump and driver are to be selected to prime a siphon system, it is necessary to estimate the pump head and the power required to produce the minimum flow needed to start the siphon.

The minimum flow required increases with the length and the diameter and decreases with the slope of the down-leg pipe.[2] Prior to the removal of all the air in the system, the pump is required to provide head to raise the liquid up to and over the siphon crest. Head above the crest is required to produce a minimum flow similar to flow over a broad-crested weir. This weir head may be an appreciable part of the total pump head if the pump is low-head and large-capacity. A conservative estimate of the pump head would include a full conduit above the siphon crest. In reality, the down leg must flow partially empty before it can flow full, and it is accurate enough to estimate that the depth of liquid above the siphon crest is at critical depth for the cross section. Table 1 can be used to estimate critical depth in circular pipes, and Figure 29 can be used to calculate the cross-sectional area of the filled pipe to determine the velocity at the siphon crest.

Until all the air is removed and all the piping becomes filled, the down leg is not part of the pumping system, and its frictional and minor losses are not to be added to the maximum system-head curve to fill and prime the siphon shown in Figure 28. The total head TH in feet (meters) to be produced by the pump in Figure 27 until the siphon is primed is

$$TH = Z_3 + h_{f(1-S)} + \frac{V_C^2}{2g} \tag{13}$$

where Z_3 = distance between suction level and centerline liquid at siphon crest, ft (m)

$h_{f(1-s)}$ = frictional and minor losses from 1 to S, ft (m)

$V_C^2/2g$ = velocity head at crest using actual liquid depth, approx. critical depth, ft (m)

Use of Eq. 13 permits plotting the maximum system-head curve to fill and prime the siphon for different flow rates. The pump priming flow is the intersection of the pump total head curve and this system-head curve. The pump selected must have a driver with power as shown in Figure 28 to prime the system during this transient condition.

For the pumping system shown in Figure 27, after the system is primed, the pump total head reduces to

$$TH = Z + h_{f(1-2)} \tag{14}$$

TABLE 1 Values for determining pipe-diameter ratio versus $(ft^3/s)/d^{5/2}$ in circular pipes

$\frac{D_{crit}}{d}$	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
0.0	...	0.0006	0.0025	0.0055	0.0098	0.0153	0.0220	0.0298	0.0389	0.0491
0.1	0.0605	0.0731	0.0868	0.1016	0.1176	0.1347	0.1530	0.1724	0.1928	0.2144
0.2	0.2371	0.2609	0.2857	0.3116	0.3386	0.3666	0.3957	0.4259	0.4571	0.4893
0.3	0.523	0.557	0.592	0.628	0.666	0.704	0.743	0.784	0.825	0.867
0.4	0.910	0.955	1.000	1.046	1.093	1.141	1.190	1.240	1.291	1.343
0.5	1.396	1.449	1.504	1.560	1.616	1.674	1.733	1.792	1.853	1.915
0.6	1.977	2.041	2.106	2.172	2.239	2.307	2.376	2.446	2.518	2.591
0.7	2.666	2.741	2.819	2.898	2.978	3.061	3.145	3.231	3.320	3.411
0.8	3.505	3.602	3.702	3.806	3.914	4.023	4.147	4.272	4.406	4.549
0.9	4.70	4.87	5.06	5.27	5.52	5.81	6.18	6.67	7.41	8.83

All tabulated values are in units of $(ft^3/s)/d^{5/2}$; d = diameter, ft; D_{crit} = critical depth, ft. For SI units, multiply $(m^3/h)/m^{5/2}$ by 5.03×10^{-4} to obtain $(ft^3/s)/ft^{5/2}$. Examples 2, 3, and 4 illustrate the use of this table.

Source: Reference 13.

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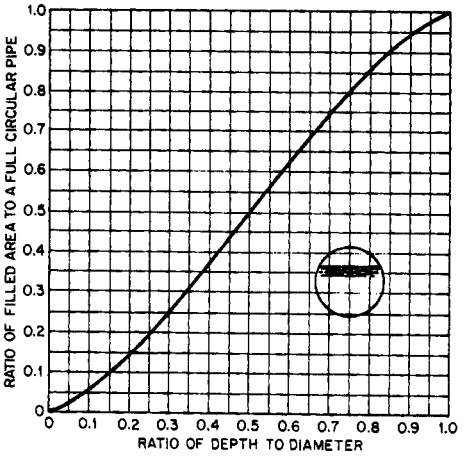


FIGURE 29 Area versus depth for a circular pipe

In SI units: $R = \frac{\rho VD}{\mu}$ (ρ in kg/m^3, V in m/s, D in m, μ in N·s/m^2)

1 meter = 3.28 feet

1 VD (V in m/s, D in m) = 129.2 VD'' (V in ft/s, D'' in inches)

where Z = distance between suction and seal well levels (or discharge pool level if no seal well is used), ft (m)

$h_{f1-2)}$ = frictional and minor losses from 1 to 2 (or 3 if no seal well is used), including exit velocity head loss at 2 (or 3), ft (m)

Use of Eq. 14 permits plotting of the normal system-head curve with a primed siphon, shown in Figure 28, for different flow rates. The normal pump flow is the intersection of the pump total head curve and the normal system-head curve.

If the pump cannot provide sufficient flow to prime the siphon, or if the driver does not have adequate power, the system must be primed externally by a vacuum or jet pump. An auxiliary priming pump can also be used to continuously vent the system because it is necessary that this be done to maintain full siphon recovery. In some systems, the water pumped is saturated with air, and as the liquid flows through the system, the pressure is reduced (and in cooling systems the temperature is increased). Both these conditions cause the release of some of the entrained air. Air will accumulate at the top of the siphon and in the upper parts of the down leg. The siphon works on the principle that an increase in elevation in the up leg produces a decrease in pressure and an equal decrease in elevation in the down leg, resulting in recovery of this pressure. This cannot occur if the density of the liquid in the down leg is decreased as a result of the formation of air pockets. These air pockets also restrict the flow area. A release of entrained air and air leakage into the system through pipe joints and fittings will result in a centrifugal pump's delivering less than design flow, as the head will be higher than estimated. Also, high-specific-speed pumps with rising power curves toward shutoff can become overloaded. In order to maintain full head recovery, it is necessary to continuously vent the siphon at the top and at several points along the down leg, especially at the beginning of a change in slope.[3] These venting points can be manifolded together and connected to a single downward venting system.

Some piping systems may contain several up and down legs; that is, several siphons in series. Each down leg, as in a single siphon, is vulnerable to air or vapor binding. The likelihood of flow reduction, and conceivably in some cases complete flow shutoff, is increased in a multiple siphon system.[3] As shown in Figure 30, system static head is increased if

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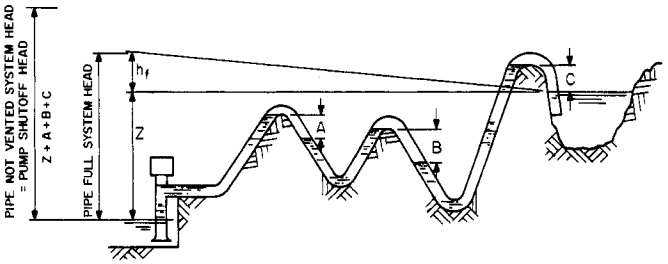


FIGURE 30 Multiple siphon system (Z = normal system static head when pipe is flowing full). Pump flow is stopped when normal static head plus sum of air pocket heights equals pump shutoff head.

proper venting is not provided at the top of each siphon. Normally the static head is the difference between outlet and inlet elevations. If air pockets exist, head cannot be recovered and the normal static head is increased by the sum of the heights of all the intermediate liquidless pockets. Flow will stop when the total static head equals the pump shutoff head.

The following examples illustrate the use of Eqs. 3, 9, 12, 13, and 14, Table 1, and Figure 29.

EXAMPLE 2 A pump is required to produce a flow of 70,000 gpm (15,900 m³/h) through the system shown in Figure 27. Calculate the system total head from point 1 to point 3 (no seal well) under the following conditions:

Specific gravity = 0.998 for 80°F (26.7°C) water

Barometric pressure = 29 in (73.7 cm) mercury abs (sp. gr. 13.6)

Suction and discharge water levels are equal, $Z = 0$

$Z_1 = 40$ ft (12.2 m)

$Hf_{(1-s)} = 3$ ft (0.91 m) up-leg frictional head

$h_{f(S-3)} = 3.3$ ft (1 m) down-leg frictional head, including exit loss

Pipe diameter = 48 in (121.9 cm) ID

Water vapor pressure = 0.507 lb/in² (3.5 kPa) abs at 80°F (26.7°C)

The maximum siphon height may be found from Eq. 12:

$$Z_s = H_B + h_{f(S-3)} - H_S + \frac{V_S^2}{2g}$$

From Eq. 3

H_B = barometric pressure head in feet (meters) of liquid pumped

In USCS units

$$\frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 = \frac{13.6}{0.998} \times \frac{29}{12} = 32.9 \text{ ft abs}$$

$$H_S = \frac{p}{\gamma} = \frac{144 \times 0.507}{62.19} = 1.17 \text{ ft abs}$$

In SI units

$$\frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 = \frac{13.6}{0.998} \times \frac{73.7}{100} = 10 \text{ m abs}$$

$$H_S = \frac{p}{\gamma} = \frac{3.5 \times 1000}{9769} = 0.358 \text{ m abs}$$

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From Eqs. 9 and 12,

in USCS units $$V_S = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{70{,}000}{48^2} \times 0.408 = 12.4 \text{ ft/s}$$

$$Z_S = 32.9 + 3.3 - 1.17 - \frac{12.4^2}{2 \times 32.17} = 32.63 \text{ ft}$$

in SI units $$V_S = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{15{,}900}{121.9^2} \times 3.54 = 3.79 \text{ m/s}$$

$$Z_S = 10 + 1 - 0.358 - \frac{3.79^2}{2 \times 9.807} = 9.1 \text{ m}$$

Because the maximum height is exceeded ($Z_1 > Z_s$), siphon recovery is not possible. The system total head is therefore found from Eq. 13:

$$TH = Z_3 + h_{f(1-S)} + \frac{V_c^2}{2g}$$

The critical depth D_{crit} is found using Table 1:

In USCS units $$\text{ft}^3/\text{s} = \frac{70{,}000}{7.481 \times 60} = 156$$

$$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = \frac{156}{4^{5/2}} = 4.88$$

$$\frac{D_{crit}}{d} = 0.901$$

$$D_{crit} = 0.901 \times 4 = 3.6 \text{ ft}$$

In SI units $$\frac{\text{m}^3/\text{h}}{d^{5/2}} = \frac{15{,}900}{1.219^{5/2}} = 9695$$

Convert to USCS units (see footnote to Table 1):

$$\frac{\text{ft}^3/\text{s}}{\text{ft}^{5/2}} = 9695 \times 5.03 \times 10^{-4} = 4.88$$

$$\frac{D_{crit}}{d} = 0.901$$

$$D_{crit} = 0.901 \times 1.219 = 1.1 \text{ m}$$

To calculate the water velocity at the siphon crest, determine the area of the filled pipe. From Figure 29, ratio of filled area to area of a full pipe is 0.95 for a depth-to-diameter ratio of 0.901:

In USCS units $$V_c = \frac{\text{gpm}}{0.95(\text{ID in inches})^2} \times 0.408 = \frac{70{,}000}{0.95 \times 48^2} \times 0.408 = 13.0 \text{ ft/s}$$

$$Z_3 = 40 - 4 + \frac{3.6}{2} = 37.8 \text{ ft}$$

From Eq. 13 $$TH = 37.8 + 3 + \frac{13.0^2}{2 \times 32.17} = 43.43 \text{ ft}$$

In SI units $$V_c = \frac{\text{m}^3/\text{h}}{0.95(ID \text{ in cm})^2} \times 3.54 = \frac{15{,}900}{0.95 \times 121.9^2} \times 3.54 = 3.99 \text{ m/s}$$

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$$Z_3 = 12.2 - 1.219 + \frac{1.1}{2} = 11.53 \text{ m}$$

From Eq. 13 $$TH = 11.53 + 0.91 + \frac{3.99^2}{2 \times 9.807} = 13.25 \text{ m}$$

EXAMPLE 3 Calculate the minimum total system head using conditions in Example 2 and a seal well, as shown in Figure 27. Use 2.8 ft (0.853 m) for the frictional head loss $h_{f(s-2)}$.

The maximum siphon height Z_s in Example 2 was found to be 32.63 ft (9.95 m). Therefore from Eq. 14 the total system head after priming is

$$TH = Z + h_{f(1-2)}$$

In USCS units $$h_{f(1-2)} = h_{f(1-S)} + h_{f(S-2)} = 3 + 2.8 = 5.8 \text{ ft}$$

$$Z = Z_1 - Z_S = 40 - 32.63 = 7.37 \text{ ft}$$

In SI units $$h_{f(1-2)} = h_{f(1-S)} + h_{f(S-2)} = 0.91 + 0.853 = 1.763 \text{ ft}$$

$$Z = Z_1 - Z_S = 12.2 - 9.95 = 2.25 \text{ m}$$

Note that the seal well elevation is above discharge level. Therefore

in USCS units $$TH = 7.37 + 5.8 = 13.17 \text{ ft}$$

in SI units $$TH = 2.25 + 1.763 = 4.01 \text{ m}$$

EXAMPLE 4 The dimensions of the down leg in Example 3 require a minimum velocity of 5 ft/s (1.52 m/s) flowing full to purge air from the system and start the siphon. Calculate the system head the pump must overcome to prime the siphon.

In USCS units $$\text{gpm} = \frac{V(\text{pipe } ID \text{ in inches})^2}{0.408} = \frac{5 \times 48^2}{0.408} = 28{,}200$$

$$\text{ft}^3/\text{s} = 28{,}200 \div 449 = 62.8$$

In SI units $$\text{m}^3/\text{h} = \frac{V(\text{pipe } ID \text{ in cm})^2}{3.54} = \frac{1.52 \times 121.9^2}{3.54} = 6400$$

The critical depth is found from Table 1:

In USCS units $$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = \frac{62.8}{4^{5/2}} = 1.97$$

$$\frac{D_{\text{crit}}}{d} = 0.6$$

$$D_{\text{crit}} = 0.6 \times 4 = 2.4 \text{ ft}$$

In SI units $$\frac{\text{m}^3/\text{h}}{d^{5/2}} = \frac{6400}{1.219^{5/2}} = 3902$$

Convert to USCS units (see footnote to Table 1):

$$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = 3902 \times 5.03 \times 10^{-4} = 1.97$$

$$\frac{D_{\text{crit}}}{d} = 0.6$$

$$D_{\text{crit}} = 0.6 \times 1.219 = 0.73 \text{ m}$$

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From Figure 29, the ratio of the filled area to the area of a full pipe is 0.625 for a depth-to-diameter ratio of 0.60:

In USCS units $$V_c = \frac{5}{0.625} = 8.0 \text{ ft/s}$$

In SI units $$V_c = \frac{1.52}{0.625} = 2.4 \text{ m/s}$$

From Eq. 13 $$TH = Z_3 + h_{f(1-S)} + \frac{V_c^2}{2g}$$

In USCS units $$Z_2 = Z_S - 4 + \frac{D_{crit}}{2} = 32.63 - 4 + \frac{2.4}{2} = 29.83 \text{ ft}$$

$Z = 7.37$ ft (from Example 3)

$$h_{f(1-S)} \text{ at } 28{,}200 \text{ gpm} = \left(\frac{28{,}200}{70{,}000}\right)^2 \times 3 = 0.49 \text{ ft}$$

$$TH = (7.37 + 29.83) + 0.49 + \frac{8^2}{2 \times 32.17} = 38.68 \text{ ft}$$

In SI units $$Z_2 = Z_S - 1.219 + \frac{D_{crit}}{2} = 9.95 - 1.219 + \frac{0.73}{2} = 9.1 \text{ m}$$

$Z = 2.25$ m (from Example 3)

$$h_{f(1-S)} \text{ at } 6400 \text{ m}^3\text{/h} = \left(\frac{6400}{15{,}900}\right)^2 \times 0.91 = 0.15 \text{ m}$$

$$TH = (2.25 + 9.1) + 0.15 + \frac{2.4^2}{2 \times 9.807} = 11.79 \text{ m}$$

If the system is not externally primed, the centrifugal pump selected must be able to deliver at least 28,000 gpm (6400 m³/h) at 38.68 ft (11.79 m) total head and must be provided with a driver having adequate power for this condition. After the system is primed, the pump must be capable of delivering at least 70,000 gpm (15,900 m³/h) at 13.17 ft (4.01 m) (see Figure 28).

HEAD LOSSES IN SYSTEM COMPONENTS

Pressure Pipes Resistance to flow through a pipe is caused by viscous shear stresses in the liquid and by turbulence at the pipe walls. *Laminar* flow occurs in a pipe when the average velocity is relatively low and the energy head is lost mainly as a result of viscosity. In laminar flow, liquid particles have no motion next to the pipe walls and flow occurs as a result of the movement of particles in parallel lines with velocity increasing toward the center. The movement of concentric cylinders past each other causes viscous shear stresses, more commonly called *friction*. As flow increases, the flow pattern changes, the average velocity becomes more uniform, and there is less viscous shear. As the laminar film decreases in thickness at the pipe walls and as the flow increases, the pipe roughness becomes important because it causes turbulence. *Turbulent* flow occurs when average pipe velocity is relatively high and energy head is lost predominantly because of turbulence caused by the wall roughness. The average velocity at which the flow changes from laminar to turbulent is not definite, and there is a critical zone in which either laminar or turbulent flow can occur.

Viscosity can be visualized as follows. If the space between two planar surfaces is filled with a liquid, a force will be required to move one surface at a constant velocity relative to the other. The velocity of the liquid will vary linearly between the surfaces. The ratio of the force per unit area, called *shear stress*, to the velocity per unit distance between surfaces, called *shear* or *deformation rate*, is a measure of a liquid's *dynamic* or *absolute viscosity*.

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Liquids such as water and mineral oil, which exhibit shear stresses proportional to shear rates, have a constant viscosity for a particular temperature and pressure and are called *Newtonian* or *true liquids.* In the normal pumping range, however, the viscosity of true liquids may be considered independent of pressure. For these liquids, the viscosity remains constant because the rate of deformation is directly proportional to the shearing stress. The viscosity and resistance to flow, however, increase with decreasing temperature.

Liquids such as molasses, grease, starch, paint, asphalt, and tar behave differently from Newtonian liquids. The viscosity of the former does not remain constant and their shear, or deformation, rate increases more than the stress increases. These liquids, called *thixotropic,* exhibit lower viscosity as they are agitated at a constant temperature.

Still other liquids, such as mineral slurries, show an increase in viscosity as the shear rate is increased and are called *dilatant.*

In USCS units, dynamic (absolute) viscosity is measured in pound-seconds per square foot or slugs per foot-second. In SI measure, the units are newton-seconds per square meter or pascal-seconds. Usually dynamic viscosity is measured in *poises* (1 P = 0.1 Pa · s) or in *centipoises* (1 cP = $\frac{1}{100}$ P):

$$1 \text{ lb} \cdot \text{s/ft}^2 = 47.8801 \text{ Pa} \cdot \text{s} = 47{,}880.1 \text{ cP}$$

The viscous property of a liquid is also sometimes expressed as *kinematic viscosity.* This is the dynamic viscosity divided by the mass density (specific weight/g). In USCS units, kinematic viscosity is measured in square feet per second. In SI measure, the units are square meters per second. Usually kinematic viscosity is measured in *stokes* (1 St = 0.0001 m^2/s) or in *centistokes* (1 cSt = $\frac{1}{100}$ St):

$$1 \text{ ft}^2/\text{s} = 0.0929034 \text{ m}^2/\text{s} = 92{,}903.4 \text{ cSt}$$

A common unit of kinematic viscosity in the United States is Saybolt seconds universal (SSU) for liquids of medium viscosity and Saybolt seconds Furol (SSF) for liquids of high viscosity. Viscosities measured in these units are determined by using an instrument that measures the length of time needed to discharge a standard volume of the sample. Water at 60°F (15.6°C) has a kinematic viscosity of approximately 31 SSU (1.0 cSt). For values of 70 cSt and above,

$$\text{cSt} = 0.216 \text{ SSU}$$

$$\text{SSU} = 10 \text{ SSF}$$

The dimensionless Reynolds number Re is used to describe the type of flow in a pipe flowing full and can be expressed as follows:

$$Re = \frac{VD}{\upsilon} = \frac{\rho VD}{\mu} \tag{15}$$

where V = average pipe velocity, ft/s (m/s)

D = inside pipe diameter, ft (m)

υ = liquid kinematic viscosity, ft^2/s (m^2/s)

ρ = liquid density, slugs/ft^3 (kg/m^3)

μ = liquid dynamic (or absolute) viscosity slug/ft · s (N · s/m^2)

Note: The dimensionless Reynolds number is the same in both USCS and SI units.

When the Reynolds number is 2000 or less, the flow is generally laminar, and when it is greater than 4000, the flow is generally turbulent. The Reynolds number for the flow of water in pipes is usually well above 4000, and therefore the flow is almost always turbulent.

The Darcy-Weisbach formula is the one most often used to calculate pipe friction. This formula recognizes that friction increases with pipe wall roughness, with wetted surface area, with velocity to a power, and with viscosity and decreases with pipe diameter to a power and with density. Specifically, the frictional head loss h_f in feet (meters) is

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$$h_f = f\frac{L}{D}\frac{V^2}{2g} \tag{16}$$

where f = friction factor
L = pipe length, ft (m)
D = inside pipe diameter, ft (m)
V = average pipe velocity, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

For laminar flow, the friction factor f is equal to $64/Re$ and is independent of pipe wall roughness. For turbulent flow, f for all incompressible fluids can be determined from the well-known Moody diagram, shown in Figure 31. To determine f, it is required that the Reynolds number and the relative pipe roughness be known. Values of relative roughness (ϵ/D), where ϵ is a measure of pipe wall roughness height in feet (meters), can be obtained from Figure 32 for different pipe diameters and materials. Figure 32 also gives values for f for the flow of 60°F (15.6°C) water in rough pipes with complete turbulence. Values of kinematic viscosity and Reynolds numbers for a number of different liquids at various temperatures are given in Figure 33. The Reynolds numbers of 60°F (15.6°C) water for various velocities and pipe diameters may be found by using the VD'' scale in Figure 31.

There are many empirical formulas for calculating pipe friction for water flowing under turbulent conditions. The most widely used is the Hazen-Williams formula:

In USCS units $$V = 1.318Cr^{0.63}S^{0.54} \tag{17a}$$

In SI units $$V = 0.8492Cr^{0.63}S^{0.54} \tag{17b}$$

where V = average pipe velocity, ft/s (m/s)
C = friction factor for this formula, which depends on roughness only
r = hydraulic radius (liquid area divided by wetted perimeter) or $D/4$ for a full pipe, ft (m)
S = hydraulic gradient or frictional head loss per unit length of pipe, ft/ft (m/m)

The effect of age on a pipe should be taken into consideration when estimating the frictional loss. A lower C value should be used, depending on the expected life of the system. Table 2 gives recommended friction factors for new and old pipes. A value of C of 150 may be used for plastic pipe. Figure 34 is a nomogram that can be used in conjunction with Table 2 for a solution to the Hazen-Williams formula.

The frictional head loss in pressure pipes can be found by using either the Darcy-Weisbach formula (Eq. 16) or the Hazen-Williams formula (Eq. 17). Tables in the appendix give Darcy-Weisbach friction values for Schedule 40 new steel pipe carrying water. Tables are also provided for losses in old cast iron piping based on the Hazen-Williams formula with $C = 100$. In addition, values of C for various pipe materials, conditions, and years of service can also be found in the appendix.

The following examples illustrate how Figures 31, 32, and 33 and Table 2 may be used.

EXAMPLE 5 Calculate the Reynolds number for 175°F (79.4°C) kerosene flowing through 4-in (10.16-cm), Schedule 40, 3.426-in (8.70-cm) ID, seamless steel pipe at a velocity of 14.6 ft/s (4.45 m/s).

In USCS units $VD'' = 14.6 \times 3.426 = 50$ ft/s × in

In SI units $VD = 4.45 \times 0.087 = 0.387$ m/s × m $= 0.387 \times 129.2 = 50$ ft/s × in

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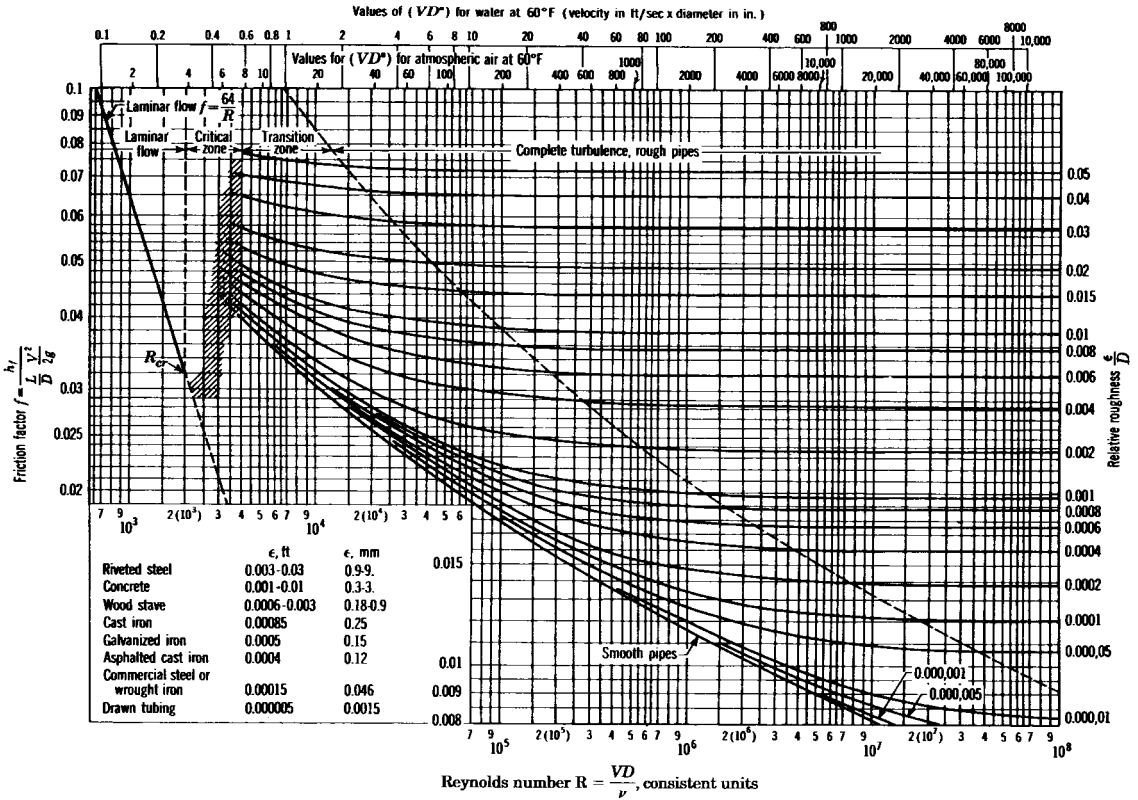


FIGURE 31 Moody diagram. (Reference 14) In SI units: $\mathbf{R} = \frac{\rho VD}{\mu}$ (ρ in kg /m^3, V in m/s, D in m, μ in N • s/m^2.) 1 meter = 3.28 ft; 1VD (V in m/s, D in m) = 129.2 VD' (V in ft/s, D' in inches).

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TABLE 2 Values of friction factor C to be used with the Hazen-Williams formula in Figure 34

Type of pipe	Age	Size, in[a]	C
Cast iron	New	All sizes	130
	5 years old	12 and over	120
		8	119
		4	118
	10 years old	24 and over	113
		12	111
		4	107
	20 years old	24 and over	100
		12	96
		4	89
	30 years old	30 and over	
		16	87
		4	75
	40 years old	30 and over	83
		16	80
		4	64
	40 and over	77	
		24	74
		4	55
Welded steel	Any age, any size		Same as for cast iron pipe 5 years old
Riveted steel	Any age, any size		Same as cast iron pipe 10 years older
Wood-stave	Average value, regardless of age and size		120
Concrete or	Large sizes, good workmanship, steel forms		140
concrete-lined	Large sizes, good workmanship, wooden forms		120
Vitrified	Centrifugally spun		135
	In good condition		110

[a] In × 25.4 = mm

Source: Adapted From Reference 15.

Follow the tracer lines in Figure 33 and read directly:

$$Re = 3.5 \times 10^5$$

EXAMPLE 6 Calculate the frictional head loss for 100 ft (30.48 m) of 20-in (50.8-cm), Schedule 20, 19.350-in (49.15-cm) ID, seamless steel pipe for 109°F (42.8°C) water flowing at a rate of 11,500 gpm (2612 m³/h). Use the Darcy-Weisbach formula.

In USCS Units

$$V = \frac{\text{gpm}}{(\text{pipe } ID \text{ in inches})^2} \times 0.408 = \frac{11{,}500}{19.35^2} \times 0.408 = 12.53 \text{ ft/s}$$

$$VD'' = 12.53 \times 19.35 = 242 \text{ ft/s} \times \text{in}$$

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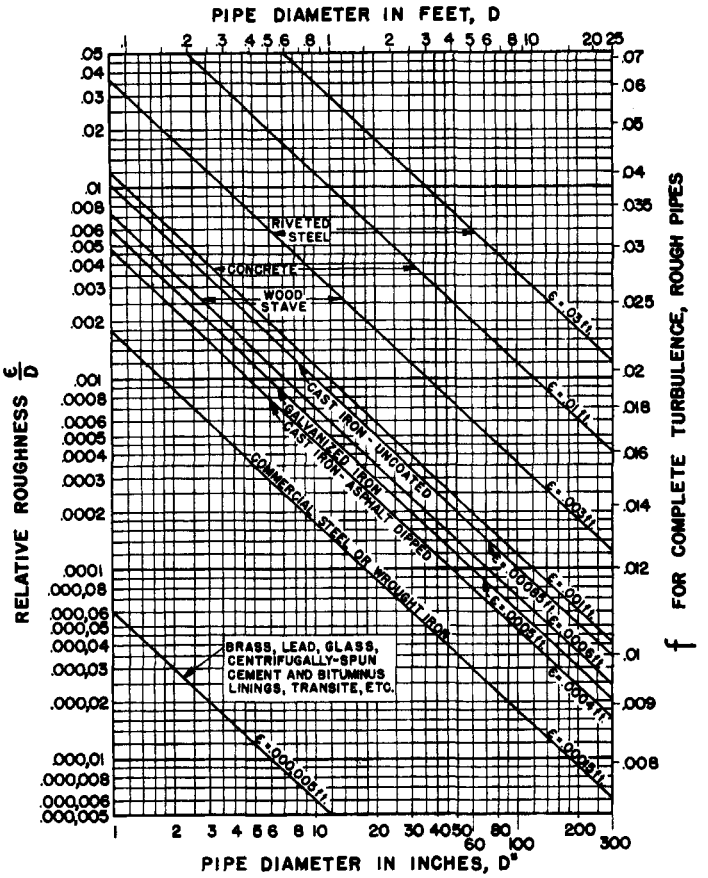


FIGURE 32 Relative roughness and friction factors for new, clean pipes for flow of 60°F (15.6°C) water (Hydraulic Institute Engineering Data Book, Reference 5) (1 meter 39.37 in = 3.28 ft)

In SI units

$$V = \frac{\text{m}^3/\text{h}}{(\text{pipe } ID \text{ in cm})} \times 3.54 = \frac{2612}{49.15^2} \times 3.54 = 3.83 \text{ m/s}$$

$$VD = 3.83 \times 0.4915 = 1.88 \text{ m/s} \times \text{m} = 1.88 \times 129.2 = 242 \text{ ft/s} \times \text{in}$$

From Figure 33 $Re = 3 \times 10^6$

From Figure 32 $\frac{\epsilon}{D} = 0.00009$

From Figure 31 $f = 0.012$

Using Eq. 16,

In USCS units $D = \frac{19.35}{12} = 1.61 \text{ ft}$

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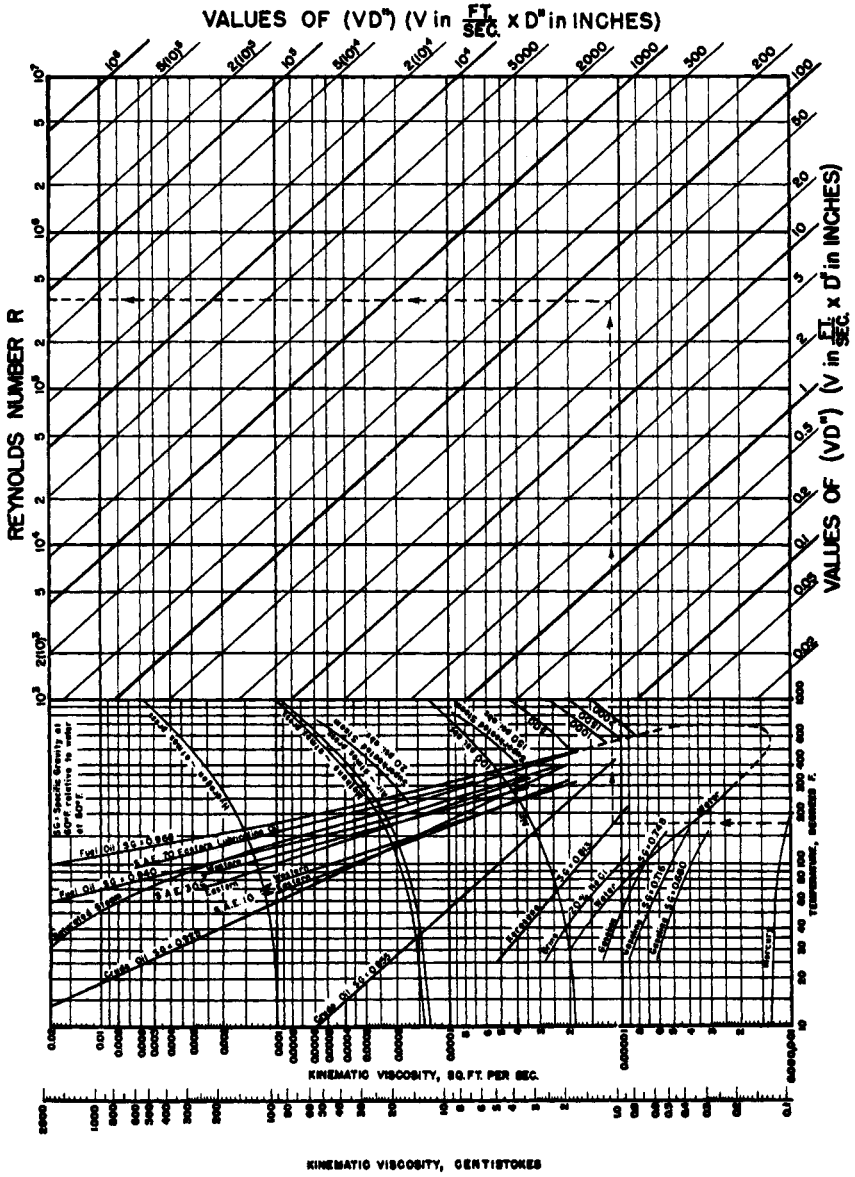


FIGURE 33 Kinematic viscosity and Reynolds number. (Hydraulic Institute Engineering Data Book, Reference 5) [1 ft^2/s = 0.0929 m^2/s; 1 cSt = 1.0 × 10^{-6} m^2/s; 1*VD* (m/s × m) = 129.2 VD" (ft/s × in); °F = (°C + 18) 1.8]

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$$h_f = f\frac{L}{d}\frac{V^2}{2g} = 0.012\,\frac{100}{1.61} \times \frac{12.53^2}{2 \times 32.17} = 1.82 \text{ ft}$$

In SI units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.012\,\frac{30.48}{0.4915} \times \frac{3.83^2}{2 \times 9.807} = 0.556 \text{ m}$$

EXAMPLE 7 The flow in Example 6 is increased until complete turbulence results. Determine the friction factor f and flow.

From Figure 31, follow the relative roughness curve $\varepsilon/D = 0.00009$ to the beginning of the zone marked "complete turbulence, rough pipes" and read

$$f = 0.0119 \quad \text{at } Re = 2 \times 10^7$$

The problem may also be solved using Figure 32. Enter relative roughness $\varepsilon/D = 0.00009$ and read directly across to

$$f = 0.0119$$

An increase in Re from 3×10^6 to 2×10^7 would require an increase in flow to

in USCS units $$\frac{2 \times 10^7}{3 \times 10^6} \times 11{,}500 = 76{,}700 \text{ gpm}$$

in SI units $$\frac{2 \times 10^7}{3 \times 10^6} \times 2612 = 17{,}413 \text{ m}^3/\text{h}$$

EXAMPLE 8 The liquid in Example 6 is changed to water at 60°F (15.6°C). Determine Re, f, and the frictional head loss per 100 ft (100 m) of pipe.

$$VD'' = 242 \text{ ft/s} \times \text{in} \qquad \text{(as in Example 6)}$$

Because the liquid is 60°F (15.6°C) water, enter Figure 31 and read directly downward from VD'' to

$$Re = 1.8 \times 10^6$$

Where the line VD'' to Re crosses $\varepsilon/D = 0.00009$ in Figure 31, read

$$f = 0.013$$

Water at 60°F (15.6°C) is more viscous than 109°F (42.8°C) water, and this accounts for the fact that Re decreases and f increases. Using Eq. 16, it can be calculated that the frictional head loss increases to

in USCS units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.013\,\frac{100}{1.61} \times \frac{12.53^2}{2 \times 32.17} = 1.97 \text{ ft}$$

in SI units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.013\,\frac{100}{0.4915} \times \frac{3.83^2}{2 \times 9.807} = 1.97 \text{ m}$$

EXAMPLE 9 A 102-in (259-cm) ID welded steel pipe is to be used to convey water at a velocity of 11.9 ft/s (3.63 m/s). Calculate the expected loss of head due to friction per 1000 ft and per 1000 m of pipe after 20 years. Use the empirical Hazen-Williams formula.

From Table 2, $C = 100$.

In USCS units $$r = \frac{D}{4} = \frac{102}{(4 \times 12)} = 2.13 \text{ ft}$$

In SI units $$r = \frac{D}{4} = \frac{2.59}{4} = 0.648 \text{ m}$$

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Substituting in Eq. 17,

in USCS units
$$S^{0.54} = \frac{V}{1.318Cr^{0.63}} = \frac{11.9}{1.318 \times 100 \times 2.13^{0.63}} = 0.0557$$
$$S = (0.0557)^{1/0.54} = 0.0048 \text{ ft/ft}$$
$$h_f = 1000 \times 0.0048 = 4.8 \text{ ft}$$

in SI units
$$S^{0.54} = \frac{V}{0.8492Cr^{0.63}} = \frac{3.63}{0.8492 \times 100 \times 0.648^{0.63}} = 0.0562$$
$$S = (0.0562)^{1/0.54} = 0.0048 \text{ m/m}$$
$$h_f = 1000 \times 0.0048 = 4.8 \text{ m}$$

The problem may also be solved by using Figure 34, following the trace lines:

$$h_f \approx 5 \text{ ft (m)}$$

Frictional Loss for Viscous Liquids Table 3 gives the frictional loss for viscous liquids flowing in new Schedule 40 steel pipe. Values of pressure loss are given for both laminar and turbulent flows.

For laminar flow, the pressure loss is directly proportional to the viscosity and the velocity of flow and inversely proportional to the pipe diameter to the fourth power. Therefore, for intermediate values of viscosity and flow, obtain the pressure loss by direct interpolation.

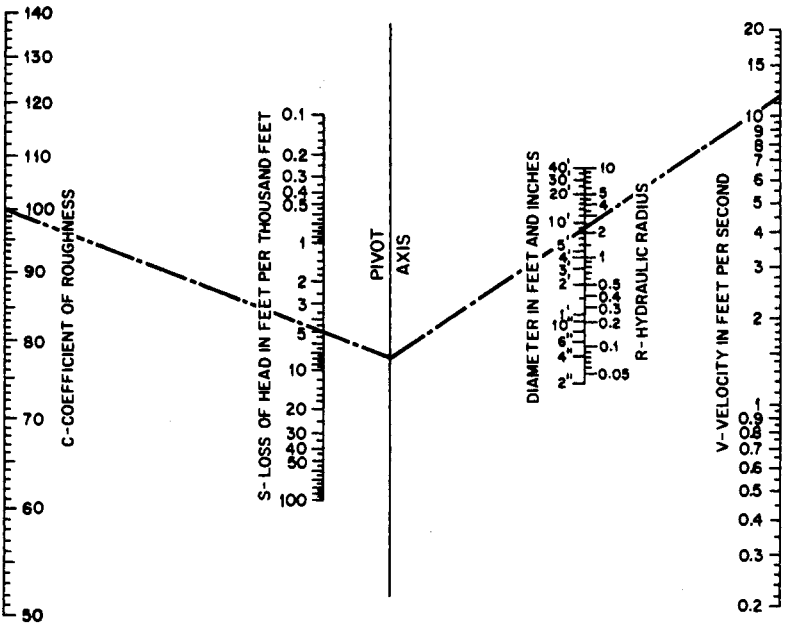


FIGURE 34 Nomogram for the solution of the the Hazen-Williams formula. Obtain values for C from Table 2 (Reference 15) (1 m/s = 3.28 ft/s; 1 m = 39.37 in)

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For pipe sizes not shown, multiply the fourth power of the ratio of any tabulated diameter to the pipe diameter wanted by the tabulated loss shown. The flow rate and viscosity must be the same for both diameters.

For turbulent flow and for rates of flow and pipe sizes not tabulated, the following procedures may be followed. For the viscosity and pipe size required, an intermediate flow loss is found by selecting the pressure loss for the next lower flow and multiplying by the square of the ratio of actual to tabulated flow rates. For the viscosity and flow required, an intermediate pipe diameter flow loss is found by selecting the pressure loss for the next smaller diameter and multiplying by the fifth power of the ratio of tabulated to actual inside diameters.

The viscosity of various common liquids can be found in tables in the appendix.

Partially Full Pipes and Open Channels Another popular empirical equation applicable to the flow of water in pipes flowing full or partially full or in open channels is the Manning formula:

In USCS units
$$V = \frac{1.486}{n} r^{2/3} S^{1/2} \tag{18a}$$

In SI units
$$V = \frac{r^{2/3} S^{1/2}}{n} \tag{18b}$$

where V = average velocity, ft/s (m/s)
n = friction factor for this formula, which depends on roughness only
r = hydraulic radius (liquid area divided by wetted perimeter), ft (m)
S = hydraulic gradient or frictional head loss per unit length of conduit, ft/ft (m/m)

The Manning formula nomogram shown in Figure 35 can be used to determine the flow or frictional head loss in open or closed conduits. Note that the hydraulic gradient S in Figure 35 is plotted in feet per 100 ft of conduit length. Values of friction factor n are given in Table 4.

If the conduit is flowing partially full, computing the hydraulic radius is sometimes difficult. When the problem to be solved deals with a pipe that is not flowing full, Figure 36 may be used to obtain multipliers for correcting the flow and velocity of a full pipe to the values needed for the actual fill condition. If the flow in a partially full pipe is known and the frictional head loss is to be determined, Figure 36 is first used to correct the flow to what it would be if the pipe were full. Then Eq. 18 or Figure 35 is used to determine the frictional head loss (which is also the hydraulic gradient and the slope of the pipe). The problem is solved in reverse if the hydraulic gradient is known and the flow is to be determined.

For full or partially full flow in conduits that are not circular in cross section, an alternate solution to using Eq. 18 is to calculate an equivalent diameter equal to four times the hydraulic radius. If the conduit is extremely narrow and width is small relative to length (annular or elongated sections), the hydraulic radius is one-half the width of the section.[4] After the equivalent diameter has been determined, the problem may be solved by using the Darcy-Weisbach formula (Eq. 16).

The hydraulic gradient in a uniform open channel is synonymous with frictional head loss in a pressure pipe. The hydraulic gradient of an open channel or of a pipe flowing partially full is the slope of the free liquid surface. In the reach of the channel where the flow is uniform, the hydraulic gradient is parallel to the slope of the channel bottom. Figure 37 shows that, in a pressure pipe of uniform cross section, the slope of both the energy and hydraulic gradients is a measure of the frictional head loss per foot (meter) of pipe between points 1 and 2. Figure 38 illustrates the flow in an open channel of varying slope. Between points 1 and 2, the flow is uniform and the liquid surface (hydraulic gradient) and channel bottom are both parallel and their slope is the frictional head loss per foot (meter) of channel length.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5)*

gpm	Pipe size	Viscosity, SSU 100	200	300	400	500	1000	1500	2000
	½	11.2	23.6	35.3	47.1	59	118	177	236
3	¾	3.7	7.6	11.5	15.3	19.1	38.2	57	76
	1	1.4	2.9	4.4	5.8	7.3	14.5	21.8	29.1
	¾	6.1	12.7	19.1	25.5	31.9	61	96	127
5	1	2.3	4.9	7.3	9.7	12.1	21.2	36.3	48.5
	1¼	0.77	1.6	2.4	3.3	4.1	8.1	12.2	16.2
	¾	8.5	17.9	26.8	35.7	44.6	89	134	178
7	1	3.2	6.8	10.2	13.6	17	33.9	51	68
	1¼	1.1	2.3	3.4	4.5	5.7	11.4	17	22.7
	1	4.9	9.7	14.5	19.4	24.2	48.5	73	97
10	1¼	1.6	3.3	4.9	6.5	8.1	16.2	24.3	32.5
	1½	0.84	1.8	2.6	3.5	4.4	8.8	13.1	17.5
	1	11	14.5	21.8	29.1	36.3	73	109	145
15	1¼	2.8	4.9	7.3	9.7	12.2	24.3	36.5	48.7
	1½	1.3	2.6	3.9	5.3	6.6	13.1	197	26.3
	1	18	18	29.1	38.8	48.5	97	145	194
	1¼	4.9	6.4	9.7	13	16.2	32.5	48.7	65
20	1½	2.3	3.5	5.3	7	8.8	17.5	26.3	35
	2	0.64	1.3	1.9	2.6	3.2	6.4	9.6	12.9
	1½	3.5	4.4	6.6	8.8	11	21.9	32.8	43.8
25	2	1	1.6	2.4	3.2	4	8	12.1	16.1
	2½	0.4	0.79	1.2	1.6	2	4	5.9	7.9
	1½	5	5.3	7.9	10.5	13.1	26.3	39.4	53
30	2	1.4	1.9	2.9	3.9	4.8	9.6	14.5	19.3
	2½	0.6	0.95	1.4	1.9	2.4	4.7	7.1	9.5
	1½	8.5	9	10.5	14	17.5	35	53	70
40	2	2.5	2.5	3.9	5.1	6.4	12.9	19.3	25.7
	2½	1.1	1.3	1.9	2.5	3.2	6.3	9.5	12.6
	1½	12.5	14	14	17.5	21.9	43.8	66	88
50	2	3.7	4	4.8	6.4	8	16.1	24.1	32.1
	2½	1.6	1.7	2.4	3.2	4	7.9	11.8	15.8
	2	5	5.8	5.8	7.7	9.6	19.3	28.9	38.5
60	2½	2.2	2.4	2.8	3.8	4.7	9.5	14.2	19
	3	0.8	0.8	1.2	1.6	2	4	6	8
	2½	2.8	3.2	3.4	4.4	5.5	11.1	16.6	22.1
70	3	1	1.1	1.4	1.9	2.3	4.6	7	9.3
	4	0.27	0.31	0.47	0.63	0.78	1.6	2.4	3.1
	2½	3.6	4.2	4.2	5.1	6.3	12.6	19	25.3
80	3	1.3	1.4	1.6	2.1	2.7	5.3	8	10.6
	4	0.36	0.36	0.54	0.72	0.89	1.8	2.7	3.6
	2½	5.3	6.1	6.4	6.4	8	15.8	23.7	31.6
100	3	1.9	2.2	2.2	2.7	3.3	6.6	9.9	13.3
	4	0.52	0.57	0.67	0.89	1.1	2.2	3.4	4.5

← TURBULENT FLOW → ← LAMINAR FLOW →

*Loss in pounds per square inch per 100 ft of new Schedule 40 steel pipe based on specific gravity of 1.00 of that liquid. For commercial installations, it is recommended that 15% be added to the values in this table.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5)* (*Continued*)

Viscosity, SSU									
2500	3000	4000	5000	6000	7000	8000	9000	10,000	15,000
294	353	471	589	706	824	942	...	...	...
96	115	153	191	229	268	306	344	382	573
36.3	43.6	58	73	87	101	116	131	145	218
159	191	255	319	382	446	510	573	637	956
61	73	97	121	145	170	194	218	242	363
20.3	24.3	32.5	40.6	48.7	57	65	73	81	122
223	268	357	416	535	624	713	803	892	...
85	102	136	170	203	237	271	305	339	509
28.4	34.1	45.4	57	68	80	91	102	114	170
121	145	194	242	291	339	388	436	485	727
40.6	48.7	65	81	97	114	130	146	162	243
21.9	26.3	35	43.8	53	61	70	79	88	131
182	218	291	363	436	509	581	654	727	...
61	73	97	122	146	170	195	219	243	365
32.8	39.4	53	66	79	92	105	118	131	197
242	291	388	485	581	678	775	872	...	...
81	97	130	162	195	227	260	292	325	487
43.8	53	70	88	105	123	140	158	175	263
16.1	19.3	25.7	32.1	38.5	45	51	58	64	96
55	66	88	110	131	153	176	197	219	328
20.1	24.1	32.1	40.2	48.2	56	61	72	80	121
9.9	11.8	15.8	19.7	23.7	27.6	31.6	35.5	39.5	59
66	79	105	131	158	184	210	237	263	394
24.1	28.9	38.5	48.2	58	67	77	87	96	145
11.8	14.2	19	23.7	28.4	33.2	37.9	42.6	47.4	71
88	105	140	175	210	245	280	315	350	526
32.1	38.5	51	64	77	90	103	116	129	193
15.8	19	25.3	31.6	37.9	44.2	51	57	63	95
110	131	175	219	263	307	350	394	438	657
40.2	48.2	64	80	96	112	129	145	161	241
19.7	23.7	31.6	39.5	47.4	55	63	71	79	118
48.2	58	77	96	116	135	154	173	193	289
23.7	28.4	37.9	47.4	57	66	76	85	95	142
9.9	11.9	15.9	19.9	23.9	27.9	31.8	35.8	39.8	60
27.6	33.2	44.2	55	66	77	88	100	111	166
11.6	13.9	18.6	23.2	27.8	32.5	37.1	41.7	46.4	70
3.9	4.7	6.3	7.8	9.4	11	12.5	14.1	15.6	23.5
31.6	37.9	51	63	76	88	101	114	126	190
13.3	15.9	21.2	26.5	31.8	37.1	42.4	47.7	53	80
4.5	5.4	7.2	8.9	10.7	12.5	14.3	16.1	17.9	26.8
39.5	47.4	63	79	95	111	127	142	158	237
16.6	19.9	26.5	33.1	39.8	46.4	53	60	66	99
5.6	6.7	8.9	11.2	13.4	15.6	17.9	20.1	22.3	33.5

← LAMINAR FLOW →

*For a liquid having a specific gravity other than 1.00, mulitply the value from the table by the specific gravity. No allowance for aging of pipe is included.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU 100	200	300	400	500	1000	1500	2000
	3	2.7	3.1	3.2	3.2	4	8	11.9	15.9
120	4	0.73	0.81	0.81	1.1	1.3	2.7	4	5.4
	6	0.98	0.11	0.16	0.21	0.26	0.52	0.78	1.0
	3	3.4	4	4.3	4.3	4.6	9.3	13.9	18.6
140	4	0.95	1.1	1.1	1.3	1.6	3.1	4.7	6.3
	6	0.17	0.18	0.21	0.28	0.35	0.69	1.0	1.4
	3	4.4	5	5.7	5.7	5.7	10.6	15.9	21.2
160	4	1.2	1.4	1.4	1.4	1.8	3.6	5.4	7.2
	6	0.17	0.18	0.21	0.28	0.35	0.69	1.0	1.4
	3	5.3	6.3	7	7	7	11.9	17.9	23.9
180	4	1.5	1.8	1.8	1.8	2	4	6	8
	6	0.2	0.24	0.24	0.31	0.39	0.78	1.2	1.6
	3	6.5	7.7	8.8	8.8	8.8	13.3	19.9	26.5
200	4	1.8	2.2	2.2	2.2	2.2	4.5	6.7	8.9
	6	0.25	0.3	0.3	0.35	0.43	0.87	1.3	1.7
	4	2.6	3.2	3.5	3.5	3.5	5.6	8.4	11.2
250	6	0.36	0.43	0.45	0.45	0.51	1.1	1.6	2.2
	8	0.95	0.12	0.12	0.15	0.18	0.36	0.54	0.72
	4	3.7	4.3	5	5	5	6.7	10.1	13.4
300	6	0.5	0.6	0.65	0.65	0.65	1.3	2	2.6
	8	0.13	0.17	0.17	0.18	0.22	0.43	0.65	0.87
	6	0.82	1	1.1	1.2	1.2	1.7	2.6	3.5
400	8	0.23	0.27	0.29	0.29	0.29	0.58	0.87	1.2
	10	0.08	0.09	0.1	0.1	0.12	0.23	0.35	0.47
	6	1.2	1.5	1.6	1.8	1.8	2.2	3.2	4.3
500	8	0.33	0.39	0.44	0.47	0.47	0.72	1.1	1.5
	10	0.11	0.14	0.15	0.15	0.15	0.29	0.44	0.58
	6	1.8	2.2	2.3	2.4	2.6	2.7	3.9	5.2
600	8	0.47	0.57	0.62	0.67	0.67	0.87	1.3	1.7
	10	0.16	0.18	0.2	0.22	0.22	0.35	0.52	0.07
	6	2.3	2.7	3	3.2	3.5	3.6	4.6	6.1
700	8	0.6	0.74	0.82	0.89	0.93	1	1.5	2
	10	0.2	0.25	0.27	0.3	0.3	0.41	0.61	0.82
	6	2.8	3.5	3.7	4	4.2	4.8	5.2	6.9
800	8	0.78	0.94	1	1.1	1.2	1.2	1.7	2.3
	10	0.26	0.3	0.34	0.38	0.4	0.47	0.7	0.92
	6	3.5	4.3	4.6	5.0	5.2	6	6	7.8
900	8	0.95	1.1	1.3	1.4	1.5	1.5	2	2.6
	10	0.32	0.37	0.43	0.46	0.5	0.52	0.79	1.1
	8	1.1	1.4	1.5	1.6	1.8	1.9	2.2	2.9
1000	10	0.38	0.45	0.5	0.55	0.6	0.6	0.87	1.2
	12	0.17	0.2	0.22	0.24	0.25	0.29	0.43	0.58

← TURBULENT FLOW → ← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU									
2500	3000	4000	5000	6000	7000	8000	9000	10,000	15,000
19.9	23.9	31.8	39.8	47.7	56	61	72	80	119
6.7	8	10.7	13.4	16.1	18.8	21.4	24.1	26.8	40.2
1.3	1.6	2.1	2.6	3.1	3.6	1.2	4.7	5.2	7.8
23.2	27.8	37.1	46.4	56	65	74	84	93	139
7.8	9.4	12.5	15.6	18.8	21.9	25	28.2	31.3	46.9
1.5	1.8	2.4	3.0	3.6	4.2	4.9	5.5	6.0	9.1
26.5	31.8	42.4	53	64	74	85	95	106	159
8.9	10.7	14.3	17.9	21.5	25	28.6	32.2	35.7	54
1.7	2.1	2.8	3.5	4.2	4.9	5.5	6.2	6.9	10.4
29.8	35.8	47.7	60	72	84	95	107	119	179
10.1	12.1	16.1	20.1	24.1	28.1	32.2	36.2	40.2	60
2	2.3	3.1	3.9	4.7	5.5	6.2	7	7.8	11.7
33.1	39.8	53	66	80	93	106	119	133	199
11.2	13.4	17.9	22.3	26.8	31.3	35.7	40.2	44.7	67
2.2	2.6	3.5	4.3	5.2	6.1	6.9	7.8	8.7	13
14	16.8	22.3	27.9	33.5	39.1	44.7	50	56	84
2.7	3.3	4.3	5.4	6.5	7.6	8.7	9.8	10.8	16.3
0.9	1.1	1.5	1.8	2.2	2.5	2.9	3.3	3.6	5.4
16.8	20.1	26.8	33.5	40.2	47	54	60	67	101
3.3	3.9	5.2	6.5	7.8	9.1	10.4	11.7	13	19.5
1.0	1.3	1.7	2.2	2.6	3	3.5	3.9	4.3	6.5
4.3	5.2	6.9	8.7	10.4	12.1	13.9	15.6	17.3	26
1.5	1.7	2.3	2.9	3.5	4.1	4.6	5.2	5.8	8.7
0.58	0.7	0.93	1.2	1.4	1.6	1.9	2.1	2.3	3.5
5.4	6.5	8.7	10.8	13	15.2	17.3	19.5	21.7	32.5
1.8	2.2	2.9	3.6	4.3	5.1	5.8	6.5	7.2	10.8
0.73	0.87	1.2	1.5	1.8	2	2.3	2.6	2.9	4.4
6.5	7.8	10.4	13	16	18.2	20.8	23.4	26	39
2.2	2.6	3.5	4.3	5.2	6.1	6.9	7.8	8.7	13
0.87	1.1	1.4	1.8	2.1	2.4	2.8	3.3	3.5	5.2
7.6	9.1	12.1	15.2	18.4	21.2	24.3	27.3	30.3	45.5
2.5	3	4.1	5.1	6.1	7.1	8.1	9.1	10.1	15.2
1	1.2	1.6	2	2.4	2.9	3.3	3.7	4.1	6.1
8.7	10.4	13.9	17.3	20.8	24.3	27.7	31.2	34.7	52
2.9	3.5	4.6	5.8	6.9	8.1	9.3	10.4	11.6	17.3
1.2	1.4	1.9	2.3	2.8	3.3	3.7	4.2	4.7	7
9.8	11.7	15.6	19.5	23.4	27.3	31.2	35.1	39	58.5
3.3	3.9	5.2	6.5	7.8	9.1	10.4	11.7	13	19.5
1.3	1.6	2.1	2.6	3.1	3.7	4.2	4.7	5.2	7.9
3.6	4.3	5.8	7.2	8.7	10.1	11.6	13	14.5	21.7
1.5	1.8	2.3	2.9	3.5	4.1	4.7	5.2	5.8	8.7
0.72	0.87	1.2	1.5	1.7	2	2.3	2.6	2.9	4.3
← LAMINAR FLOW →									

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU 20,000	25,000	30,000	40,000	50,000	60,000
	2	19.3	24.1	28.9	38.5	48.2	58
3	2½	9.5	11.8	14.2	19	23.7	28.4
	3	4	5	6	8	9.9	11.9
	2	32	40	48.2	64	80	96
5	2½	15.8	19.7	23.7	31.6	39.5	47.4
	3	6.6	8.3	9.9	13.3	16.6	9.9
	2	45	56	67	90	112	135
7	2½	22.1	27.6	33.2	44.2	55	66
	3	9.3	10.6	13.9	18.6	23.2	27.8
	2½	31.6	39.5	47.4	63	79	95
10	3	13.3	16.6	19.9	26.5	33.1	39.8
	4	4.5	5.6	6.7	8.9	11.2	13.4
	2½	47.4	59	71	95	118	142
15	3	19.9	24.9	29.8	39.8	49.7	60
	4	6.7	8.4	10.1	13.4	16.8	20.1
	3	26.5	33.1	39.8	53	66	80
20	4	8.9	11.2	13.4	17.9	22.3	26.8
	6	1.7	2.2	2.6	3.5	4.3	5.2
	3	33.1	41.4	49.7	66	83	99
25	4	11.2	14	16.8	22.3	27.9	33.5
	6	2.2	2.7	3.3	4.3	5.4	6.5
	3	39.8	49.7	60	80	99	119
30	4	13.4	16.8	20.1	26.8	33.5	40.2
	6	2.6	3.3	3.9	5.2	6.5	7.8
	3	53	66	80	106	133	160
40	4	17.9	22.3	26.8	35.7	44.7	54
	6	3.5	4.3	5.2	6.9	8.7	10.4
	4	22.3	27.9	33.5	44.7	56	67
50	6	4.3	5.4	6.5	8.7	10.8	13
	8	1.5	1.8	2.7	2.9	3.6	4.3
	4	26.8	33.5	40.2	54	67	80
60	6	5.2	6.5	7.8	10.4	13	16
	8	1.7	2.2	2.6	3.5	4.3	5.2
	4	31.3	39.1	46.9	63	78	94
70	6	6.1	7.6	9.1	12.1	15.2	18.4
	8	2	2.5	3	4.1	5.1	6.1
	6	6.9	8.7	10.4	13.9	17.3	20.8
80	8	2.3	2.9	3.5	4.6	5.8	6.9
	10	0.93	1.2	1.4	1.9	2.3	2.8
	6	7.8	9.8	11.7	15.6	19.5	23.4
90	8	2.6	3.3	3.9	5.2	6.5	7.8
	10	1.1	1.3	1.6	2.1	2.6	3.1

← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU								
70,000	80,000	90,000	100,000	125,000	150,000	175,000	200,000	500,000
67	77	87	96	120	145	169	193	482
332	37.9	42.6	47.4	59	71	83	95	237
13.9	15.9	17.9	19.9	24.9	29.8	34.8	39.8	99
112	129	145	161	200	241	281	321	803
55	63	71	79	99	118	138	158	395
23.2	26.5	29.8	33	41.4	49.7	58	66	166
157	180	202	225	281	337	393	450	. . .
77	88	100	111	138	166	194	221	553
32.5	37.1	40.7	46.4	58	70	81	93	232
111	126	142	158	197	237	276	316	790
46.4	53	60	66	83	99	116	133	331
15.6	17.9	20.1	22.3	27.9	33.5	39.1	44.7	112
166	190	213	237	296	355	415	474	. . .
70	80	89	99	124	149	174	199	497
23.5	26.8	30.2	33.5	41.9	50	59	67	168
93	106	119	133	166	199	232	265	663
31.3	35.7	40.2	44.7	56	67	78	89	223
6.1	6.9	7.8	8.7	10.8	13	15.2	17.3	43.3
116	133	149	166	207	49	290	331	828
39.1	44.7	50	56	70	84	98	112	279
7.6	8.7	9.8	10.8	13.5	16.3	19	21.7	54
139	159	179	199	249	298	348	398	. . .
46.9	54	60	67	84	101	117	134	335
9.1	10.4	11.7	13	16.3	19.5	22.7	26	65
186	212	239	265	331	398	464	532	. . .
63	72	80	89	112	134	156	179	447
12.1	13.9	15.6	17.3	21.7	26	30.3	34.7	87
78	89	101	112	140	168	196	223	559
15.2	17.3	19.5	21.7	27.1	32.5	37.9	43.3	107
5.1	5.8	6.5	7.2	9	10.8	12.6	14.5	36.1
94	107	121	134	168	201	235	268	670
18.2	20.8	23.4	26	32.5	39	45.5	52	130
6.1	6.9	7.8	8.7	10.8	13	15.2	17.3	43.4
110	125	141	156	196	235	274	313	782
21.2	24.3	27.3	30.3	37.9	45.5	53	61	152
7.1	8.1	9.1	10.1	12.6	15.2	17.7	20.2	51
24.3	27.7	31.2	34.7	43.3	52	61	69	173
8.1	9.3	10.4	11.6	14.5	17.3	20.2	23.1	58
3.3	3.7	4.2	4.7	5.8	7	8.2	9.3	23.3
27.3	31.2	35.1	39	48.7	59	68	78	195
9.1	10.4	11.7	13	16.3	19.5	22.8	26	65
3.7	4.2	4.7	5.2	6.6	7.9	9.2	10.5	26.2

← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU					
		20,000	25,000	30,000	40,000	50,000	60,000
100	6	8.7	10.8	13	17.3	21.7	26
	8	2.9	3.6	4.3	5.8	7.2	8.7
	10	1.2	1.5	1.8	2.3	2.9	3.5
120	6	10.4	13	15.6	20.8	26	31.2
	8	3.5	4.3	5.2	6.9	8.7	10.4
	10	1.4	1.8	2.1	2.8	3.5	4.2
140	6	12.1	15.2	18.2	24.3	30.3	36.4
	8	4	5.1	6.1	8.1	10.1	12.1
	10	1.7	2	2.4	3.3	4.1	4.9
160	6	13.9	17.3	20.8	27.7	34.7	41.6
	8	4.6	5.8	6.9	9.3	11.6	13.8
	10	1.9	2.3	2.8	3.7	4.7	5.6
180	6	15.6	19.5	23.4	31.2	39	46.8
	8	5.2	6.5	7.8	10.4	13	15.6
	10	2.1	2.6	3.1	4.2	5.2	6.3
200	8	5.8	7.2	8.7	11.6	14.5	17.3
	10	2.3	2.9	3.5	4.7	5.8	7
	12	1.2	1.5	1.7	2.3	2.9	3.5
250	8	7.2	9	10.8	14.5	18.1	21.7
	10	2.9	3.6	4.4	5.8	7.3	8.7
	12	1.5	1.8	2.2	2.9	3.6	4.3
300	8	8.7	10.8	13	17.3	21.7	26
	10	3.5	4.4	5.2	7	8.7	10.5
	12	1.7	2.2	2.6	3.5	4.3	5.2
400	8	11.6	14.5	17.3	23	28.9	34.7
	10	4.7	5.8	7	9.3	11.6	14
	12	2.3	2.9	3.5	4.6	5.8	7
500	8	14.5	18.1	21.7	28.9	36.1	43.4
	10	5.8	7.3	8.7	11.6	14.6	17.5
	12	2.9	3.6	4.3	5.8	7.2	8.7
600	8	17.3	21.7	26	34.7	43.4	52
	10	7	8.7	10.5	14	17.5	21
	12	3.5	4.3	5.2	7	8.7	10.4
700	8	20.2	25.3	30.3	40.5	51	61
	10	8.2	10.2	12.2	16.3	20.4	24.4
	12	4.1	5.1	6.1	8.1	10.1	12.2
800	8	23.1	28.9	34.7	46.2	58	69
	10	9.3	11.6	14	18.6	23.3	27.9
	12	4.6	5.8	7	9.3	11.6	13.9
900	8	26	32.5	39	52	65	78
	10	10.5	13.1	15.7	21	26.2	31.4
	12	5.2	6.5	7.8	10.4	13	15.6
1000	8	28.9	36.1	43.4	58	72	87
	10	11.6	14.6	17.5	23.3	29.1	34.9
	12	5.8	7.2	8.7	11.6	14.5	17.4

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU								
70,000	80,000	90,000	100,000	125,000	150,000	175,000	200,000	500,000
30.3	34.7	39	43.3	54	65	76	87	217
10.1	11.6	13	14.5	18.1	21.7	25.3	28.9	72
4.2	4.7	5.2	5.8	7.3	8.7	10.2	11.6	29.1
36.4	41.6	46.8	52	65	78	91	104	260
12.1	13.9	15.6	17.3	21.7	26	30.4	34.7	87
4.9	5.6	6.3	7	8.7	10.5	12.2	14	34.9
42.5	48.5	55	61	76	91	106	121	303
14.2	16.2	18.2	20.2	25.3	30.4	35.4	40.5	101
5.7	6.5	7.3	8.1	10.2	12.2	14.3	16.3	40.7
48.5	56	62	69	87	104	121	139	347
16.2	18.5	20.8	23.1	28.9	34.7	40.5	46.2	116
6.5	7.5	8.4	9.3	11.6	14	16.3	18.6	46.6
55	62	70	78	98	117	137	156	390
18.2	20.8	23.4	26	32.5	39	45.5	52	130
7.3	8.4	9.4	10.5	13.1	15.7	18.3	21	52
20.2	23.1	26	28.9	36.1	43.4	51	58	145
8.2	9.3	10.5	11.6	14.6	17.5	20.4	23.3	58
4.1	4.6	5.2	5.8	7.2	8.7	10.1	11.6	28.9
25.3	28.9	32.5	36.1	45.2	54	63	72	181
10.2	11.6	13.1	14.6	18.2	21.8	25.5	29.1	73
5.1	5.8	6.5	7.2	9	10.9	12.7	14.5	36.2
30.4	34.7	39	43.4	54	65	76	87	217
12.2	14	15.7	17.5	21.8	26.2	30.6	34.9	87
6.1	7	7.8	8.7	10.9	13	15.2	17.4	43.4
40.5	46.2	52	58	72	87	101	116	289
16.3	18.6	21	23.3	29.6	34.9	40.7	46.6	116
8.1	9.3	10.4	11.6	14.5	17.4	20.3	23.2	58
51	58	65	72	90	108	126	145	361
20.4	23.3	26.2	29.1	36.4	43.7	51	58	146
10.1	11.6	13	14.5	18.1	21.7	25.3	28.9	72
61	69	78	87	107	130	152	173	434
24.4	27.9	31.4	34.9	43.7	52	61	70	175
12.2	13.9	15.6	17.4	21.7	26.1	30.4	34.7	87
71	81	91	101	126	152	177	202	506
28.5	32.6	36.7	40.7	51	61	71	82	204
14.2	16.2	18.2	20.3	25.3	30.4	35.5	40.5	101
81	93	104	116	145	173	202	231	578
32.6	37.3	41.9	46.6	58	70	82	93	233
16.2	18.5	20.8	23.1	28.9	34.7	40.5	46.3	116
91	104	117	130	163	195	228	260	650
36.7	41.9	47.1	52	66	79	92	105	262
18.2	20.8	23.4	26.1	32.6	39.1	45.6	52	130
101	116	130	145	181	217	253	289	723
40.7	46.6	52	58	72	87	102	116	291
20.3	23.2	26.1	28.9	36.2	43.4	51	58	145

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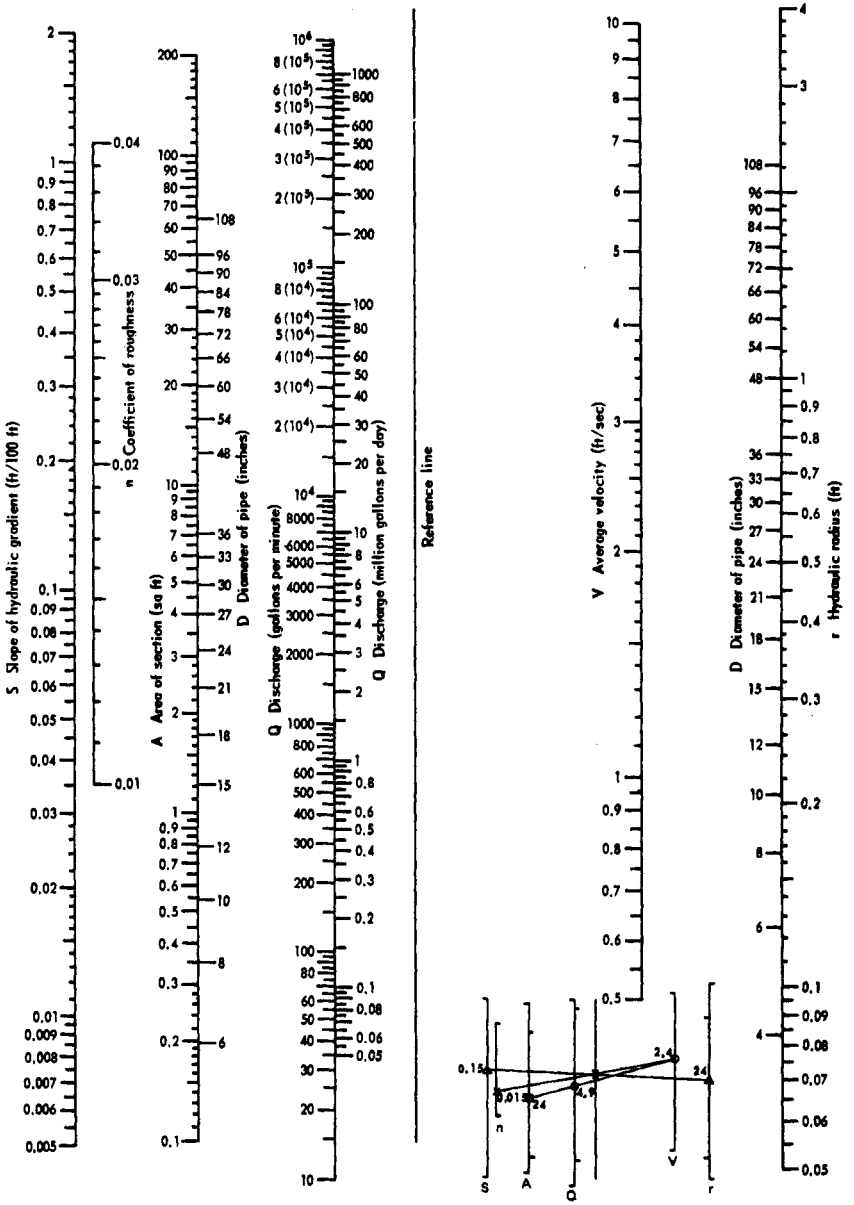


FIGURE 35 Nomogram for the solution of the Manning formula. Obtain values of n from Table 4. To solve for S, align A with Q and read V, align V and n intersecting reference line, align r with reference line intersection, and read S (1 m = 3.28 ft; 1 m = 39.37 in; 1 m^2 = 1550 in^2; 1 m^3/h = 4.4 gpm; 1 m/s = 3.28 ft/s; 1 m/m = 1 ft/ft).

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TABLE 4 Values of friction factor n to be used with the Manning Formula in Figure 35

Surface	Rest	Good	Fair	Bad
Uncoated cast iron pipe	0.012	0.013	0.014	0.015
Coated cast iron pipe	0.011	0.012[a]	0.013[a]	
Commercial wrought iron pipe, black	0.012	0.013	0.014	0.015
Commercial wrought iron pipe, galvanized	0.013	0.014	0.015	0.017
Smooth brass and glass pipe	0.009	0.010	0.011	0.013
Smooth lockbar and welded OD pipe	0.010	0.011[a]	0.013[a]	
Riveted and spiral steel pipe	0.013	0.015[a]	0.017[a]	
Vitrified sewer pipe	0.010	0.013[a]	0.015	0.017
	0.011			
Common clay drainage tile	0.011	0.012[a]	0.014[a]	0.017
Glazed brickwork	0.011	0.012	0.013[a]	0.015
Brick in cement mortar; brick sewers	0.012	0.013	0.015[a]	0.017
Neat cement surfaces	0.010	0.011	0.012	0.013
Cement mortar surfaces	0.011	0.012	0.013[a]	0.015
Concrete pipe	0.012	0.013	0.015[a]	0.016
Wood-stave pipe	0.010	0.011	0.012	0.013
Plank flumes:				
Planed	0.010	0.012[a]	0.013	0.014
Unplaned	0.011	0.013[a]	0.014	0.015
With battens	0.012	0.015[a]	0.016	
Concrete-lined channels	0.012	0.014[a]	0.016[a]	0.018
Cement-rubble surface	0.017	0.020	0.025	0.030
Dry-rubble surface	0.025	0.030	0.033	0.035
Dressed-ashlar surface	0.013	0.014	0.015	0.017
Semicircular metal flumes, smooth	0.011	0.012	0.013	0.015
Semicircular metal flumes, corrugated	0.0225	0.025	0.0275	0.0.30
Canals and ditches:				
Earth, straight, and uniform	0.017	0.020	0.0225[a]	0.025
Rock cuts, smooth, and uniform	0.025	0.030	0.033[a]	0.035
Rock cuts, jagged, and irregular	0.035	0.040	0.045	
Winding sluggish canals	0.0225	0.025[a]	0.0275	0.030
Dredged earth channels	0.025	0.0275[a]	0.030	0.033
Canals with rough stony beds, weeds on earth banks	0.025	0.030	0.035[a]	0.040
Earth bottom, rubble sides	0.028	0.030[a]	0.033[a]	0.035
Natural stream channels:				
(1) Clean, straight bank, full stage, no rifts, or deep pools	0.025	0.0275	0.030	0.033
(2) Same as (1), but some weeds and stones	0.030	0.033	0.035	0.040
(3) Winding, some pools and shoals, clean	0.033	0.035	0.040	0.045
(4) Same as (3), lower stages, more ineffective slope and sections	0.040	0.045	0.050	0.055
(5) Same as (3), some weeds and stones	0.035	0.040	0.045	0.050
(6) Same as (4), stony sections	0.045	0.050	0.055	0.060
(7) Sluggish river reaches, rather weedy or with very deep pools	0.050	0.060	0.070	0.080
(8) Very weedy reaches	0.075	0.100	0.125	0.150

[a]Values commonly used in designing.

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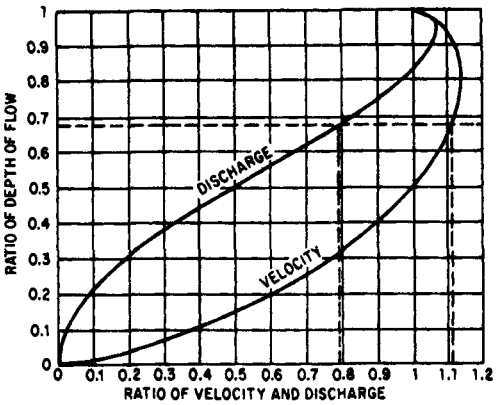


FIGURE 36 Discharge velocity of a partially full circular pipe versus that of a full pipe

A rearrangement of Eq. 18 gives

in USCS units
$$S = h_f = \left(\frac{Vn}{1.486r^{2/3}}\right)^2 \tag{19a}$$

in SI units
$$S = h_f = \left(\frac{Vn}{r^{2/3}}\right)^2 \tag{19b}$$

The following examples illustrate the application of the Manning formula (Eq. 19), Figures 35 and 36, and Table 4 to the solution of problems involving the flow of water in open channels.

EXAMPLE 10 The flow through a 24-in (61.0-cm) ID commercial wrought iron pipe in fair condition is 4.9 mgd (772.7 m^3/h). Determine the loss of head as a result of friction in feet per 100 ft (meters per 100 m) of pipe and the slope required to maintain a full, uniformly flowing pipe.

In USCS units
$$V = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{4.9 \times 10^6 \times 0.408}{24 \times 60 \times 24^2} = 2.41 \text{ ft/s}$$

In SI units
$$V = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{772.7}{60.96^2} \times 3.54 = 0.736 \text{ m/s}$$

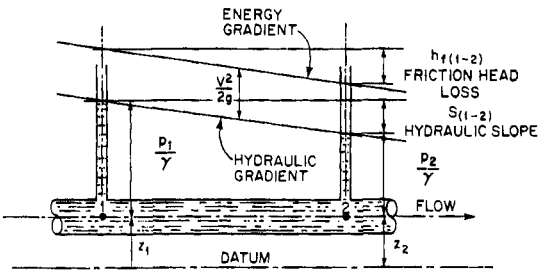


FIGURE 37 Slopes of energy and hydraulic gradients measure frictional head loss ft/ft (m/m) of pipe length

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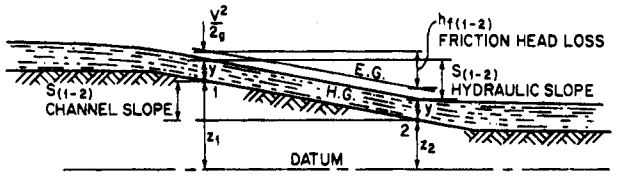


FIGURE 38 In an open channel with uniform flow, slopes of channel bottom, energy, and hydraulic gradients are the same as frictional head loss ft/ft (m/m) of channel length

From Table 4, $n = 0.015$.

In USCS units

$$r = D/4 = 2/4 = 0.5 \text{ ft} \qquad \text{(hydraulic radius)}$$

$$S = h_f = \left(\frac{Vn}{1.486r^{2/3}}\right)^2 = \left(\frac{2.41 \times 0.015}{1.486 \times 0.5^{2/3}}\right)^2 \qquad \text{(from Eq. 19a)}$$

$$= 0.0015 \text{ ft/ft}$$

$$100 \times 0.0015 = 0.15 \text{ ft}/100 \text{ ft} \qquad \text{(slope and frictional head)}$$

In SI units

$$r = D/4 = 0.6096/4 = 0.1524 \text{ m} \qquad \text{(hydraulic radius)}$$

$$S = h_f = \left(\frac{Vn}{r^{2/3}}\right)^2 = \left(\frac{0.736 \times 0.015}{0.1524^{2/3}}\right)^2 \qquad \text{(from Eq. 19b)}$$

$$= 0.0015 \text{ m/m}$$

$$100 \times 0.0015 = 0.15 \text{ m}/100 \text{ m} \qquad \text{(slope and frictional head)}$$

The problem may also be solved by using Figure 35 and following the trace lines:

$$S = h_f = 0.15 \text{ ft}/100 \text{ ft (m}/100 \text{ m)}$$

EXAMPLE 11 Determine what the flow and velocity would be if the pipe in Example 10 were flowing two-thirds full and were laid on the same slope.

Follow the trace lines in Figure 36 and note that the multipliers for discharge and velocity are 0.79 and 1.11, respectively. Therefore

in USCS units

$$\text{Flow} = 0.79 \times 4.9 = 3.87 \text{ mgd}$$

$$\text{Velocity} = 1.11 \times 2.41 = 2.68 \text{ ft/s}$$

in SI units

$$\text{Flow} = 0.79 \times 772.7 = 610 \text{ m}^3/\text{h}$$

$$\text{Velocity} = 1.11 \times 0.736 = 0.817 \text{ m/s}$$

Pipe Fittings Invariably, a system containing piping will have connections that change the size or direction of the conduit. These fittings add frictional losses, called *minor losses,* to the system head. Fitting losses are generally the result of changes in velocity or direction. A decreasing velocity results in more loss in head than an increasing velocity, as the former causes energy-dissipating eddies. Experimental results have

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indicated that minor losses vary approximately as the square of the velocity through the fittings.

VALVES AND STANDARD FITTINGS The resistance to flow through valves and fittings may be found in References 4, 5, 17, and other sources. Losses are usually expressed in terms of a *resistance coefficient* K and the average velocity head in a pipe having the same diameter as the valve or fitting. The frictional resistance h in feet (meters) is found from the equation

$$h = K\frac{V^2}{2g} \tag{20}$$

where K = resistance coefficient, which depends on design and size of valve or fitting
V = average velocity in pipe of corresponding internal diameter, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

A comparison of the Darcy-Weisbach equation (Eq. 16) with Eq. 20 suggests that K equals $f(L/D)$, where L is the equivalent length of pipe in feet (meters) and D is the inside pipe diameter in feet (meters), to produce the same head loss in a straight pipe as through a valve or fitting. The friction in an "equivalent length of pipe" has been another method used to estimate head loss through valves and fittings. Values of the ratio L/D have been experimentally determined. This ratio multiplied by the inside diameter of a pipe of specified schedule for the valve or fitting being considered gives the equivalent length of pipe to use to calculate the head lost.

Loss of head in straight pipe depends on the friction factor or Reynolds number. However, with valves and fittings, head is lost primarily because of change in direction of flow, change in cross section, and obstructions in the flow path. For this reason, the resistance coefficient is practically constant for a particular shape of valve or fitting for all flow conditions, including laminar flow. The resistance coefficient would theoretically be constant for all sizes of a particular design of valve or fitting except that all sizes are not geometrically similar. The Crane Company has reported the results of tests that show that the resistance coefficient for a number of lines of valves and fittings decreases with increasing size at flow conditions of equal friction factor and that the equivalent length L/D tends to be constant for the various sizes at the same flow conditions.

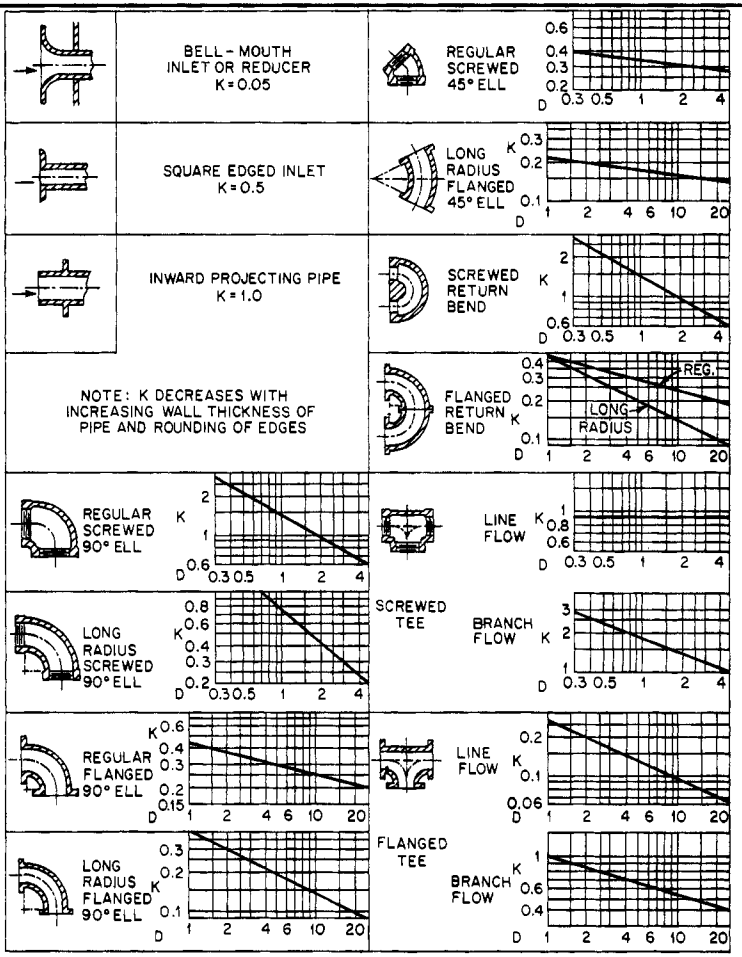
When available, the K factor furnished by the valve or fitting manufacturer should always be used rather than the value from a general listing. The Hydraulic Institute lists losses in terms of K through valves and fittings (Tables 5A to 5C); these losses vary with size of the valve or fitting but are independent of friction factor. The Crane Company provides a similar listing of K values (Tables 6A to 6E). The latter listing of flow coefficients is associated with the velocity head $V^2/2g$ that would occur through the internal diameter of the schedule pipes for the various ANSI classes of valves and fittings shown in Table 6e. If the connecting pipe is of a different size or schedule, either use the velocity for the pipe shown in Table 6a or use the actual pipe velocity head and correct the resistance coefficient obtained from this table by the multiplier

$$\left(\frac{\text{Actual pipe } ID}{\text{Standard pipe } ID}\right)^4$$

Tables 6 are based on the use of an equivalent length constant for complete turbulent flow for each valve or fitting shown. This constant is shown as the multiplier of the friction factor f_T for the corresponding clean commercial steel pipe with completely turbulent flow. The product $(L/D)(f_T)$ is the coefficient K. The friction factors are given in Table 6a for different pipe sizes, or they can be obtained from Figure 31 or 32. If the valve or fitting has a sudden or gradual contraction, enlargement, or change in direction of flow, appropriate formulas for these conditions are given for the determination of K. If flow is laminar, valve and fitting resistance coefficients are obtained from Table 6a based on completely turbulent

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TABLE 5A Resistance coefficients for K for valves and fittings



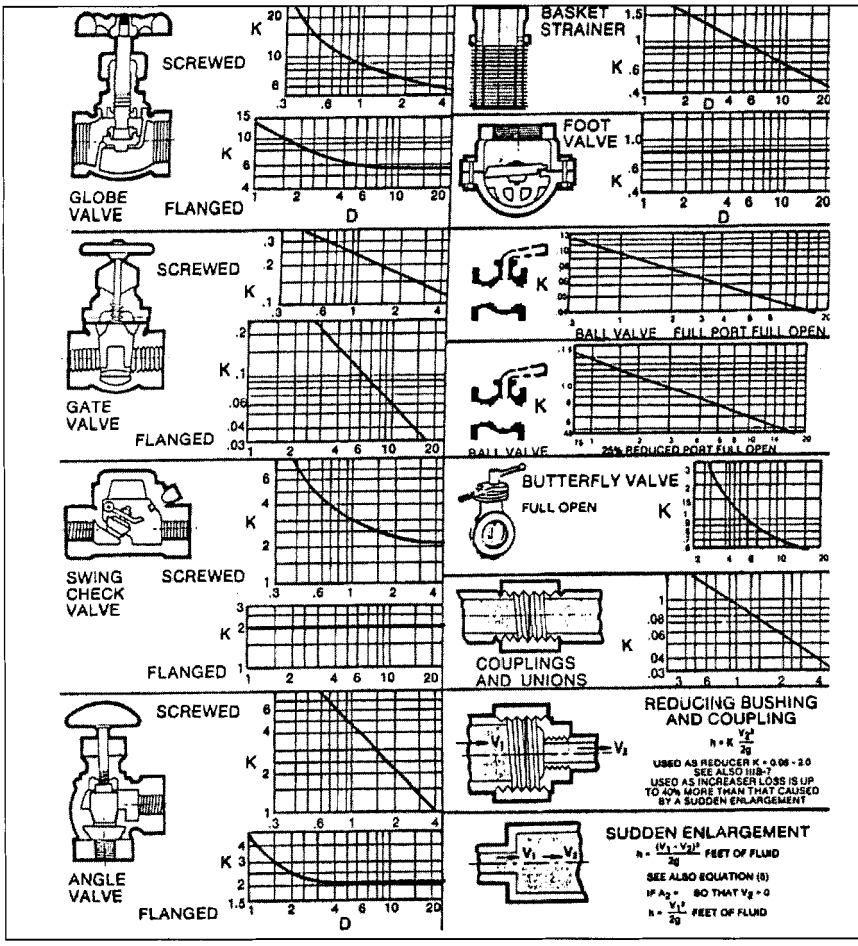
NOTE: D = nominal iron pipe size in inches (in × 25.4 = mm). (Hydraulic Institute Engineering Data Book, Reference 5)

flow, but the pipe frictional loss is calculated using the laminar friction factor $f = 64/Re$ instead of f_T. Also shown in Tables 6 for check valves is the minimum pipe velocity required for full disk lift for the coefficient of resistance listed ($\bar{V}$ = liquid specific volume in cubic feet per pound).

Prior to the 15th printing (1976) of the Crane Company Technical Paper 410, and as shown in the first edition of this text, valve and fitting losses were calculated using an equivalent length of pipe rather than the coefficient K. The Crane Company states that this conceptual change regarding the values of equivalent length L/D and resistance coefficient K for valves and fittings relative to the friction factor in pipes has a relatively minor

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TABLE 5B Resistance coefficients K for valves and fittings



NOTE: D = nominal iron pipe size in inches (in × 25.4 = mm). For velocities below 15 ft/s (4.6 m/s), check valves and foot valves will be only partially open and will exhibit higher values of K than shown.

$$h = K\frac{V^2}{2g}, \text{ ft (m) of fluid}$$

(Hydraulic Institute Engineering Data Book, Reference 5)

effect on most problems dealing with turbulent flow and avoids a significant overstatement of pressure drop in the laminar zone.

Valve Flow Coefficient The loss of head through valves, particularly control valves, is often expressed in terms of a *flow coefficient* C_v in USCS units (K_v in SI units). The flow of water in gallons per minute (cubic meters per hour) at 60°F (15.6°C) that will pass through a valve with a 1-lb/in^2 (1-bar) pressure drop is defined as the flow coefficient for a particular valve opening. Because loss of head h is a measure of energy loss per unit weight (force) and because $h = p/\gamma$ and head loss varies directly with the square of the flow through a certain fixed opening, the formulas for flow coefficient are

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TABLE 5C Approximate variation for K listed in Tables 5a and 5b

	Fitting	Range of variation, %
90° elbow	Regular screwed	±20 above 2-in size[a]
	Regular screwed	±40 below 2-in size
	Long radius, screwed	±25
	Regular flanged	±35
	Long radius, flanged	±30
45° elbow	Regular screwed	±10
	Long radius, flanged	±10
180° bend	Regular screwed	±25
	Regular flanged	±35
	Long radius, flanged	±30
T	Screwed, line or branch flow	±25
	Flanged, line or branch flow	±35
Globe valve	Screwed	±25
	Flanged	±25
Gate valve	Screwed	±25
	Flanged	±50
Check valve[b]	Screwed	±30
	Flanged	+200 / −80
Sleeve check valve	—	Multiply flanged values by 0.2 to 0.5
Tilting check valve	—	Multiply flanged values by 0.13 to 0.19
Drainage gate check	—	Multiply flanged values by 0.03 to 0.07
Angle valve	Screwed	±20
	Flanged	±50
Basket strainer	—	±50
Foot valve[a]	—	±50
Couplings	—	±50
Unions	—	±50
Reducers	—	±50

[a]In × 25.4 = mm.

[b]For velocities below 15 ft/s (4.6 m/s), check valves and foot valves will be only partially open and will exhibit higher values of K than shown.

(Hydraulic Institute Engineering Data Book, Reference 5)

In USCS units
$$C_v = \text{gpm}\sqrt{\frac{\text{sp. gr.}}{\text{lb/in}^2}} \tag{21a}$$

also
$$C_v = 29.9d^2/\sqrt{K} \tag{22a}$$

In SI units
$$K_v = \text{m}^3/\text{h}\sqrt{\frac{\text{sp. gr.}}{\text{bar}}} \tag{21b}$$

also
$$K_v = 0.04d^2/\sqrt{K} \tag{22b}$$

where d= internal diameter of pipe corresponding to K and as shown in Table 6E, in (mm) and 1 bar = 100 kPa. The conversion from the SI flow coefficient to the USCS flow coefficient is

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TABLE 6A Resistance coefficient K for valves and fittings

PIPE FRICTION DATA FOR CLEAN COMMERCIAL STEEL PIPE WITH FLOW IN ZONE OF COMPLETE TURBULENCE

Nominal Size	½"	¾"	1"	1¼"	1½"	2"	2½, 3"	4"	5"	6"	8-10"	12-16"	18-24"
Friction Factor (f_T)	.027	.025	.023	.022	.021	.019	.018	.017	.016	.015	.014	.013	.012

FORMULAS FOR CALCULATING "K" FACTORS FOR VALVES AND FITTINGS WITH REDUCED PORT

• **Formula 1**

$$K_2 = \frac{0.8 \sin\frac{\theta}{2}(1-\beta^2)}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 2**

$$K_2 = \frac{0.5\,(1-\beta^2)\sqrt{\sin\frac{\theta}{2}}}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 3**

$$K_2 = \frac{2.6 \sin\frac{\theta}{2}(1-\beta^2)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 4**

$$K_2 = \frac{(1-\beta^2)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 5**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_2 = \frac{K_1 + \sin\frac{\theta}{2}[0.8\,(1-\beta^2) + 2.6\,(1-\beta^2)^2]}{\beta^4}$$

• **Formula 6**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_2 = \frac{K_1 + 0.5\sqrt{\sin\frac{\theta}{2}}(1-\beta^2) + (1-\beta^2)^2}{\beta^4}$$

• **Formula 7**

$$K_2 = \frac{K_1}{\beta^4} + \beta\,(\text{Formula 2} + \text{Formula 4}) \text{ when } \theta = 180°$$

$$K_2 = \frac{K_1 + \beta\left[0.5\,(1-\beta^2) + (1-\beta^2)^2\right]}{\beta^4}$$

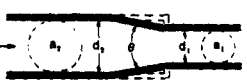
NOTES:

$$\beta = \frac{d_1}{d_2}$$

$$\beta^2 = \left(\frac{d_1}{d_2}\right)^2 = \frac{a_1}{a_2}$$

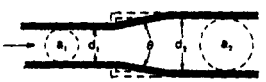
Subscript 1 defines dimensions and coefficients with reference to the smaller diameter.
Subscript 2 refers to the larger diameter.

SUDDEN AND GRADUAL CONTRACTION



If: $\theta \leq 45°$ . . . . . K_2 = Formula 1
$45° < \theta \leq 180°$ . . . K_2 = Formula 2

SUDDEN AND GRADUAL ENLARGEMENT



If: $\theta \leq 45°$ . . . . . K_2 = Formula 3
$45° < \theta \leq 180°$ . . . K_2 = Formula 4

NOTE: K is based on use of schedule pipe as listed in Table 6e. In × 25.4 = mm.
SOURCE: Reference 4.

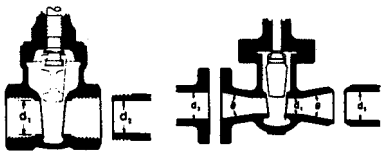
$$C_v = 1.156K_v$$

EXAMPLE 12 A pumping system consists of 20 ft (6.1 m) of 2-in (51-mm) suction pipe and 300 ft (91.5 m) of 1½-in (38-mm) discharge pipe, both Schedule 40 new steel. Also included are a bell mouth inlet, a 90° short radius (SR) suction elbow, a full port suction

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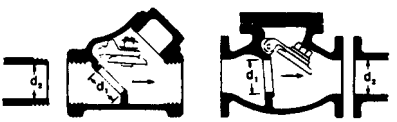
TABLE 6B Resistance coefficient K for valves and fittings

GATE VALVES
Wedge Disc, Double Disc, or Plug Type



If: $\beta = 1, \theta = 0 \ldots K_1 = 8 f_T$
$\beta < 1$ and $\theta \leq 45° \ldots K_2 =$ Formula 5
$\beta < 1$ and $45° < \theta \leq 180° \ldots K_2 =$ Formula 6

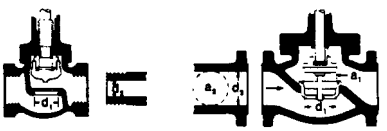
SWING CHECK VALVES



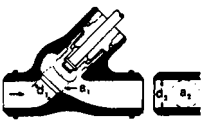
$K = 100 f_T$
Minimum pipe velocity (fps) for full disc lift $= 35 \sqrt{V}$

$K = 50 f_T$
Minimum pipe velocity (fps) for full disc lift $= 48 \sqrt{V}$

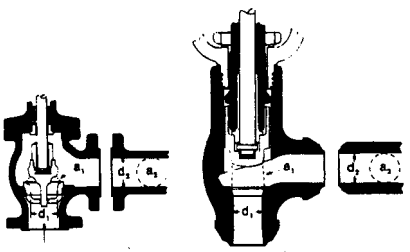
GLOBE AND ANGLE VALVES



If: $\beta = 1 \ldots K_1 = 340 f_T$



If: $\beta = 1 \ldots K_1 = 55 f_T$



If: $\beta = 1 \ldots K_1 = 150 f_T$

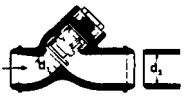
If: $\beta = 1 \ldots K_1 = 55 f_T$

All globe and angle valves, whether reduced seat or throttled,
If: $\beta < 1 \ldots K_2 =$ Formula 7

LIFT CHECK VALVES

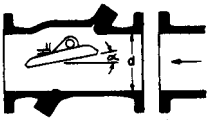


If: $\beta = 1 \ldots K_1 = 600 f_T$
$\beta < 1 \ldots K_2 =$ Formula 7
Minimum pipe velocity (fps) for full disc lift $= 40 \beta^2 \sqrt{V}$



If: $\beta = 1 \ldots K_1 = 55 f_T$
$\beta < 1 \ldots K_2 =$ Formula 7
Minimum pipe velocity (fps) for full disc lift $= 140 \beta^2 \sqrt{V}$

TILTING DISC CHECK VALVES



	$\alpha = 5°$	$\alpha = 15°$
Sizes 2 to 8" ... $K =$	$40 f_T$	$120 f_T$
Sizes 10 to 14" ... $K =$	$30 f_T$	$90 f_T$
Sizes 16 to 48" ... $K =$	$20 f_T$	$60 f_T$
Minimum pipe velocity (fps) for full disc lift =	$80 \sqrt{V}$	$30 \sqrt{V}$

gate valve of Class 150 steel, a full port discharge gate valve of Class 400 steel, and a swing check valve. The valves and fittings are screw-connected and the same size as the connecting pipe.

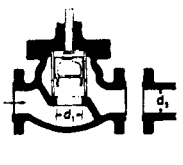
Determine the pipe, valve, and fitting losses when 60°F (15.6°C) oil having a specific gravity of 0.855 is pumped at a rate of 60 gpm (13.6 m³/h). Use resistance coefficients from Tables 5.

The inner diameter of the suction pipe is 2.067 in (52.5mm), and from Figure 32. $\epsilon/D = 0.00087$. From Eq. 9,

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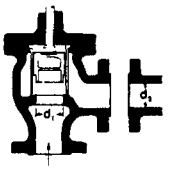
TABLE 6C Resistance coefficient K for valves and fittings

STOP-CHECK VALVES (Globe and Angle Types)



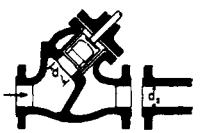
If:
$\beta = 1 \ldots K_1 = 400\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 55\, \beta^2 \sqrt{\overline{V}}$

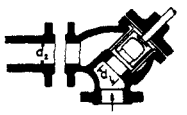


If:
$\beta = 1 \ldots K_1 = 200\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 75\, \beta^2 \sqrt{\overline{V}}$

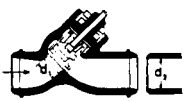


If:
$\beta = 1 \ldots K_1 = 300\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

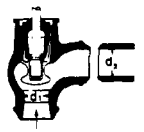


If:
$\beta = 1 \ldots K_1 = 350\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 60\, \beta^2 \sqrt{\overline{V}}$



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\, \beta^2 \sqrt{\overline{V}}$

FOOT VALVES WITH STRAINER

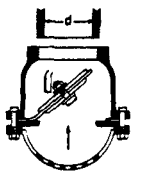
Poppet Disc



$K = 420 f_T$

Minimum pipe velocity (fps) for full disc lift
$= 15 \sqrt{\overline{V}}$

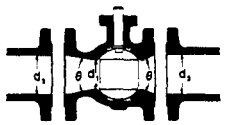
Hinged Disc



$K = 75 f_T$

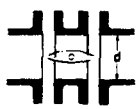
Minimum pipe velocity (fps) for full disc lift
$= 35 \sqrt{\overline{V}}$

BALL VALVES



If: $\beta = 1, \theta = 0 \ldots K_1 = 3\, f_T$
$\beta < 1$ and $\theta \leq 45° \ldots K_2 =$ Formula 5
$\beta < 1$ and $45° < \theta \leq 180° \ldots K_2 =$ Formula 6

BUTTERFLY VALVES



Sizes 2 to 8" $\ldots K = 45\, f_T$
Sizes 10 to 14" $\ldots K = 35\, f_T$
Sizes 16 to 24" $\ldots K = 25\, f_T$

In USCS units, $$V = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{60}{2.067^2} \times 0.408 = 5.73 \text{ ft}$$

$$VD'' = 5.73 \times 2.067 = 11.8 \text{ ft/s} \times \text{in}$$

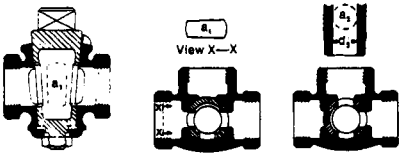
In SI units $$V = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{13.6}{5.25^2} \times 3.54 = 1.75 \text{ m/s}$$

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TABLE 6D Resistance coefficient K for valves and fittings

PLUG VALVES AND COCKS

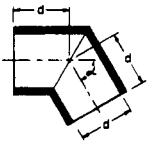
Straight-Way 3-Way



If: $\beta = 1$, $K_1 = 18\, f_T$ If: $\beta = 1$, $K_1 = 30\, f_T$ If: $\beta = 1$, $K_1 = 90\, f_T$

If: $\beta < 1 \ldots K_1$ = Formula 6

MITRE BENDS



α	K
0°	2 f_T
15°	4 f_T
30°	8 f_T
45°	15 f_T
60°	25 f_T
75°	40 f_T
90°	60 f_T

90° PIPE BENDS AND FLANGED OR BUTT-WELDING 90° ELBOWS



r/d	K	r/d	K
1	20 f_T	8	24 f_T
1.5	14 f_T	10	30 f_T
2	12 f_T	12	34 f_T
3	12 f_T	14	38 f_T
4	14 f_T	16	42 f_T
6	17 f_T	20	50 f_T

The resistance coefficient, K_B, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\,\pi\, f_T \frac{r}{d} + 0.5\,K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bend (per table)

CLOSE PATTERN RETURN BENDS



$K = 50\, f_T$

STANDARD ELBOWS

90° 45°



$K = 30\, f_T$ $K = 16\, f_T$

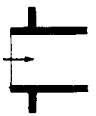
STANDARD TEES



Flow thru run. . . . . $K = 20\, f_T$
Flow thru branch. . . . $K = 60\, f_T$

PIPE ENTRANCE

Inward Projecting

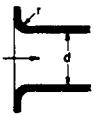


$K = 0.78$

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

*Sharp-edged

Flush



For K, see table

PIPE EXIT

Projecting Sharp-Edged Rounded



$K = 1.0$ $K = 1.0$ $K = 1.0$

$$VD = 1.75 \times 0.0525 = 0.0919 \text{ m/s} \times \text{m}$$
$$(VD'' = 0.0919 \times 129.2 = 11.8 \text{ ft/s} \times \text{in})$$

From Figure 33, $Re = 1 \times 10^4$, and from Figure 31, $f = 0.031$. From Eq. 16

in USCS units $$h_{fs} = 0.031 \frac{20 \times 12}{2.067} \times \frac{5.73^2}{2 \times 32.17} = 1.84 \text{ ft}$$

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TABLE 6E Pipe schedule for different classes of valves and fittings associated with K factors used in Tables 6A to 6D

Class	Schedule
300 and lower	40
400 and 600	80
900	120
1500	160
2500 (sizes $\frac{1}{2}$ to 6 in)[a]	XXS
2500 (sizes 8 in and up)[a]	160

[a]In × 25.4 = mm.
Source: Reference 4.

In SI units $$h_{fs} = 0.031 \frac{6.1}{0.0525} \times \frac{1.75^2}{2 \times 9.807} = 0.56 \text{ m}$$

The inner diameter of the discharge pipe is 1.610 in (40.89 mm), and from Figure 32, $\varepsilon/D = 0.0011$. From Eq. 9

in USCS units $$V = \frac{60}{1.601^2} \times 0.408 = 9.44 \text{ ft/s}$$
$$VD'' = 9.44 \times 1.601 = 15.2 \text{ ft/s} \times \text{in}$$

in SI units $$V = \frac{13.6}{4.089^2} \times 3.54 = 2.88 \text{ m/s}$$
$$VD = 2.88 \times 0.04089 = 0.118 \text{ m/s} \times \text{m}$$
$$(VD'' = 0.118 \times 129.2 = 15.2 \text{ ft/s} \times \text{in})$$

From Figure 33, $Re = 1.5 \times 10^4$, and from Figure 31, $f = 0.030$. From Eq. 16

in USCS units $$h_{fd} = 0.030 \frac{300 \times 12}{1.610} \times \frac{9.44^2}{2 \times 32.17} = 9.29 \text{ ft}$$

in SI units $$h_{fd} = 0.030 \frac{91.5}{0.04089} \times \frac{2.88^2}{2 \times 9.807} = 28.39 \text{ m}$$

The valve and fitting losses from Tables 5 and Eq. 20 are 2-in (51-mm) bellmouth, $K = 0.05$:

In USCS units $$h_{f1} = 0.05 \frac{5.73^2}{2 \times 32.17} = 0.026 \text{ ft}$$

In SI units $$h_{f1} = 0.05 \frac{1.75^2}{2 \times 9.807} = 0.0078 \text{ m}$$

2-in (51-mm) SR 90° elbow, $K = 0.95 \pm 30\%$

In USCS units $$h_{f2} = 0.95 \frac{5.73^2}{2 \times 32.17} = 0.48 \pm 0.14 \text{ ft}$$

In SI units $$h_{f2} = 0.95 \frac{1.75^2}{2 \times 9.807} = 0.15 \pm 0.044 \text{ m}$$

2-in (51-mm) gate valve, $K = 0.16 \pm 25\%$

In USCS units $$h_{f3} = 0.16 \frac{5.73^2}{2 \times 32.17} = 0.082 \pm 0.021 \text{ ft}$$

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In SI units $h_{f3} = 0.16 \dfrac{1.75^2}{2 \times 9.807} = 0.0250 \pm 0.0062$ m

$1\frac{1}{2}$-in (38-mm) gate valve, $K = 0.19 \pm 25\%$

In USCS units $h_{f4} = 0.19 \dfrac{9.44^2}{2 \times 32.17} = 2.63 \pm 0.066$ ft

In SI units $h_{f4} = 0.19 \dfrac{2.88^2}{2 \times 9.807} = 0.0803 \pm 0.02$ m

$1\frac{1}{2}$-in (38-mm) swing check valve, $K = 2.5 \pm 30\%$

In USCS units $h_{f5} = 2.5 \dfrac{9.44^2}{2 \times 32.17} = 3.46 \pm 1.0$ ft

In SI units $h_{f5} = 2.5 \dfrac{2.88^2}{2 \times 9.807} = 1.06 \pm 0.32$ m

The total pipe, valve, and fitting losses are

In USCS units
$$\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$$
$$= 1.84 + 92.9 + 0.026 + 0.48 + 0.082 + 0.263 + 3.46 = 99.05 \text{ ft}$$
$$\text{Total variation} = \pm(0.14 + 0.021 + 0.066 + 1.0) = \pm 1.23 \text{ ft}$$

In SI units
$$\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$$
$$= 0.56 + 28.39 + 0.0078 + 0.044 + 0.0250 + 0.0803 + 1.06 = 30.17 \text{ m}$$
$$\text{Total variation} = \pm(0.044 + 0.0062 + 0.02 + 0.32) = \pm 0.39 \text{ m}$$

EXAMPLE 13 Solve Example 12 using resistance coefficients from Tables 6.

Suction pipe:

In USCS units $h_{fs} = 1.84$ ft (same as in Example 12)

In SI units $h_{fs} = 0.56$ m (same as in Example 12)

Discharge pipe,

In USCS units $h_{fd} = 92.9$ ft (same as in Example 12)

In SI units $h_{fd} = 28.39$ m (same as in Example 12)

Valve and fitting losses from Tables 6 and Eq. 20: 2-in (51-mm) bellmouth, $K = 0.04$

In USCS units $h_{f1} = 0.04 \dfrac{5.73^2}{2 \times 32.17} = 0.020$ ft

In SI units $h_{f1} = 0.04 \dfrac{1.75^2}{2 \times 9.807} = 0.0062$ m

2-in (51-mm) SR 90° elbow, $K = 30 f_T$

$f_T = 0.019$ (from Table 6A)

$K = 30 \times 0.019 = 0.57$

In USCS units $h_{f2} = 0.57 \dfrac{5.73^2}{2 \times 32.17} = 0.29$ ft

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In SI units $h_{f2} = 0.57 \dfrac{1.57^2}{2 \times 9.807} = 0.089$ m

2-in (51-mm) gate valve, $\beta = 1, \theta = 0, k = 8f_T$ $\left(\beta = \dfrac{d_1}{d_2} = 1 \text{ from Table 6A}\right)$

$K = 8 \times 0.019 = 0.15$

In USCS units $h_{f3} = 0.15 \dfrac{5.73^2}{2 \times 32.17} = 0.077$ ft

In SI units $h_{f3} = 0.15 \dfrac{1.75^2}{2 \times 9.807} = 0.023$ m

1-in (38-mm) gate valve, $\beta = 1, \theta = 0, K = 8f_T$ for Schedule 80 (from Table 6E)

$K = 8f_T (1.10/1.500)^4 = 10.62f_T$ for Schedule 40

$f_T = 0.021$ (from Table 6A)

$K = 10.62 \times 0.021 = 0.22$

In USCS units $h_{f4} = 0.22 \dfrac{9.44^2}{2 \times 32.17} = 0.30$ ft

In SI units $h_{f4} = 0.22 \dfrac{2.88^2}{2 \times 9.807} = 0.093$ m

$1\frac{1}{2}$-in (38-mm) swing check valve, $K = 100f_T$ (from Table 6B)

Minimum pipe velocity for full disk lift $= 35\sqrt{V} = 35\sqrt{0.0189} = 4.81$ ft/s < 9.44 ft/s

$K = 100 \times 0.021 = 2.1$

In USCS units $h_{f5} = 2.1 \dfrac{9.44^2}{2 \times 32.17} = 2.91$ ft

In SI units $h_{f5} = 2.1 \dfrac{2.88^2}{2 \times 9.807} = 0.89$ m

Total pipe, valve, and fitting losses:

In USCS units $\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$

$= 1.84 + 92.9 + 0.020 + 0.29 + 0.077 + 0.30 + 2.91$

$= 98.34$ ft

In SI units $\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$

$= 0.56 + 28.39 + 0.062 + 0.089 + 0.023 + 0.093 + 0.89$

$= 30.05$ m

INCREASERS The head lost when there is a sudden increase in pipe diameter, with velocity changing from V_1 to V_2 in the direction of flow, can be calculated analytically. Computed results have been confirmed experimentally to be true to within ±3%. The head loss is expressed as shown, with K computed to be equal to unity:

$$h = K \frac{(V_1 - V_2)^2}{2g} = K\left[1 - \left(\frac{D_1}{D_2}\right)^2\right]^2 \frac{V_1^2}{2g} = K\left[\left(\frac{D_2}{D_1}\right)^2 - 1\right]^2 \frac{V_2^2}{2g} \tag{23}$$

The value of K is also approximately equal to unity if a pipe discharges into a relatively large reservoir. This indicates that all the kinetic energy $V_1^2/2g$ is lost and V_2 equals zero.

The loss of head for a gradual increase in pipe diameter when the flow is through a diffuser can be found from Figure 39 and Table 6A. The diffuser converts some of the kinetic

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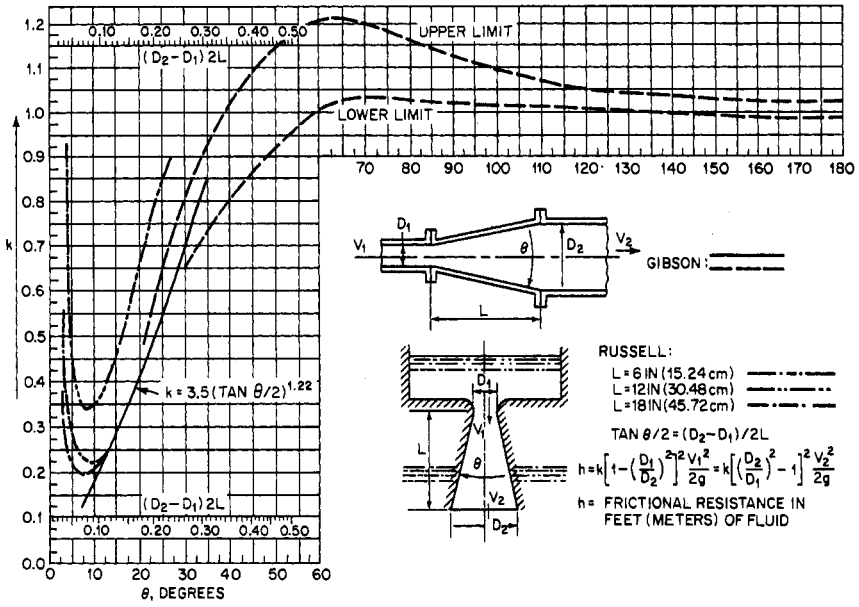


FIGURE 39 Resistance coefficients for increasers and diffusers (D = in or ft [m]; V = ft/s [m/s]) (Hydraulic Institute Engineering Data Book, Reference 5)

energy to pressure. Values for the coefficient used with Eq. 23 for calculating head loss are shown in Figure 39. The optimum total angle appears to be 7.5°. Angles greater than this result in shorter diffusers and less friction, but separation and turbulence occur. For angles greater than 50°, it is preferable to use a sudden enlargement.

REDUCERS Figure 40 and Table 6A give values of the resistance coefficient to be used for sudden reducers.

BENDS Figure 41 may be used to determine the resistance coefficient for 90° pipe bends of uniform diameter. Figure 42 gives resistance coefficients for bends that are less than 90° and can be used for surfaces having moderate roughness such as clean steel and cast iron. Figures 41 and 42 are not recommended for elbows with R/D below 1. Tables 6D and 7 give values of resistance coefficients for miter bends.

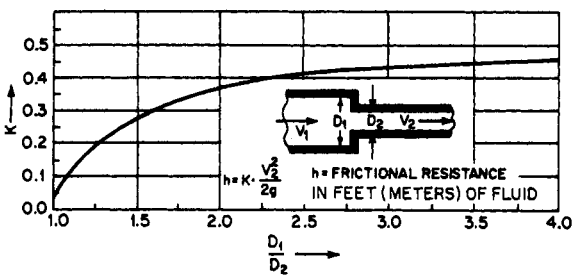


FIGURE 40 Resistance coefficients for reducers (Hydraulic Institute Engineering Data Book, Reference 5)

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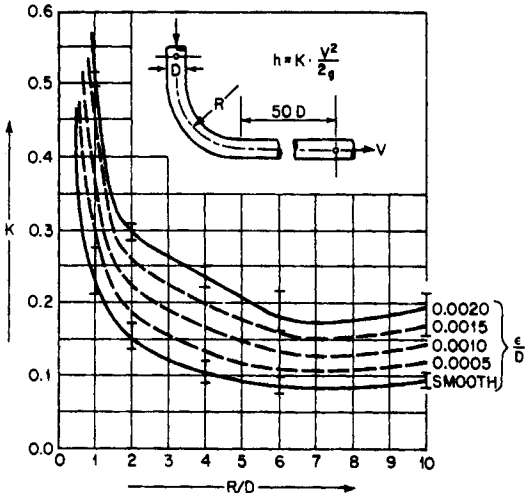


FIGURE 41 Resistance coefficients for 90° bends of uniform diameter (Hydraulic Institute Engineering Data Book, Reference 5)

PUMP SUCTION ELBOWS Figures 43 and 44 illustrate two typical rectangular to round reducing suction elbows. Elbows of this configuration are sometimes used under dry-pit vertical volute pumps. These elbows are formed in concrete and are designed to require a minimum height, thus permitting a higher pump setting with reduced excavation. Figure 43 shows a long-radius elbow, and Figure 44 a short-radius elbow. The resulting velocity distribution into the impeller eye and the loss of head are shown for these two designs.

METERS Orifice, nozzle, and venturi meters (Figures 45–47) are used to measure rate of flow. These meters, however, introduce additional loss of head into the pumping system. Each of these meters is designed to create a pressure differential through the primary element. The magnitude of the pressure differential depends on the velocity and density of the liquid and the design of the element. The primary element restricts the area of flow, increases the velocity, and decreases the pressure. An expanding section following the

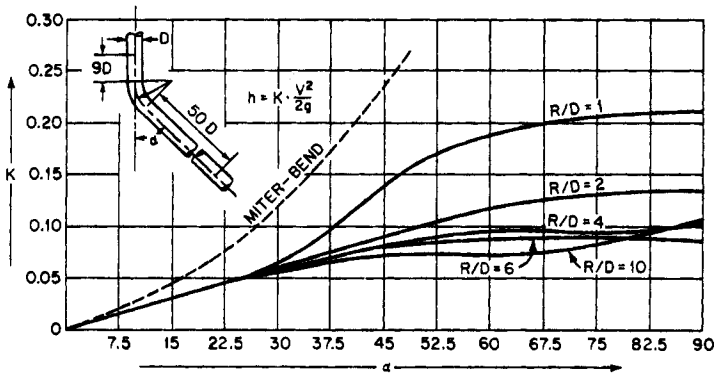
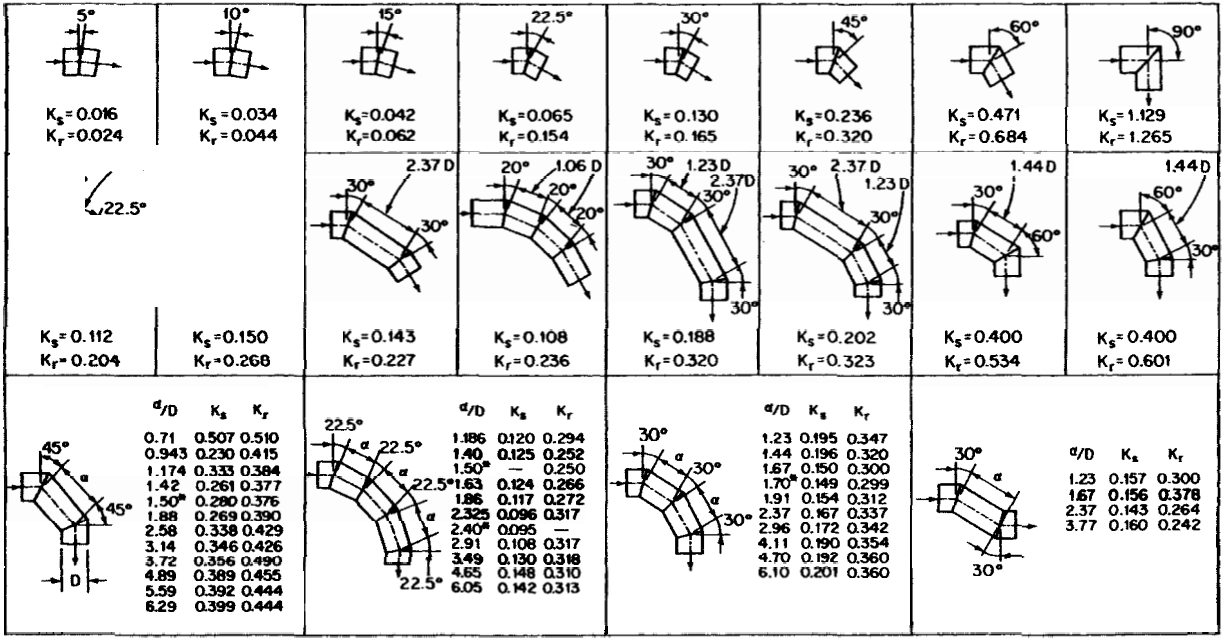


FIGURE 42 Resistance coefficients for bends of uniform diameter and smooth surface at Reynolds number $\approx 2.25 \times 10^5$ (Hydraulic Institute Engineering Data Book, Reference 5)

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TABLE 7 Resistance coefficients for miter bends at reynolds number $\approx 2.25 \times 10^5$



K_s = RESISTANCE COEFFICIENT FOR SMOOTH SURFACE
K_r = RESISTANCE COEFFICIENT FOR ROUGH SURFACE, $\frac{\epsilon}{D} = 0.0022$

*OPTIMUM VALUE OF α INTERPOLATED

Source: Hydraulic Institute Engineering Data Book, Reference 5

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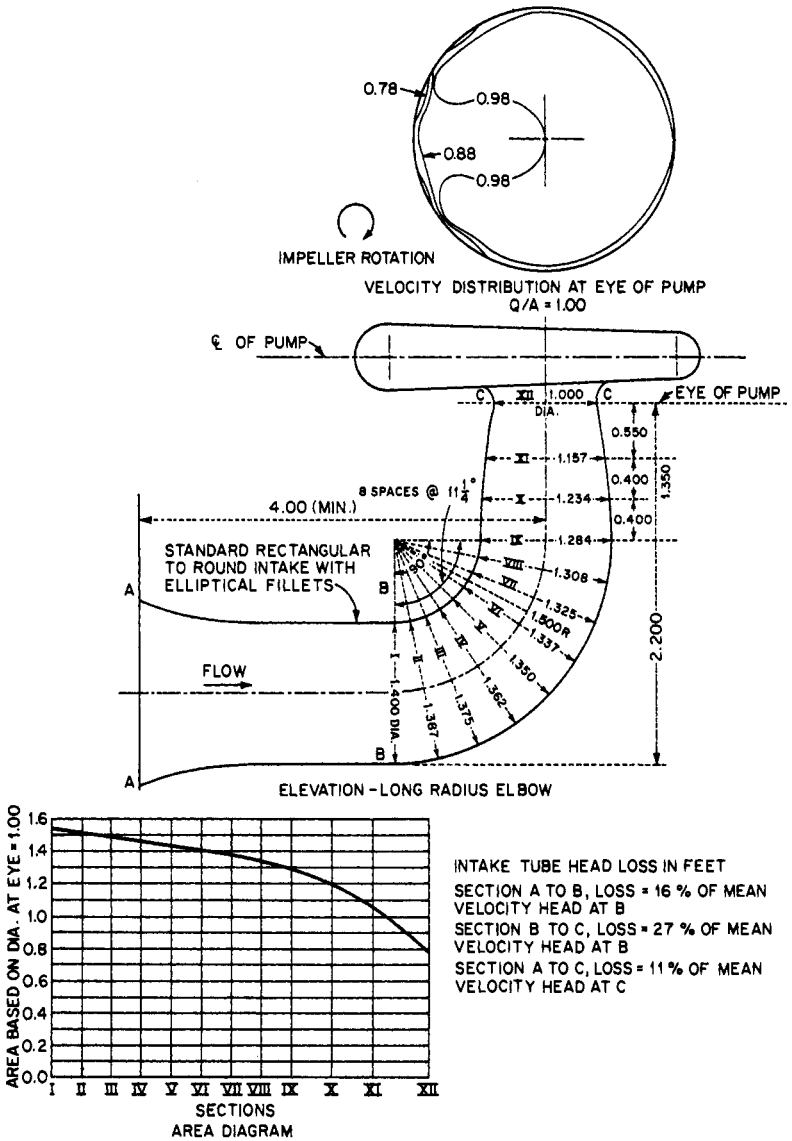


FIGURE 43 Head loss in a long-radius pump suction elbow (in × 25.4 = mm) (Reference 16)

primary element provides pressure head recovery and determines the meter efficiency. The pressure differential between inlet and throat taps measure rate of flow; the pressure differential between inlet and outlet taps measures the meter head loss (an outlet tap is not usually provided). Of the three types, venturi meters offer the least resistance to flow, and orifice meters the most.

When meters are designed and pressure taps located as recommended,[6] Figures 48–50 may be used to estimate the overall pressure loss. In these figures, the loss of pressure is expressed as a percentage of the differential pressure measured at the appropriate taps

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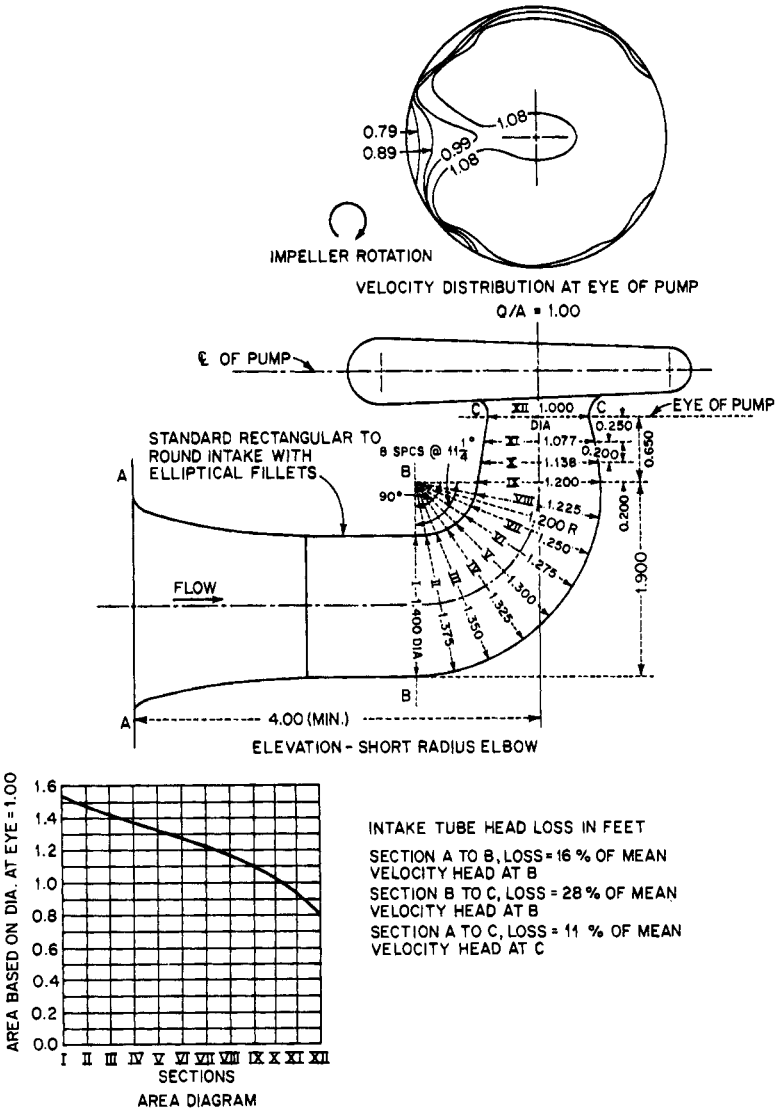


FIGURE 44 Head loss in a short-radius pump suction elbow (in × 25.4 = mm) (Reference 16)

and values are given for various sizes of meters. This loss of pressure is also the meter total head, or energy loss, because there is no change in velocity head if the pipe inside diameters are the same at the various measuring points. The meter loss of head should be in units of feet (meters) of liquid pumped if other system losses are expressed this way. Reference 6 should be consulted for information concerning formulas and coefficients for calculating differential pressure versus rate of flow.

Screens, Perforated Plates, and Bar Racks Obstructions to the flow of liquid in the form of multiple orifices uniformly distributed across an open or closed conduit may be used to remove solids, throttle flow, and produce or reduce turbulence. They may be used

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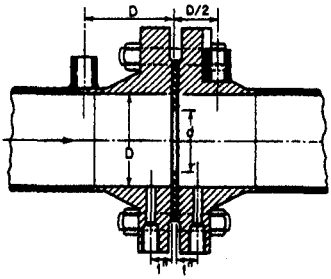


FIGURE 45 Thin-plate, square-edged orifice meter, showing alternate locations of pressure taps (Reference 6)

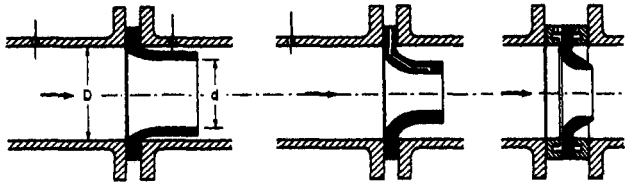


FIGURE 46 Shapes of flow nozzle meters and locations of pressure taps (Reference 6)

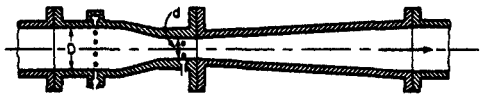


FIGURE 47 Herschel-type venturi meter, showing locations of pressure taps (Reference 6)

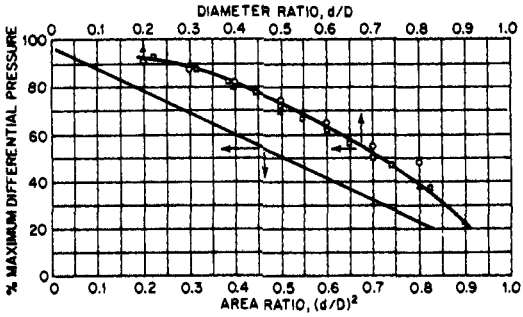


FIGURE 48 *Overall pressure loss across thin-plate orifices (Reference 6)*

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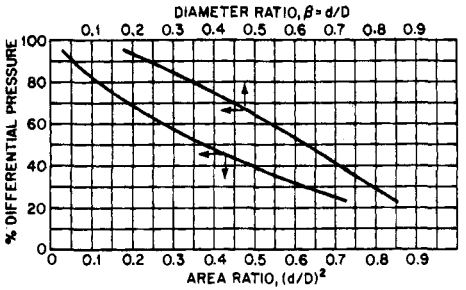


FIGURE 49 Overall pressure loss across flow nozzles (*Reference 6*)

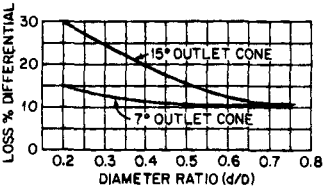


FIGURE 50 Overall pressure loss across venturi tubes (*Reference 6*)

upstream or downstream from a pump, depending on their purpose, and they therefore introduce a loss of head that must be accounted for. When an obstruction is placed upstream from a pump, a significant reduction in suction pressure and NPSH available can result.

The loss of head results from an increase in velocity at the entrance to the openings, friction, and the sudden decrease in velocity following the expansion of the numerous liquid jets. The total head loss is a function of the ratio of the total area of the openings to the area of the conduit before the obstruction, the thickness of the obstruction, the Reynolds number, and the velocities. Various investigators have determined values for resistance coefficients that can be multiplied by the approach velocity head to obtain the loss through these obstructions. According to Idel'chik,[7] loss of head h in feet (meters) may be calculated from the equation

$$h = K_1 \frac{V_1^2}{2g} \tag{24}$$

where K_1 = resistance coefficient
V_1 = average velocity in the conduit approaching the obstruction, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

ROUND-WIRE MESH SCREENS For flow having Reynolds numbers equal to or greater than 400, the resistance coefficient for flow through a round-wire, plain square mesh screen (Figure 51a) may be estimated as a function of percentage of open area using the equations

in USCS units
$$Re = \frac{V_o W_d}{\upsilon 12} \geq 400 \tag{25a}$$

in SI units
$$Re = \frac{V_o W_d}{\upsilon 1000} \geq 400 \tag{25b}$$

$$K_1 = k_0\left(1 - \frac{A_r}{100}\right) + \left(\frac{100}{A_r} - 1\right)^2 \tag{26}$$

$$A_r = 100\left(\frac{S}{S + W_d}\right)^2 \tag{27}$$

$$A_r = 100(1 - MW_d)^2 \tag{28}$$

where Re = screen Reynolds number referred to wire diameter
V_o = velocity through area of rectangular opening = $100V_1/A_r$, ft/s (m/s)

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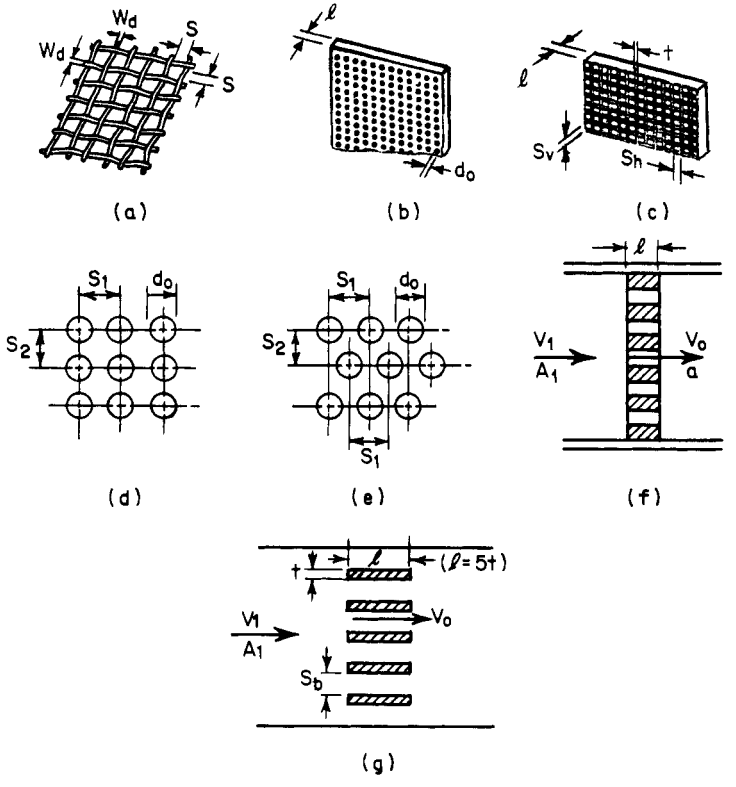


FIGURE 51 Explanation of terms used in Eq. 25–28 and 30–37 and Tables 8 and 9 for calculating resistance coefficient K, (a) round-wire, plain square, mesh screen, (b) round-hole perforated plate, (c) rectangular-grid perforated plate, (d) perforated plate, holes in vertical columns, (e) perforated plate, staggered holes, (f) perforated plate, cross section, (g) rectangular bar rack, cross section

W_d = screen wire diameter, in (mm)
υ = kinematic viscosity, ft²/s (m²/s)
k_0 = 1.0 for new, perfectly clean screens to 1.3 for normal screens
A_r = percentage of open area
S = square space between wires, in (mm)
M = mesh of screen or number of wires per in (mm)

Using water in a flow range of Reynolds numbers of approximately 60 to 800, Padmanabhan and Vigander[8] experimented with 1.3- to 12.0-mesh screens, 51.4 to 56% open area, and found their results to be comparable with those of other investigators. Values of the coefficient of resistance K_1, from the Padmanabhan-Vigander tests and others, vary from 2 decreasing asymptotically to 1 with increase in Reynolds number (Eq. 25) up to approximately 1000 for 47 to 56% open area. Smaller percentage open area values have larger coefficients.

Armour and Cannon,[9] using nitrogen as the fluid, tested various types of fine woven wire screens and derived equations for pressure drop in terms of a friction factor and the screen Reynolds number. Approach velocities ranged from 0.1 to 30 ft/s (0.03 to 9 m/s),

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TABLE 8 Plain square screen geometry

Screen mesh size M, in^{-1} (mm^{-1})	Wire diameter W_d, in $\times 10^{-4}$ (mm $\times 10^{-4}$)	Open area A_r, %	Screen constant C_1, $ft^{-1} \times 10^3$ ($m^{-1} \times 10^3$)	Screen constant C_2
30 × 30 (1.18 × 1.18)	9.45 (240)	94.4	2.553 (8.376)	1.553
150 × 150 (5.91 × 5.91)	2.36 (59.9)	93.0	8.267 (27.12)	2.594
250 × 250 (9.84 × 9.84)	1.69 (42.9)	91.7	16.21 (53.18)	2.934
400 × 400 (15.7 × 15.7)	1.00 (25.4)	92.2	30.68 (100.7)	2.680

Source: Reference 9.

resulting in Reynolds numbers from 350 to 275,000. To simply calculations, screen geometry constants C_1 = (surface area to unit volume ratio)^2 × (pore diameter) and C_2 = (screen thickness) ÷ (void fraction)^2 × (pore diameter) have been added to the reference authors' equations to obtain the following expression for screen head loss h in feet (meters):

$$h = C_2 \frac{V_1}{g}(8.61\upsilon C_1 + 0.52V_1) \tag{29}$$

where V_1 = average velocity in the conduit approaching the screen, ft/s (m/s)

Table 8 lists C_1 and C_2 values for a sample of plain square screens tested by Armour and Cannon.

PERFORATED PLATES AND BAR RACKS For flow having Reynolds numbers equal to or greater than 10^5, the resistance coefficients for flow through thick, square-edge perforated plates with round (Figure 51b) or rectangular (Figure 51c) openings and racks with rectangular cross-section bars (length = 5 times thickness; Figure 51g) may be calculated using Eq. 24, Table 9, and the following equations:

$$Re = \frac{V_o D_h}{\upsilon} \geq 10^5 \tag{30}$$

In USCS units

$$D_h = \frac{a}{3p} \quad \text{or} \quad D_h = \frac{d_o}{12} \tag{31a}$$

In SI units

$$D_h = 0.004\frac{a}{\rho} \quad \text{or} \quad D_h = \frac{d_o}{1000} \tag{31b}$$

for plates with round holes

$$A_r = 100\left(\frac{0.785d_o^2}{S_1 S_2}\right) \tag{32}$$

for plates with single hole in center

$$A_r = 100\left(\frac{d_o}{d_1}\right)^2 \tag{33}$$

for plates with square openings ($S = S_h = S_v$)

$$A_r = 100\left(\frac{S}{S+t}\right)^2 \tag{34}$$

for plates with rectangular openings

$$A_r = 100\left[\frac{S_h S_v}{(S_h + t)(S_v + t)}\right] \tag{35}$$

for any plate or bar rack

$$A_r = 100\frac{A_o}{A_1} \tag{36}$$

for single vertical or horizontal bar racks

$$A_r = 100\left(\frac{S_b}{S_b + t}\right) \tag{37}$$

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TABLE 9 Values of resistance coefficient K_1 for perforated plates and bar racks

	$A_r/100$ *or* A_o/A_1															
l/Dh	0.02	0.04	0.06	0.08	0.10	0.15	0.20	0.25	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.0
0	7000	1670	730	400	245	96.0	51.5	30.0	18.2	8.25	4.00	2.00	0.97	0.42	0.13	0
0.2	6600	1600	687	374	230	94.0	48.0	28.0	17.4	7.70	3.75	1.87	0.91	0.40	0.13	0.01
0.4	6310	1530	660	356	221	89.0	46.0	26.5	16.6	7.40	3.60	1.80	0.88	0.39	0.13	0.01
0.6	5700	1380	590	322	199	81.0	42.0	24.0	15.0	6.60	3.20	1.60	0.80	0.36	0.13	0.01
0.8	4680	1130	486	264	164	66.0	34.0	19.6	12.2	5.50	2.70	1.34	0.66	0.31	0.12	0.02
1.0	4260	1030	443	240	149	60.0	31.0	17.8	11.1	5.00	2.40	1.20	0.61	0.29	0.11	0.02
1.4	3930	950	408	221	137	55.6	28.4	16.4	10.3	4.60	2.25	1.15	0.58	0.28	0.11	0.03
2.0	3770	910	391	212	134	53.0	27.4	15.8	9.90	4.40	2.20	1.13	0.58	0.28	0.12	0.04
3.0	3765	913	392	214	132	53.5	27.5	15.9	10.0	4.50	2.24	1.17	0.61	0.31	0.15	0.06
4.0	3775	930	400	215	132	53.8	27.7	16.2	10.0	4.60	2.25	1.20	0.64	0.35	0.16	0.08
5.0	3850	936	400	220	133	55.5	28.5	16.5	10.5	4.75	2.40	1.28	0.69	0.37	0.19	0.10
6.0	3870	940	400	222	133	55.8	28.5	16.6	10.5	4.80	2.42	1.32	0.70	0.40	0.21	0.12
7.0	4000	950	405	230	135	55.9	29.0	17.0	10.9	5.00	2.50	1.38	0.74	0.43	0.23	0.14
8.0	4000	965	410	236	137	56.0	30.0	17.2	11.1	5.10	2.58	1.45	0.80	0.45	0.25	0.16
9.0	4080	985	420	240	140	57.0	30.0	17.4	11.4	5.30	2.62	1.50	0.82	0.50	0.28	0.18
10	4110	1000	430	245	146	59.7	31.0	18.2	11.5	5.40	2.80	1.57	0.89	0.53	0.32	0.20

Note: l = thickness of perforated plate or length of bars, ft (m); D_h, A_r, A_o, and A_1 are as defined following Eq. 37.
Source: Reference 7.

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where Re = Reynolds number referred to hydraulic diameter
V_O = velocity through area of opening, ft/s (m/s)
D_h = hydraulic diameter (= diameter if openings are round holes), ft (m)
v = kinematic viscosity, ft²/s (m²/s)
a = area of single opening, in² (mm²)
p = perimeter of single opening, in (mm)
d_O = diameter of hole, in (mm)
A_r = percentage of open area
S_1 = horizontal spacing of holes, in (mm)
S_2 = vertical spacing of holes, in (mm)
d_1 = diameter of approach, in (mm)
S_h = horizontal apace between vertical bars, in (mm)
S_v = vertical space between horizontal bars, in (mm)
t = thickness of plate laths or bars, in (mm)
A_O = total area of openings, ft² (m²)
A_1 = total area of approach, ft² (m²)
S_b = space between single vertical or horizontal bars, in (mm)

The loss of head through perforated plates may also be calculated by using an *orifice coefficient C,* as suggested by Smith and Van Winkle[10] and Kolodzie and Van Winkle.[11] Test results using air and other gases with equilateral-triangle pitch perforated plates are shown in Table 10 for Reynolds numbers 400 to 20,000. Using single-orifice relations, the following expression equates flow rate to pressure drop:

$$\omega = CA_o \sqrt{\frac{2g\Delta P}{\gamma\left[1 - \left(\frac{A_r}{100}\right)^2\right]}} \tag{38}$$

where ω = flow rate, ft³/s (m³/s)
C = orifice coefficient from Table 10
A_O = total area of openings, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)
ΔP = pressure drop, lb/ft² (N/m²)
γ = specific weight (force) of liquid, lb/ft³ (N/m³)
A_r = percentage of open area

The resistance coefficient K_1 and the orifice coefficient C may be interchanged in Eqs. 24 and 38 for loss through perforated plates:

$$C = \sqrt{\frac{1 - \left(\frac{A_r}{100}\right)^2}{K_1 - \left(\frac{A_r}{100}\right)^2}} \tag{39}$$

$$K_1 = \frac{1 - \left(\frac{A_r}{100}\right)^2}{C^2\left(\frac{A_r}{100}\right)^2} \tag{40}$$

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TABLE 10 Orifice coefficients C for equilateral-triangle pitch perforated plates

		l/D_h								
Pitch/D_h	Re	0.33	0.43	0.50	0.52	0.65	0.75	0.80	1.0	2.0
2.0	400–4000	0.74–0.69	0.75–0.71	0.76–0.74	—	0.79–0.85	0.82–0.86	0.83–0.87	0.84–0.89	0.77–0.92
2.0	4000–20,000	0.69	0.71	0.74	0.77	0.85	0.86	0.87	0.89	0.92
3.0	400–4000	0.70–0.67	0.72–0.68	0.73–0.72	—	0.75–0.81	0.76–0.82	0.77–0.84	0.78–0.85	0.70–0.87
3.0	4000–20,000	0.67	0.68	0.72	0.74	0.81	0.82	0.84	0.85	0.87
4.0	400–4000	0.71–0.66	0.72–0.67	0.72–0.69	—	0.73–0.77	0.74–0.79	0.75–0.81	0.76–0.53	0.69–0.84
4.0	4000–20,000	0.66	0.67	0.69	0.72	0.77	0.79	0.81	0.88	0.84
5.0	400–4000	0.68–0.65	0.69–0.66	0.7–0.68	—	0.72–0.76	0.73–0.77	0.74–0.78	0.75–0.81	0.65–0.82
5.0	4000–20,000	0.65	0.66	0.68	0.71	0.76	0.77	0.78	0.81	0.82

Note: Pitch/D_h = pitch-to-hole-diameter ratio; l/D_h = plate-thickness-to-hole diameter ratio; Re = Reynolds number
Source: References 10 and 11.

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Throttling Orifices In addition to measuring flow, orifices can be used to (a) reduce flow by adding artificial resistance to increase system head, (b) dissipate energy to provide a desired pressure reduction, and (c) create a high-velocity jet. Orifices for these purposes are called *throttling orifices.* In order to maintain a desired minimum flow to prevent damage to a centrifugal pump, a throttling orifice, or a series of throttling orifices, can be used in the bypass system. The orifice provides additional bypass resistance to maintain the required bypass flow.

A throttling orifice can be fabricated by drilling a hole in a metal plate (or through bar stock) that, when inserted between flanges in a pipe (or threaded to pipe), will create the desired loss of head at the design flow. The resistance coefficient K_1 may be calculated as if the throttling device were a single hole in a perforated plate, using Table 9 and Eqs. 30, 31, and 33, and the loss of head calculated using Eq. 24.

Energy is dissipated through a throttling orifice because pressure head is converted to velocity head and this conversion is followed by an inefficient pressure head recovery. Conditions may exist at the orifice *vena contracta* that could cause vaporization of the high-velocity, low-pressure liquid jet. Care must be taken in selecting the orifice size to avoid excessive cavitation noise or choke flow. An orifice cavitation index used to check the orifice selection is described by Tung and Mikasinovic,[12] who also discuss the use of orifices in series to avoid cavitation.

Although orifices are used to meter flow, accuracy requires that they be fabricated to standard proportions and that pressure taps be precisely located (Reference 6 and Figure 45). A distinction should be made between *meter differential pressure,* which is used to measure flow and is the difference in pressure at the upstream and downstream *vena contracta* taps, and *meter loss of head* calculated using the resistance coefficient K_1, which is the total overall loss of energy as measured at the upstream tap and past the downstream *vena contracta* tap. The loss of head through a standard orifice meter should be calculated as discussed previously under Meters.

Water Meters and Backflow Preventors Tables 11 and 12 from Reference 17 provide indicative values of losses through these devices.

PUMP FLOW, HEAD, AND POWER IN VARYING TEMPERATURE SYSTEMS

In a pumping system where the weight of liquid pumped is constant, the volumetric flow rate will vary through system components having different temperatures. An example would be the condensate and feedwater system in a steam power plant. The following equation may be used to calculate volumetric flow rate using the specific gravity corresponding to the temperature of the liquid at the location in the system where flow is required:

In USCS units
$$\text{gpm} = \frac{\text{lb/h}}{500\ (\text{sp. gr.})} \tag{41a}$$

In SI units
$$\text{m}^3/\text{h} = \frac{\text{kg/h}}{998\ (\text{sp. gr.})} \tag{41b}$$

When calculating the total head required of a pump or pumps to overcome total system component losses, the actual volumetric flow rate, and temperature through each component must be used because head loss is a function of velocity and viscosity. Information provided in this chapter permits computing pipe, valve, and fitting losses in ft · lb/lb or ft (N · m/N or m) of liquid passing through the component. If the pump is at a location in the system where the temperature is different than at the locations where the head losses are calculated, the total head to be produced by the pump cannot be found by simply adding the individual component heads.

The pump head required to produce a specific increase in pressure varies inversely with specific gravity. Therefore, to calculate pump total head, either the component total

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TABLE 11 Pressure losses for turbine-type water meters

Flow Rate, gpm	Pressure Loss, psi, when Meter Size is:						
	2 in	3 in	4 in	6 in	8 in	10 in	12 in
100	0.8	0.8	0.5				
150	3.5	3.5	1.5				
200	6.5	6.5	3.0				
250	10.0	10.0	4.5				
300		14.0	5.5				
350			7.0				
400			8.0	0.5			
450			9.0	1.0			
500			10.5	1.2	0.5	0.5	
600				2.0	1.0	0.6	
700				3.0	1.1	0.7	
800				4.0	1.4	0.8	
900				5.0	1.7	1.0	
1000				6.0	2.0	1.1	0.6
1500				13.0	4.5	2.0	1.0
2000					8.0	3.0	1.6
2500					13.0	4.0	3.0
3000						6.0	4.5
3500						8.5	6.0
4000						11.0	7.7
5000							11.0
6000							18.0

in × 25.4 mm gpm × .227 = m^3/h psi × .069 = bar

Source: Reference 17

heads must be converted to pump equivalent heads and added together, or all component losses must be expressed in pressure units so the total of these pressures can be converted to an equivalent pump head at the pump temperature. From Eqs. 2 and 3, the pump equivalent head is

$$h_2 = \frac{\gamma_1}{\gamma_2} h_1$$

or

$$h_2 = \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1$$

where the subscript 1 denotes the component and the subscript 2 denotes the pump. From Eq. 6, individual component head losses can be converted to lb/ft^2 (N/m^2) using

$$p_1 = h_1 \gamma_1$$

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TABLE 12 Pressure losses for backflow preventers

Flow Rate, gpm	Pressure Loss, psi, when Preventer Size is:					
	2 in	3 in	4 in	6 in	8 in	10 in
100	3.0					
150	5.5	1.5				
200		2.2	3.0			
300		4.0	2.0			
400			2.5	3.5		
500			3.0	2.5		
600			4.0	2.4	4.5	
700			5.0	2.6	3.6	
800				2.8	3.0	
900				3.0	2.8	
1000				3.3	2.5	4.0
1200				3.7	2.0	3.5
1400				4.3	2.0	2.9
1600				5.0	2.4	2.7
1800					2.6	2.6
2000					2.8	2.5
2500					3.5	2.5
3000						3.0
3500						3.6
4000						4.0
4500						5.0

in × 25.4 mm gpm × .227 = m^3/h psi × .069 = bar

Source: Reference 17

Also for Eq. 6,

$$TH = \frac{P_\Delta}{\gamma_2} = \frac{\Sigma_{p1}}{\gamma_2}$$

from which total component losses in lb/ft^2 (N/m^2) can be converted to an equivalent total pump head in feet (meters) of liquid to be produced at the pumping temperature.

In a varying temperature system, the positive or negative static head required to raise or lower the liquid pumped is not simply a difference in elevation. A pump must produce pressure in a pipe to raise liquid; the pressure required is proportional to the specific weight (force) of the liquid. The static head required at the pump should be found by expressing the suction and discharge elevation heads Z as pressures at the pump suction and discharge connections (corrected to the reference datum plane, if it is not at the pump centerline elevation) and using actual specific weights (forces) along the pipe. This differential pressure, in lb/ft^2 (N/m^2), is then converted to an equivalent static head using the specific weight (force) or specific gravity at the pump in the above appropriate equations.

When designing a pumping system, there may be several locations for placing a pump to produce a specified flow rate in lb/h (kg/h) and an increase in pressure in lb/ft^2 (N/m^2). If the temperature of the liquid varies at the different pump locations being considered (for example, before or after a feedwater heater in a steam power plant), the pump total head in feet (meters) and the volumetric flow rate in gpm (m^3/h) will vary. Although it is true

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that pump power is proportional to the product of volumetric flow × head × specific gravity, higher pumping temperature (lower specific gravity) will nevertheless result in higher pumping power. For the same conditions of weight (or mass) flow and differential pressure, pump power varies inversely with specific gravity because of the following relationships:

$$\text{Pump power} \propto \text{volumetric flow} \times \text{total head} \times \text{sp. gr.}$$

$$\text{Volumetric flow} \propto \frac{\text{weight or mass flow}}{\text{sp. gr.}}$$

$$\text{Total head} \propto \frac{\text{pressure}}{\text{sp. gr.}}$$

therefore,

$$\text{Pump power} \propto \frac{\text{weight or mass flow}}{\text{sp. gr.}} \times \frac{\text{pressure}}{\text{sp. gr.}} \times \text{sp. gr.}$$

then,

$$\text{Pump power} \propto \frac{(\text{weight or mass flow}) \times \text{pressure}}{\text{sp. gr.}}$$

Following are formulas for calculating pump input power in brake horsepower or brake kilowatts:

In USCS units

$$\text{bhp} = \frac{\text{gpm} \times TH \times \text{sp. gr.}}{3960 \times \text{pump eff.}} \tag{42a}$$

$$= \frac{\text{lb/h} \times \text{lb/in}^2}{858{,}600 \times \text{pump eff.} \times \text{sp. gr.}} \tag{43a}$$

In SI units

$$\text{bkW} = \frac{\text{m}^3\text{/h} \times \text{m} \times \text{sp. gr.}}{367.7 \times \text{pump eff.}} \tag{42b}$$

$$= \frac{\text{kg/h} \times \text{kPa}}{3{,}593{,}000 \times \text{pump eff.} \times \text{sp. gr.}} \tag{43b}$$

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SECTION 11.2
BRANCH-LINE PUMPING SYSTEMS

J. P. MESSINA

In some systems, the liquid leaving the pump or pumps will divide into a network of pipes. If the pump is of the centrifugal type, the total pump flow is dependent on the combined system resistance. The total pump flow and flow through each branch can be determined by the following methods. (Review "Pump Total Head and System-Head Curves" in Section 11.1.)

BRANCHES IN CLOSED-LOOP SYSTEMS

Figure 1 illustrates a pump and network of piping consisting of three parallel branches in series with common supply and return headers. Junction points 1 and 2 need not be at the same elevation (provided the liquid density remains constant and the pipes flow full and free of vapor) because, in a closed-loop system, the net change in elevation is zero. Figure 2 shows the system total-head curves for each branch line and header considered independent of the others. These curves are constructed for several flow rates by adding the frictional resistances of the pipes, fittings, and head losses through the equipment serviced from point 1 to point 2. Curves *A, B, C,* and *D* therefore represent the variation in system resistance in feet (meters) versus flow through each branch and header.

If the valves are open in all branches, the total system resistance, total pump flow, and individual branch flows are found by the following method. First observe that (a) the total flow must be equal to the sum of the branch flows, (b) the head loss or pressure drop across each branch from junction 1 to junction 2 is identical, and (c) the flow divides to produce these identical head losses. Therefore, at several head points, add together the flow through each branch and obtain curve $A + B + C$. *Header D* is in series with branches *A, B,* and *C,* and their system heads are added together for several flow conditions to obtain

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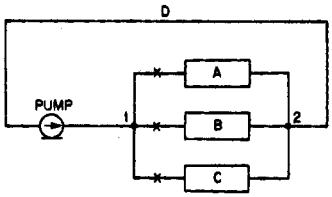


FIGURE 1 Closed-loop pumping system with branch lines

curve $(A + B + C) + D$. On curve E, the head-capacity characteristics of a centrifugal pump, point X represents the pump flow because at this point the system total head and pump total head are equal. Point Y_{1-2} represents the total head across points 1 and 2, and this head determines the flow through each branch; consequently points a, b, and c give individual branch flows. Curve F represents the head-capacity characteristics of a positive displacement pump (constant capacity) that would produce the same flow conditions.

If valve A is open and valves B and C are closed, Figure 3 shows the construction of the curves required to determine pump flow point X'. Obviously the pump flow and branch A flow are the same. Note that the total flow of point X is less than when all valves are open as a result of an increase in system head. If all valves were open and the total flow was obtained by a positive displacement pump having a constant capacity curve F, closing valves B and C would not change the flow. The system head would, however, increase to point X'' and the head would be greater than for a centrifugal pump having curve E.

Also shown in Figure 3 are the system total-head curves for different combinations of open valves A, B, and C and the resulting flow caused by a pump having characteristic curve E. For these various valve combinations, the head differential across the junction points is found by subtracting the head of the curve D from the system total head for the condition investigated, for example, point Y'_{1-2} for only valve A open. The intersection of a horizontal line through point Y'_{1-2} and the individual branch curves gives the branch flow, as illustrated in Figure 2.

BRANCHES IN OPEN-ENDED SYSTEMS

Figure 4 illustrates a pump supplying three branch lines that are open-ended and terminate at different elevations. Figure 5 shows the system total-head curves for each branch

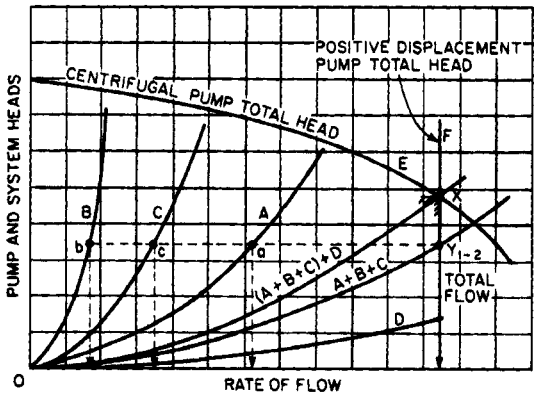


FIGURE 2 System-head curves for pump and branch lines shown in Figure 1 with all valves open

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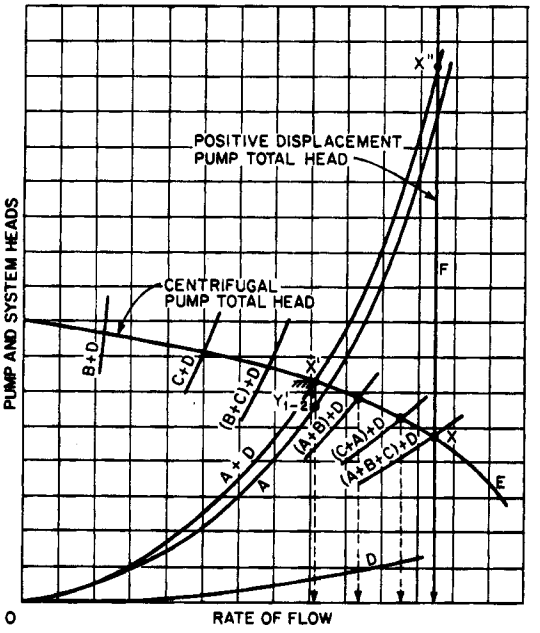


FIGURE 3 System-head curves for pump and branch line shown in Figure 1 with different combinations of open valves

line and main supply line considered independently of each other. These curves are constructed by starting at elevation heads Z_A, Z_B, Z_C, and Z_{DA} at zero flow. To each of these heads is added the frictional resistances in each line for several flow rates. Frictional losses from the suction tank to junction 1 are included in curve D. Curves A, B, C, and D therefore represent the variation in system resistance in feet (meters) versus flow through each branch and supply line. Note that Z_D is negative because in line D there is a decrease in elevation to point 1.

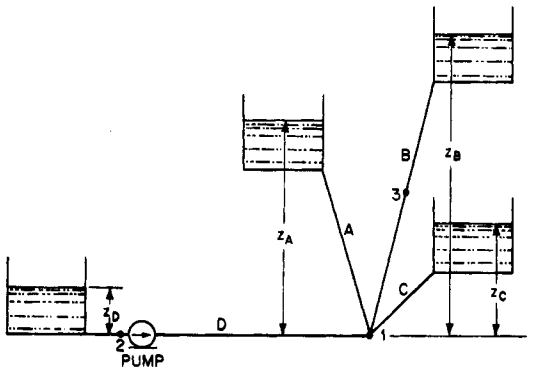


FIGURE 4 Open-ended pumping system with branch lines

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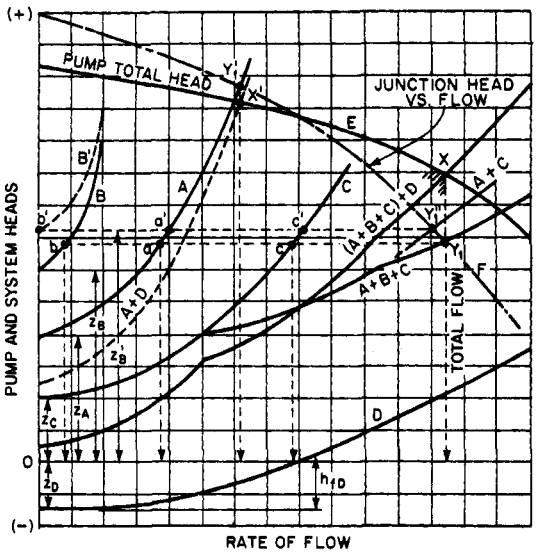


FIGURE 5 System-head curves for pump and branch lines shown in Figure 4

The total head at the junction is the head Z_D in the suction tank measured above point 1 plus the pump total head less the frictional head loss h_{fD} in line D, and it varies with flow, as illustrated by curve F.

The total system resistance, total pump flow, and individual branch flows are found by the following method. First observe that (a) the total flow must be equal to the sum of the branch flows, (b) the frictional resistance plus the elevation head measured relative junction 1 for each branch is identical, and (c) the flow divides to produce these identical total branch heads. Therefore, at several head points, add together the flow of each branch to obtain curve $A + B + C$. Supply line D is in series with branches A, B, and C, and their system heads are added algebraically for several flow conditions to obtain curve $(A + B + C) + D$. If curve E is the head-capacity characteristics of a centrifugal pump, point X represents the pump flow because at this point the system total head and pump total head are equal. Point Y_1 represents the total head at junction 1, and this head determines the flow through each branch; consequently points a, b, and c give individual branch flows.

In this example the pump discharges to all tanks, but this should not be assumed. There is a limiting liquid level elevation for each tank, and, if this level is exceeded, flow will be from the tank into the junction. Therefore, it is possible for the lower-level tanks to be fed by the higher-level tank and the pump. The limit for the liquid elevation in tank B is Z'_B, and it is found from the intersection of curve $A + C$ with curve F, point Y''_1. The flow in branches A and C is at rates a' and c' when there is no flow in branch B. This is also a condition similar to closing a valve in branch B.

If elevation Z_B is greater than previously found limiting height Z'_B flow in branches A and C is determined in the following manner. Construct a curve for junction head versus flow by adding heads and flows that result when the pump and suction tank are in a series with each other and tank B (less line losses) is in parallel with the pump and suction tank. The intersection of this curve with curve $A + C$ will give the junction head required to determine the individual flows from the pump and tank and the flows to tanks A and C (not illustrated).

If flow to branches B and C is shut off, Figure 5 illustrates the construction of the curves required to determine the pump flow point X' and junction head point Y'_1.

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CENTRIFUGAL PUMP BYPASS

Bypass orifices around centrifugal pumps are often used to maintain a minimum flow recommended by the pump manufacturer because of one or more of the following reasons:

- To limit the temperature rise to prevent seizing and cavitation
- To reduce shaft and bearing loads
- To prevent excessive recirculation in the impeller and casing
- To prevent overloading of driver if pump power increases with decrease in flow

Figure 6 illustrates a system that under certain conditions reduces pump flow below the recommended minimum. The pump delivers its flow to either tank A or tank B. Figure 7 shows the separate system-head curves for flow to tank A and for flow to tank B. Curve E is the head-capacity characteristics of the centrifugal pump. Individual flow rates to each tank are shown as Q_A and Q_B. The recommended minimum flow is Q_R, which is greater than Q_B by the amount shown. In order to maintain the minimum flow, a bypass orifice with necessary pipe, valves, and fittings is required to pass flow Q_c at total head H_R is when the pump discharges to tank B only.

Figure 8, (curve C) shows the construction necessary to determine the required bypass head versus flow characteristics of the orifice and pipe. The bypass system-head curve C includes the pipe, valve, and fitting losses from the pump connection between the suction tank and the end of the bypass piping below the suction water level. These losses must be deducted from the total bypass losses to determine the required orifice head.

Figure 9 illustrates the resultant pump flow with the bypass in operation. Curve C is added to curve B to obtain curve $B + C$ by combining flows through each system at the same heads. Note that flow through the piping from the suction tank to junction 1 is the total from both systems. Therefore, the combined system-head curve $B + C$ should take this into consideration. Similarly, curve C is added to curve A to obtain curve $A + C$. Note that when the flow is directed to tank B with the bypass open, pump flow is increased from Q_B to Q_R and tank flow is decreased from Q_B to Q'_B. When the flow is directed to tank A with the bypass open, pump flow is increased from Q_A to Q_p and tank flow is decreased from Q_A to Q'_A.

If it is desired that there be no reduction in flow or that there be no waste of pumping power when flow is to tank A, the bypass can be closed either manually or automatically. If pump flow is monitored, this measurement can be used to open, close, or modulate the bypass valve automatically to maintain desired flow. Refer to Section 11.4 for more detailed information.

If operating procedures require that the pump occasionally be run with a closed valve (at the pump discharge or at tanks A and B), the bypass line must be designed to recirculate all of the minimum required pump flow Q_R, dissipating head H_R, shown as curve C' and Figures 8 and 9.

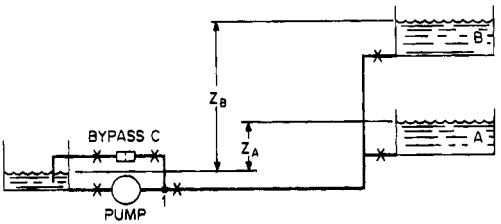


FIGURE 6 Pump with bypass to maintain minimum flow

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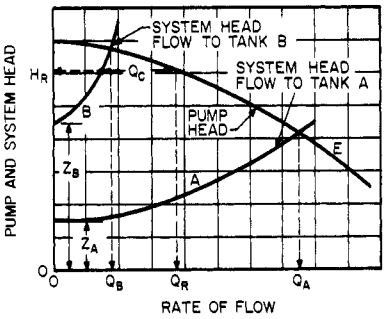


FIGURE 7 System-head curves for pump and tanks shown in Figure 6 with bypass valve closed

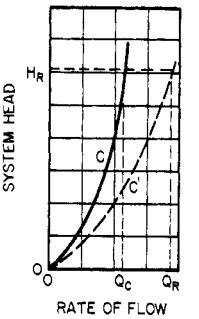


FIGURE 8 Bypass orifice system requirements

FLOW CONTROL THROUGH BRANCHES

The flow through branches *A, B,* and *C* in Figures 1 and 4 is dependent on the individual branch characteristics. When parallel branches are connected to a pump, the resulting division of flow may not satisfy the requirements of the individual lines. If it is desired that the flow to each branch meet or exceed specified individual line requirements, it is necessary only to select a pump to provide the maximum head required by any one branch. In those branches where this head is more than required, the flow will be greater than the desired amount. A throttling valve or other flow-restricting device may be used to reduce the flow in these branches to the desired quantity. If the flow is controlled in this manner, the pump need be selected to produce only the minimum total system flow at a total head required to satisfy the branch needing the highest head at junction 1 in Figures 1 and 4.

The flow through a branch is sometimes dictated by the requirement of a component elsewhere in the system. For example, in the system shown in Figure 10, the required flow through component *V* could be greater than the sum of the required flows through components *A, B,* and *C*. An additional branch line and control valve *F* around components *A, B,* and *C* may be used to bypass the additional flow needed by component *D*. Individual component throttling valves may also be used, if needed, to adjust the flow in each branch.

The following example illustrates how flow through branches may be controlled and how pump total head is calculated.

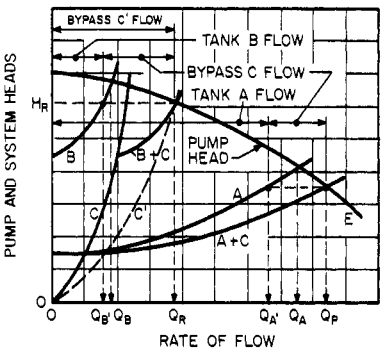


FIGURE 9 System-head curves for pump and tanks shown in Figure 6 with bypass valve open

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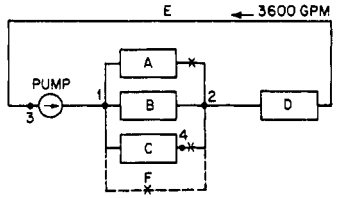


FIGURE 10 Example of a branch-flow pumping system

EXAMPLE A pump is required to circulate water at a rate of 3600 gpm (817 m^3/h) through the system shown in Figure 10. The head versus flow characteristics of the system components *A, B,* C, *D,* and *E* (system pipe and fittings) are shown in Figure 11. The branch pipe and fitting losses from point 1 to point 2 are included in the total heads for components *A, B,* and *C*. Determine

1. The pump total head required and the individual flows through components *A, B,* and *C*
2. The pump total head required if the flow through components *A, B,* and *C* need be only 800, 700, and 1000 gpm (182, 159, and 227 m^3/h), respectively, and a bypass *F* is installed

Calculate the individual throttling valve head drops to achieve a controlled branch flow system.

Solution

1. Head versus flow curves *A, B,* and *C* of Figure 11 are added together in parallel, giving curve *(A + B + C)*. Curves (A + *B* + *C*), *D*, and *E* are added together in

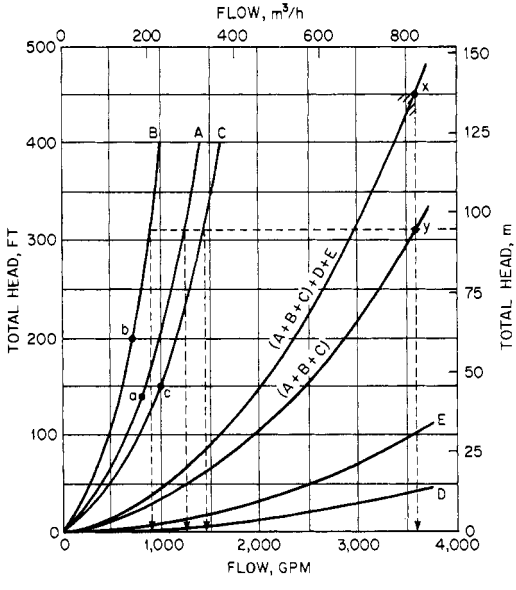


FIGURE 11 System-head curves required for solutions to example problems

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series, resulting in curve $(A + B + C) + D + E$. This latter curve indicates that 450 ft (137 m) total pump head is required at 3600 gpm (817 m³/h), point *X*. Curve $(A + B + C)$ crosses point *Y* at 3600 gpm (817 m³/h) flow through the branches, and this condition requires 310 ft (94.5 m) total head, which is the head across branch points 1 and 2. From each individual component curve, the flow through branches *A*, *B*, and *C* can be read as 1250, 900, and 1450 gpm (284, 204, and 329 m³/h), respectively.

2. Because the total flow through components *A*, *B*, and *C* need be only 2500 gpm (800 + 700 + 1000) [568 m³/h (182 + 159 + 227)], the bypass should be designed to pass 1100 gpm (249 m³/h). Component *B* requires the maximum head, 200 ft (61 m) differential (point *b*) across points 1 and 2. Throttling valves are needed in components *A* and *C* and bypass *F* to increase the head in each branch to 200 ft (61 m) at the required flows. Branch *A* (point *a*) requires only 140 ft (42.7 m) total head to pass 800 gpm (182 m³/h); therefore, the throttling valve must be designed for a 60-ft (42.7-m) head loss. Branch *C* (point *c*) requires only 150 ft (45.7 m) total head to pass 1000 gpm (227 m³/h), requiring a throttling valve for a 50-ft (15.3-m) head loss. The bypass control valve and piping should be designed to produce a 200-ft (61-m) head drop at 1100 gpm (249 m³/h).

At 3600 gpm (817 m³/h), the pump is now required to overcome 200 ft (61 m) total head across points 1 and 2, 40 ft (12.2 m) total head through component *D*, and 100 ft (30.1 m) total head through the system pipe and fittings, component *E*, for a total of 340 ft (103.3 m). The reduction in pumping head from 450 to 340 ft (137.1 to 103.6 m), a saving of 110 ft (33.5 m), or 24.4% water power, is the result of decreasing the branch head from 310 to 200 ft (94.5 to 61 m) by bypassing the excess flow.

PUMP TOTAL HEAD IN BRANCH-LINE SYSTEMS

The total head produced is the difference in total heads measured across the suction and discharge connections of a pump. As explained in Section 11.1, the total head is also the difference between the heads at any two points in the pumping system, one on each side of the pump, plus the sum of the head losses between these two points. Confusion sometimes results when the flow through the pump divides into branches in either closed-loop or open-ended systems. The points of head measurement can be in any branch line, upstream or downstream from the pump, regardless of the flow rate in these lines.

In part 2 of the previous example, the pump or system total head could be measured using points 3 and 4 in Figure 10. If the total head measured at the pump suction, point 3, were 25 ft (7.62 m) gage, the head measured at point 4 would be 205 ft (62.48 m) gage above the same reference datum plane, assuming 10 ft (3.05 m) of friction between the pump discharge and point 1. The difference between the head at point 3 and that at point 4 is 180 ft (54.86 m). The loss of head due to friction and the head drop through component *C* is 10 + 150 = 160 ft (3.05 + 47.5 = 48.75 m). The pump and system total head at 3600 gpm (817 m³/h) is therefore 180 + 160 = 340 ft (54.86 + 48.75 = 103.63 m).

Similarly, the pump and system total head for an open-ended system, such as the one shown in Figure 4, could be found by measuring, for example, the difference between the head at point 2 and that at point 3. Each head measurement would be referred to a common datum plane. The total head loss from 2 to 1 plus the head loss from 1 to 3, at the rate of flow in their respective lines, added to the difference between the head at 2 and that at 3, is the pump and system total head.

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SECTION 11.3
WATERHAMMER

C. SAMUEL MARTIN

INTRODUCTION

By definition waterhammer is a pressure (acoustic) wave phenomenon created by relatively sudden changes in the liquid velocity. In pipelines, sudden changes in the flow (velocity) can occur as a result of (1) pump and valve operation in pipelines, (2) vapor pocket collapse, or (3) even the impact of water following the rapid expulsion of air out of a vent or a partially open valve. Although the name waterhammer may appear to be a misnomer in that it implies only water and the connotation of a "hammering" noise, it has become a generic term for pressure wave effects in liquids. Strictly speaking, waterhammer can be directly related to the compressibility of the liquid, primarily water, in this handbook. For slow changes in pipeline flow for which pressure waves have little to no effect, the unsteady flow phenomenon is called surging.

Potentially, waterhammer can create serious consequences for pipeline designers if not properly recognized and addressed by analysis and design modifications. There have been numerous pipeline failures of varying degrees and resulting repercussions of loss of property and life. Three principal design tactics for mitigation of waterhammer are (1) alteration of pipeline properties such as profile and diameter, (2) implementation of improved valve and pump control procedures, and (3) design and installation of surge control devices.

In this section waterhammer and surging are defined and discussed in detail with reference to the two dominant sources of waterhammer—pump and/or valve operation. Detailed discussion of the hydraulic aspects of both valves and pumps and their effect on hydraulic transients will be presented. The undesirable and unwanted, but often potentially possible, event of liquid column separation and rejoining are a common justification for surge protection devices. Both the beneficial and detrimental effect of free (entrained or entrapped) air in water pipelines will be discussed with reference to waterhammer and surging. Finally, the efficacy of various surge protection devices for mitigation of waterhammer is included.

NOMENCLATURE

a	Sonic (acoustic) velocity of plane wave in a pipe
a_o	Sonic (acoustic) velocity of plane wave in a pipe
A	Cross-sectional area of pipe
A_v	Projected area of partially open valve
A_{vo}	Projected area of fully open valve

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C_f	Valve discharge coefficient based on A_v and ΔH (partially open valve)
C_{fo}	Valve discharge coefficient based on A_{vo} and ΔH (fully open valve)
$C_H=gH/\omega^2D^2$	Dimensionless head coefficient
C_Q	Valve discharge coefficient based on A_{vo} and H
$C_Q=Q/\omega D^3$	Dimensionless flow coefficient
$C_P=P/\rho\omega^3D_I^5$	Dimensionless power coefficient
$C_T=T/\rho\omega^2D_I^5$	Dimensionless torque coefficient
D	Internal diameter of pipe
D_I	Diameter of pump impeller
e	Thickness of pipe wall
E	Elastic (Young's) modulus of pipe wall material
g	Gravitational acceleration
$h=H/H_R$	Dimensionless head ratio
H	Total dynamic head (TDH) across pump
H_R	Rated head, preferably at best efficiency point (BEP)
ΔH	Energy (head) loss attributed to valve
K	Bulk modulus of fluid
K_L	Head loss coefficient based upon ΔH and V
K_{Lo}	Head loss coefficient when valve is fully open
L	Length of pipe
N	Rotational speed of pump impeller (rpm)
N_R	Rated rotational speed of pump impeller
P	Power input on pump shaft
P_R	Rated power of pump
Q	Discharge through pump. Positive during normal pumping
Q_R	Rated discharge through pump
t_c	Valve closure time
t_e	Elastic time constant $(2L/a)$
t_f	Inertia time constant $(LV_o)/(gH_o)$
$T=P/\omega$	Fluid torque on pump shaft
T_R	Rated fluid torque on pump shaft
$v=Q/Q_R$	Dimensionless discharge ratio
V	Fluid velocity
$\alpha=N/N_R$	Dimensionless rotational speed of pump impeller
$\beta=T/T_R$	Dimensionless torque ratio
ρ	Fluid mass density
ω	Rotational speed of pump impeller (rad/sec)
ω_R	Rated rotational speed of pump impeller (rad/sec)

FUNDAMENTALS OF WATERHAMMER AND SURGE

The fundamentals of waterhammer, an elastic process; and surging, an incompressible phenomenon; are both developed on the basis of the basic conservational relationships of physics or fluid mechanics. The acoustic velocity stems from mass balance (continuity),

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while the fundamental waterhammer equation of Joukowsky originates from the application of linear momentum.

Definitions Some of the terms frequently employed in waterhammer are defined as follow:

Waterhammer. A pressure wave phenomenon for which liquid compressibility plays a role.

Surging. An unsteady phenomenon governed solely by inertia. Often termed mass oscillation or referred to as either rigid column or inelastic effect.

Liquid column separation. The formation of vapor cavities and their subsequent collapse and associated waterhammer on rejoining.

Entrapped air. Free air located in a pipeline as a result of incomplete filling, inadequate venting, leaks under vacuum, air entrained from pump intake vortexing, and other sources.[1]

Acoustic velocity. The speed of a waterhammer or pressure wave in a pipeline.

Joukowsky equation. Fundamental relationship relating waterhammer pressure change with velocity change and acoustic velocity. Strictly speaking, valid only for sudden flow changes.

Acoustic Velocity For wave propagation in liquid-filled pipes the acoustic (sonic) velocity is modified by the pipe wall elasticity by varying degrees, depending upon the elastic properties of the wall material and the relative wall thickness. The expression for the wave speed is

$$a = \frac{\sqrt{K/\rho}}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} = \frac{a_o}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} \tag{1}$$

where E is the elastic modulus of the pipe wall, D is the inside diameter of the pipe, e is the wall thickness, and a_o is the acoustic velocity in the liquid medium. In a very rigid pipe, or in a tank, or in large water bodies, the acoustic velocity a reduces to the well-known relationship $a = a_o = /(K/\rho)$. For water $K = 2.19$ GPa (318,000 psi) and $\rho = 998$ kg/m^3 (1.936 slug/ft^3 or 62.3 lbm/ft^3), yielding a value of $a_o = 1483$ m/s (4865 ft/s), a value many times that of any liquid velocity V.

Joukowsky (Waterhammer) Equation There is always a pressure change Δp associated with the rapid velocity change ΔV across a waterhammer (pressure) wave. The relationship between Δp and ΔV from the basic physics of linear momentum yields the well-known Joukowsky equation.

$$\Delta p = -\rho a \Delta V \tag{2}$$

where ρ is the liquid mass density, and a is the sonic velocity of the pressure wave in the fluid medium in the conduit. Conveniently using the concept of head, the Joukowsky head rise for instantaneous valve closure is

$$\Delta H = \frac{\Delta p}{\rho g} = -\frac{\rho a \Delta V}{\rho g} = \frac{a V_o}{g} \tag{3}$$

The compliance of a conduit or pipe wall can have a significant effect on modification of (1) the acoustic velocity and (2) any resultant waterhammer, as can be shown from Eq. 1 and Eq. 2, respectively. For simple waterhammer waves for which only radial pipe motion (hoop stress) effects are considered, the germane physical pipe properties are Young's elastic modulus (E) and Poisson ratio (μ). Table 1 summarizes appropriate values of these two physical properties for some common pipe materials.

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TABLE 1 Physical properties of common pipe materials

Material	Young's modulus, E GPa	Young's modulus, E 10^6 psi	Poisson's ratio μ
Asbestos cement	23 to 24	3.3 to 3.5	–
Cast iron	80 to 170	11.6 to 24.7	0.25 to 0.27
Concrete	14 to 30	2.0 to 4.4	0.10 to 0.15
Concrete (reinforced)	30 to 60	4.4 to 8.7	–
Ductile iron	172	25.0	0.30
Polyethylene	0.7 to 0.8	0.10 to 0.12	0.46
PVC (polyvinyl chloride)	2.4 to 3.5	0.35 to 0.51	0.46
Steel	200 to 207	29.0 to 30.0	0.30

The effect of the elastic modulus (E) on the acoustic velocity in water-filled circular pipes for a range of the ratio of internal pipe diameter to wall thickness (D/e) is shown in Figure 1 for various pipe materials.

HYDRAULIC CHARACTERISTICS OF VALVES

Valves are integral elements of any piping system used for the handling and transport of liquids. Their primary purposes are flow control, energy dissipation, and isolation of portions of the piping system for maintenance. It is important for the purposes of design and final operation to understand the hydraulic characteristics of valves under both steady and unsteady flow conditions. Examples of dynamic conditions are direct opening or closing of valves by a motor, the response of a swing check valve under unsteady conditions, and the action of hydraulic servo valves. The hydraulic characteristics of valves under either noncavitating or cavitating conditions vary considerably from one type of valve design to another. Moreover, valve characteristics also depend upon particular valve design for a special function, upon absolute size, on manufacturer as well as the type of pipefitting employed. In this section the fundamentals of valve hydraulics are presented in terms of pressure drop (head loss) characteristics. Typical flow characteristics of selected valve types of control—gate, ball, butterfly, and cone—are presented. The hydraulic characteristics of swing check valves and their dynamic response are also described.

Descriptions of Various Types of Valves Valves used for the control of liquid flow vary widely in size, shape, and overall design due to vast differences in application. They can vary in size from a few millimeters in small tubing to many meters in hydroelectric installations, for which spherical and butterfly valves of very special design are built. The hydraulic characteristics of all types of valves, albeit different in design and size, can always be reduced to the same basic coefficients, notwithstanding fluid effects such as viscosity and cavitation. Figure 2 shows cross-sections of some valve types to be discussed with relation to hydraulic performance.

Definition of Geometric Characteristics of Valves The valve geometry, expressed in terms of cross-sectional area at any opening, sharpness of edges, type of passage, and valve shape, has a considerable influence on the eventual hydraulic characteristics. In order to understand the hydraulic characteristics of valves it is useful, however, to express the projected area of the valve in terms of geometric quantities. With reference to Figure 2 the ratio of the projected open area of the valve A_v to the full open valve A_{vo} can be related to the valve opening, either a linear measure for a gate valve, or an angular one for rotary valves such as ball, cone, plug, and butterfly types. It should be noted that this geometric feature of the valve clearly has a bearing on the valve hydraulic performance, but should not be used directly for prediction of hydraulic performance—either steady state or transient.

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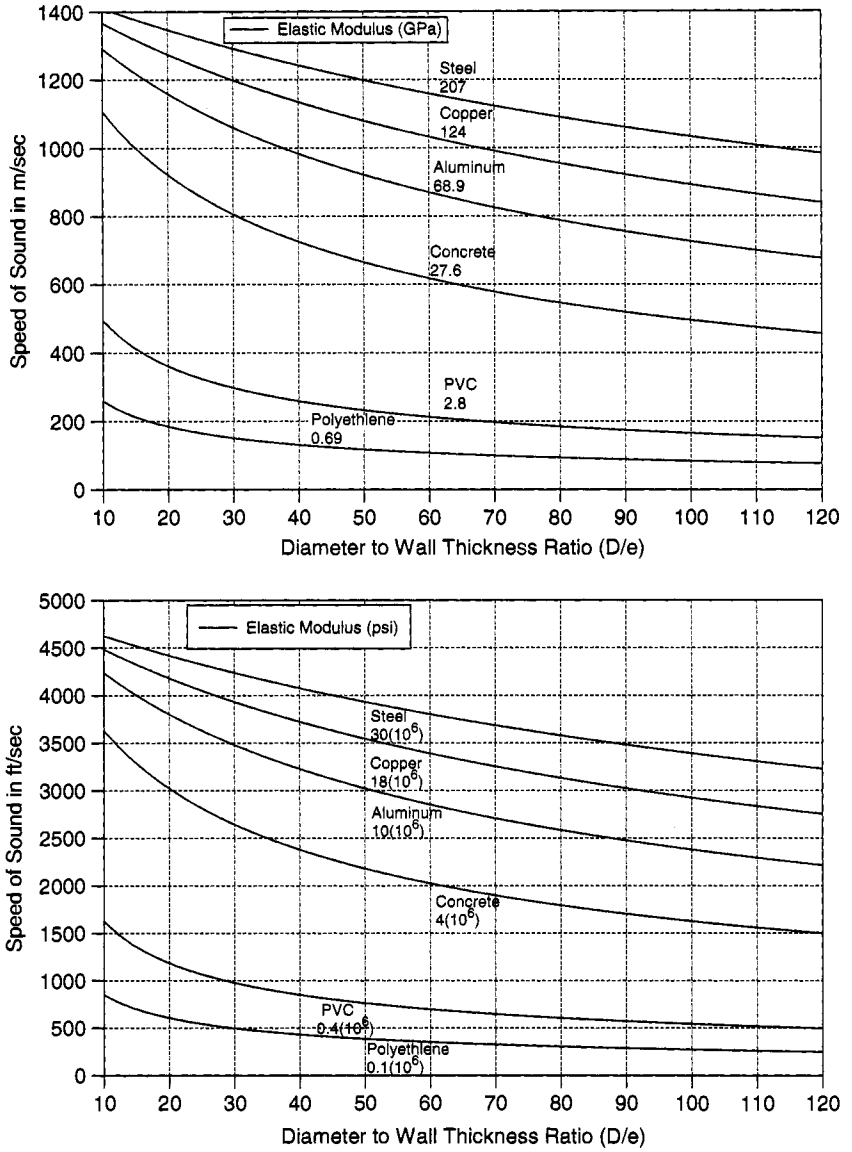


FIGURE 1 Effect of wall thickness of various pipe materials on acoustic velocity in water-filled pipes

The actual hydraulic performance to be used in transient calculations should originate from experiment.

Definition of Hydraulic Performance of Valves The hydraulic performance of a valve depends upon the flow passage through the valve opening and the subsequent recovery of pressure. The hydraulic characteristics of a valve under partial to fully opened conditions

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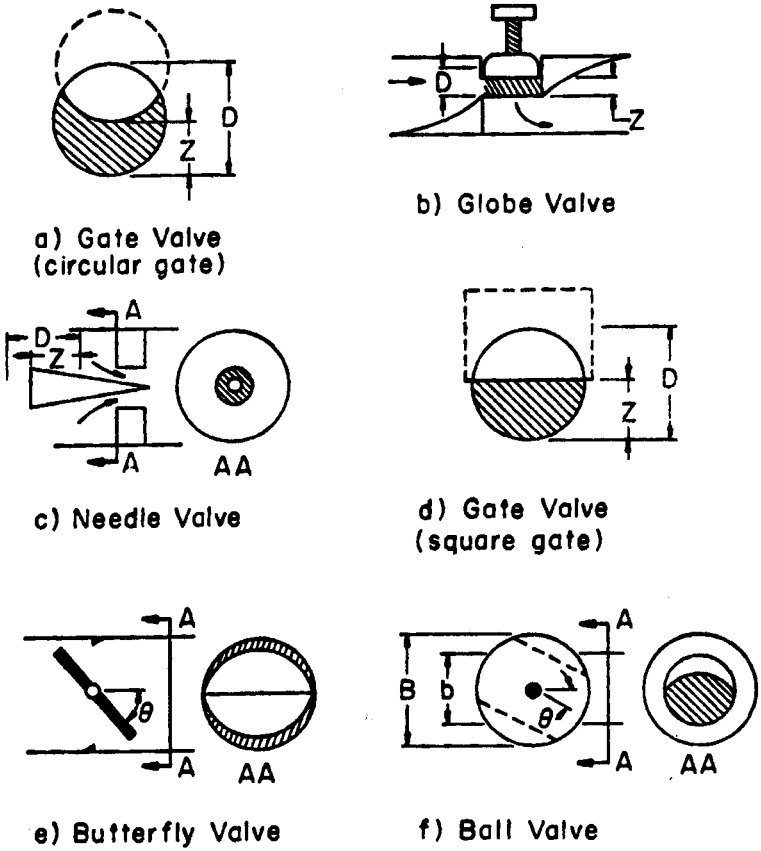


FIGURE 2 Cross sections of selected control valves (after Wood and Jones, Reference 2)

typically relate the volumetric flow rate to a characteristic valve area and the head loss ΔH across the valve. The principal fluid properties that can affect the flow characteristics are fluid density ρ, fluid viscosity μ, and liquid vapor pressure p_v if cavitation occurs. Except for very small valves and/or quite viscous liquids, Reynolds number effects are usually not important, and will be neglected with reference to water. A valve in a pipeline acts as an obstruction, disturbs the flow, and in general causes a loss in energy as well as affecting the pressure distribution both upstream and downstream. The characteristics are expressed either in terms of (a) flow capacity as a function of a defined pressure drop or (b) energy dissipation (head loss) as a function of pipe velocity. In both instances the pressure or head drop is usually the difference in total head caused by the presence of the valve itself, minus any loss due to regular pipe friction between measuring stations.

The proper manner in determining ΔH experimentally is to measure the hydraulic grade line (HGL) far enough both upstream and downstream of the valve so that uniform flow sections to the left of and to the right of the valve can be established, allowing for the extrapolation of the energy grade lines (EGL) to the plane of the valve. Otherwise, the valve head loss is not properly defined. It is common to express the hydraulic characteristics either in terms of a head loss coefficient K_L or as a discharge coefficient where A_v is the area of the valve at any opening, and ΔH is the head loss defined for the valve. Frequently a discharge coefficient is defined in terms of the fully-open valve area. The hydraulic coefficients embody not only the geometric features of the valve through A_v but also the flow characteristics.

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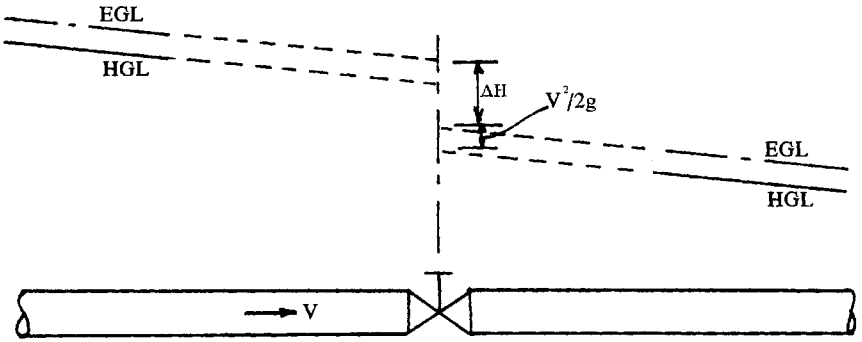


FIGURE 3 Definition of head loss characteristics of a valve

Unless uniform flow is established far upstream and downstream of a valve in a pipeline the value of any of the coefficients can be affected by effects of nonuniform flow. It is not unusual for investigators to use only two pressure taps—one upstream and one downstream, frequently one and ten diameters, respectively. The flow characteristics of valves in terms of pressure drop or head loss have been determined for numerous valves by many investigators and countless manufacturers. Only a few sets of data and typical curves will be presented here for ball, butterfly, gate, and check valves. For a valve located in the interior of a long continuous pipe, as shown in Figure 3, the presence of the valve disturbs the flow both upstream and downstream of the obstruction as reflected by the velocity distribution, and the pressure variation, which will be nonhydrostatic in the regions of nonuniform flow. Accounting for the pipe friction between upstream and downstream uniform flow sections, the head loss across the valve is expressed in terms of the pipe velocity and a head-loss coefficient K_L.

$$\Delta H = K_L \frac{V^2}{2g} \tag{4}$$

Often manufacturers represent the hydraulic characteristics in terms of discharge coefficients.

$$Q = C_f A_{vo} \sqrt{2g\,\Delta H} = C_Q A_{vo} \sqrt{2gH} \tag{5}$$

where

$$H = \Delta H + \frac{V^2}{2g} \tag{6}$$

Both discharge coefficients are defined in terms of the nominal full-open valve area A_{vo} and a representative head, ΔH for C_f and H for C_Q, the latter definition generally reserved for large valves employed in the hydroelectric industry. The interrelationship between C_f, C_Q, and K_L is

$$K_L = \frac{1}{C_f^2} = \frac{1 - C_Q^2}{C_Q^2} \tag{7}$$

Frequently valve characteristics are expressed in terms of a dimensional flow coefficient C_v from the valve industry.

$$Q = C_v \sqrt{\Delta p} \tag{8}$$

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where Q is in US flow units of gpm and Δp is the pressure loss in psi. In transient analysis it is convenient to relate either the loss coefficient or the discharge coefficient to the corresponding value at the fully open valve position, for which the $C_f = C_{fo}$. Hence,

$$\frac{Q}{Q_o} = \frac{C_f}{C_{fo}}\sqrt{\frac{\Delta H}{\Delta H_o}} = \tau\sqrt{\frac{\Delta H}{\Delta H_o}} \tag{9}$$

Traditionally the dimensionless valve discharge coefficient is termed τ and defined by

$$\tau = \frac{C_f}{C_{fo}} = \frac{C_v}{C_{vo}} = \frac{C_Q}{C_{Qo}} = \sqrt{\frac{K_{Lo}}{K_L}} \tag{10}$$

Typical Geometric and Hydraulic Valve Characteristics The geometric projected area of valves shown in Figure 2 can be calculated for ball, butterfly, and gate valves using simple expressions. The dimensionless hydraulic flow coefficient τ is plotted in Figure 4 for various valve openings for the three selected valves along with the area ratio for comparison. The lower diagram, which is based on hydraulic measurements, should be used for transients calculations rather than the upper one, which is strictly geometric.

Valve Operation The instantaneous closure of a valve at the end of a pipe will yield a pressure rise satisfying Joukowsky's equation—Eq. 2 or Eq. 3. In this case the velocity difference $\Delta V = 0 - V_o$, where V_o is the initial velocity of liquid in the pipe. Although Eq. 2 applies across every wavelet, the effect of complete valve closure over a period of time greater than $2L/a$, where L is the distance along the pipe from the point of wave creation to the location of the first pipe area change, can be quite beneficial. Actually, for a simple pipeline the maximum head rise remains that from Eq. 3 for times of valve closure $t_c = 2L/a$, where L is the length of pipe. If the value of $t_c > 2L/a$, then there can be a considerable reduction of the peak pressure resulting from beneficial effects of negative wave reflections from the open end or reservoir considered in the analysis. The phenomenon can still be classified as waterhammer until the time of closure $t_c \gg 2L/a$, beyond which time there are only inertial or incompressible deceleration effects, referred to as surging, also known as rigid column analysis. Table 2 is furnished to classify four types of valve closure, independent of type of valve.

Using standard waterhammer programs, parametric analyses can be conducted for the preparation of charts to demonstrate the effect of time of closure, type of valve, and an indication of the physical process—waterhammer or simply inertia effects of deceleration. The charts are based on analysis of valve closure for a simple reservoir-pipe-valve arrangement. For simplicity fluid friction is often neglected, a reasonable assumption for pipes on the order of hundreds of feet in length.

HYDRAULIC CHARACTERISTICS OF PUMPS

Transient analyses of piping systems involving centrifugal, mixed-flow, and axial-flow pumps require detailed information describing the characteristics of the respective turbomachine, which may pass through unusual, indeed abnormal, flow regimes. Since little if any information is available regarding the dynamic behavior of the pump in question, invariably the decision must be made to use the steady-flow characteristics of the machine gathered from laboratory tests. Moreover, complete steady-flow characteristics of the machine may not be available for all possible modes of operation that may be encountered in practice.

In this section steady-flow characteristics of pumps in all possible zones of operation are defined. The importance of geometric and dynamic similitude is first discussed with respect to both (1) homologous relationships for steady flow and (2) the importance of the assumption of similarity for transient analysis. The significance of the eight zones of operation

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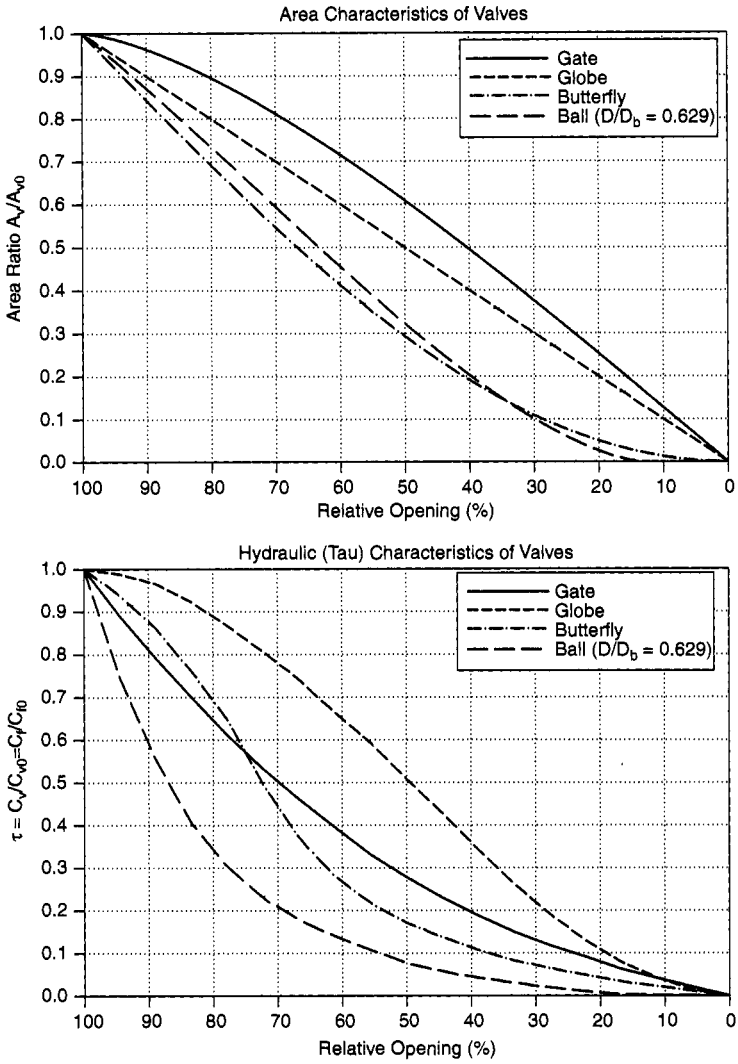


FIGURE 4 Geometric and hydraulic characteristics of typical control valves

TABLE 2 Classification of valve closure

Time of closure t_c	Type of closure	Maximum head ΔH_{max}	Phenomenon
0	Instantaneous	aV_o/g	Waterhammer
$= 2L/a$	Rapid	aV_o/g	Waterhammer
$> 2L/a$	Gradual	$< aV_o/g$	Waterhammer
$\gg 2L/a$	Slow	$\ll aV_o/g$	Surging

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within each of the four quadrants is presented in detail with reference to three possible modes of data representation. The steady-flow characteristics of pumps are discussed in detail with regard to the complete range of possible operation. The loss of driving power to a pump is usually the most critical transient case to consider for pumps, because of the possibility of low pipeline pressures which may lead to (1) pipe collapse due to buckling, or (2) the formation of a vapor cavity and its subsequent collapse. Other waterhammer problems may occur due to slam of a swing check valve, or from a discharge valve closing either too quickly (column separation), or too slowly (surging from reverse flow). For radial-flow pumps for which the reverse flow reaches a maximum just subsequent to passing through zero speed (locked rotor point), and then is decelerated as the shaft runs faster in the turbine zone, the head will usually rise above the nominal operating value. As reported by Donsky, mixed-flow and axial-flow pumps may not even experience an upsurge in the turbine zone because the maximum flow tends to occur closer to runaway conditions.[3]

Definition of Pump Characteristics The essential parameters for definition of hydraulic performance of pumps are defined as

Impeller diameter. Exit diameter of pump rotor D_I

Rotational speed. The angular velocity (rad/sec) is ω, while $N = 2\pi\omega/60$ is in rpm

Flow rate. Capacity Q at operating point in chosen units

Total dynamic head (TDH). The total energy gain (or loss) H across pump, defined as

$$H = \left(\frac{p_d}{\gamma} + z_d\right) - \left(\frac{p_s}{\gamma} + z_s\right) + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} \tag{11}$$

where subscripts s and d refer to suction and discharge sides of the pump, respectively.

Homologous (Affinity) Laws Dynamic similitude, or dimensionless representation of test results, has been applied with perhaps more success in the area of hydraulic machinery than in any other field involving fluid mechanics. Due to the sheer magnitude of the problem of data handling it is imperative that dimensionless parameters be employed for transient analysis of hydraulic machines that are continually experiencing changes in speed as well as passing through several zones of normal and abnormal operation. For liquids for which thermal effects may be neglected, the remaining fluid-related forces are pressure (head), fluid inertia, resistance, phase change (cavitation), surface tension, compressibility, and gravity. If the discussion is limited to single-phase liquid flow, three of the above fluid effects—cavitation, surface tension, and gravity (no interfaces within machine)—can be eliminated, leaving the forces of pressure, inertia, viscous resistance, and compressibility. For the steady or even transient behavior of hydraulic machinery conducting liquids the effect of compressibility may be neglected.

In terms of dimensionless ratios the three forces yield an Euler number (ratio of inertia force to pressure force), which is dependent upon geometry, and a Reynolds number. For all flowing situations the viscous force, as represented by the Reynolds number, is definitely present. If water is the fluid medium, the effect of the Reynolds number on the characteristics of hydraulic machinery can usually be neglected, with the major exception being the prediction of the performance of a large hydraulic turbine on the basis of model data. For the transient behavior of a given machine the actual change in the value of the Reynolds number is usually inconsequential anyway. The elimination of the viscous force from the original list reduces the number of fluid-type forces from seven to two—pressure (head) and inertia, as exemplified by the Euler number. The appellation geometry in the functional relationship in the above equation embodies primarily the shape of the rotating impeller; the entrance and exit flow passages, including effects of vanes, diffusers, and so on; secondly the effect of surface roughness; and lastly the geometry of the streamline pattern, better known as kinematic similitude in contrast to the first two, which are related to geometric similarity. Kinematic similarity is invoked on the assumption that similar flow patterns can be specified by congruent velocity triangles composed of peripheral speed U and absolute fluid velocity V at inlet or exit to the vanes.

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This allows for the definition of a flow coefficient, expressed in terms of impeller diameter D_I and angular velocity ω

$$C_Q = \frac{Q}{\omega D_I^3} \tag{12}$$

The reciprocal of the Euler number (ratio of pressure force to inertia force) is the head coefficient, defined as

$$C_H = \frac{gH}{\omega^2 D_I^2} \tag{13}$$

For power requirements a power coefficient can be defined

$$C_P = \frac{P}{\rho \omega^3 D_I^5} \tag{14}$$

For transient analysis, the desired parameter for the continuous prediction of pump speed ω is the unbalance torque T. Since $T = P/\omega$, the torque coefficient becomes

$$C_T = \frac{T}{\rho \omega^2 D_I^5} \tag{15}$$

It is traditional in hydraulic transient analysis to refer pump characteristics to so-called *rated conditions,* which preferably should be the optimum or best efficiency point (BEP), but sometimes defined as the duty, nameplate, or design point. Nevertheless, in terms of *rated conditions*, for which the subscript R is employed, the following ratios are defined

$$\textit{Flow: } v = \frac{Q}{Q_R} \qquad \textit{Speed: } \alpha = \frac{\omega}{\omega_R} = \frac{N}{N_R} \qquad \textit{Head: } h = \frac{H}{H_R} \qquad \textit{Torque: } \beta = \frac{T}{T_R}$$

Next, for a given pump undergoing a transient, for which D_I is a constant, Eqs. 12 to 15 can be written in terms of the above ratios:

$$\frac{v}{\alpha} = \frac{C_Q}{C_{QR}} = \frac{Q_R}{Q}\frac{\omega}{\omega_R} \qquad \frac{h}{\alpha^2} = \frac{C_H}{C_{HR}} = \frac{H}{H_R}\frac{\omega_R^2}{\omega^2} \qquad \frac{\beta}{\alpha^2} = \frac{C_T}{C_{TR}} = \frac{T}{T_R}\frac{\omega_R^2}{\omega^2}$$

Abnormal Pump (Four Quadrant) Characteristics The performance characteristics discussed up to this point correspond to pumps operating normally. During a transient, however, the machine may experience either a reversal in flow, or rotational speed, or both, depending upon the situation. It is also possible that the torque and head may reverse in sign during passage of the machine through abnormal zones of performance. The need for characteristics of a pump in abnormal zones of operation can best be described with reference to Figure 5, which is a simulated pump power-failure transient. A centrifugal pump is delivering water at a constant rate when there is a sudden loss of power from the prime mover—in this case an electric motor. For the postulated case of no discharge valves, or other means of controlling the flow, the loss of driving torque leads to an immediate deceleration of the shaft speed, and in turn the flow. The three curves are dimensionless head (h), flow (v), and speed (α), shown in Figure 5 respectively as percentages H, Q, and N. With no additional means of controlling the flow, the higher head at the final delivery point (another reservoir) will eventually cause the flow to reverse ($v < 0$) while the inertia of the rotating parts has maintained positive rotation ($\alpha > 0$). Up until the time of flow reversal the pump has been operating in the normal zone, albeit at a number of off-peak flows.

In order to predict system performance in regions of negative rotation and/or negative flow the analyst requires characteristics in these regions for the machine in question. Indeed, any peculiar characteristic of the pump in these regions could be expected to have

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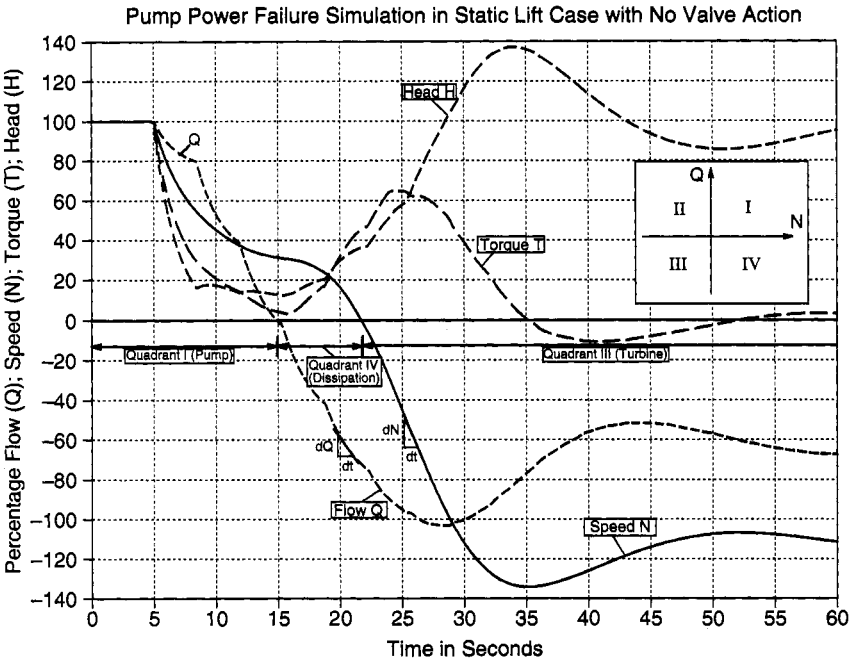


FIGURE 5 Simulated pump trip without valves in a single pipeline system

an influence on the hydraulic transients. It is important to stress that the results of such analyses are critically governed by the following three factors: (1) availability of complete pump characteristics in zones the pump will operate, (2) complete reliance upon dynamic similitude (homologous) laws during transients, and (3) assumption that steady-flow derived pump characteristics are valid for transient analysis.

Investigations by Knapp[4] and Kittredge[5] facilitated the understanding of abnormal operation, as well as served to reinforce the need for test data. (Kittredge's summary of that work appears in Section 2.1.3 of this book.) Following the work by Knapp[6] and Swanson,[7] and a summary of their results by Donsky,[3] eight possible zones of operation, four normal and four abnormal, will be discussed here with reference to Figure 6, developed by Martin.[8] In Figure 6 the head H is shown as the difference in the two reservoir elevations to simplify the illustration. The effect of pipe friction may be ignored for this discussion by assuming that the pipe is short and of relatively large diameter. The regions referred to on Figure 6 are termed Zones and Quadrants, the latter definition originating from plots of lines of constant head and constant torque on a flow-speed plane ($v - \alpha$ axes). Quadrants I ($v > 0$, $\alpha > 0$) and III ($v < 0$, $\alpha < 0$) are defined in general as regions of pump or turbine operation, respectively. It will be seen, however, that abnormal operation (neither pump nor turbine mode) may occur in either of these two quadrants. A very detailed description of each of the eight zones of operation is in order. It should be noted that all of the conditions shown schematically in Figure 6 can be contrived in a laboratory test loop using an additional pump (or two) as the master and the test pump as a slave. Most, if not all, of the zones shown can also be experienced by a pump during a transient under the appropriate set of circumstances.

QUADRANT I Zone A (normal pumping) in Figure 6 depicts a pump under normal operation for which all four quantities, Q, N, H, and T, are regarded as positive. In this case $Q > 0$, indicating useful application of energy. Zone B (energy dissipation) is a condition of positive flow, positive rotation, and positive torque, but negative head—quite an abnormal

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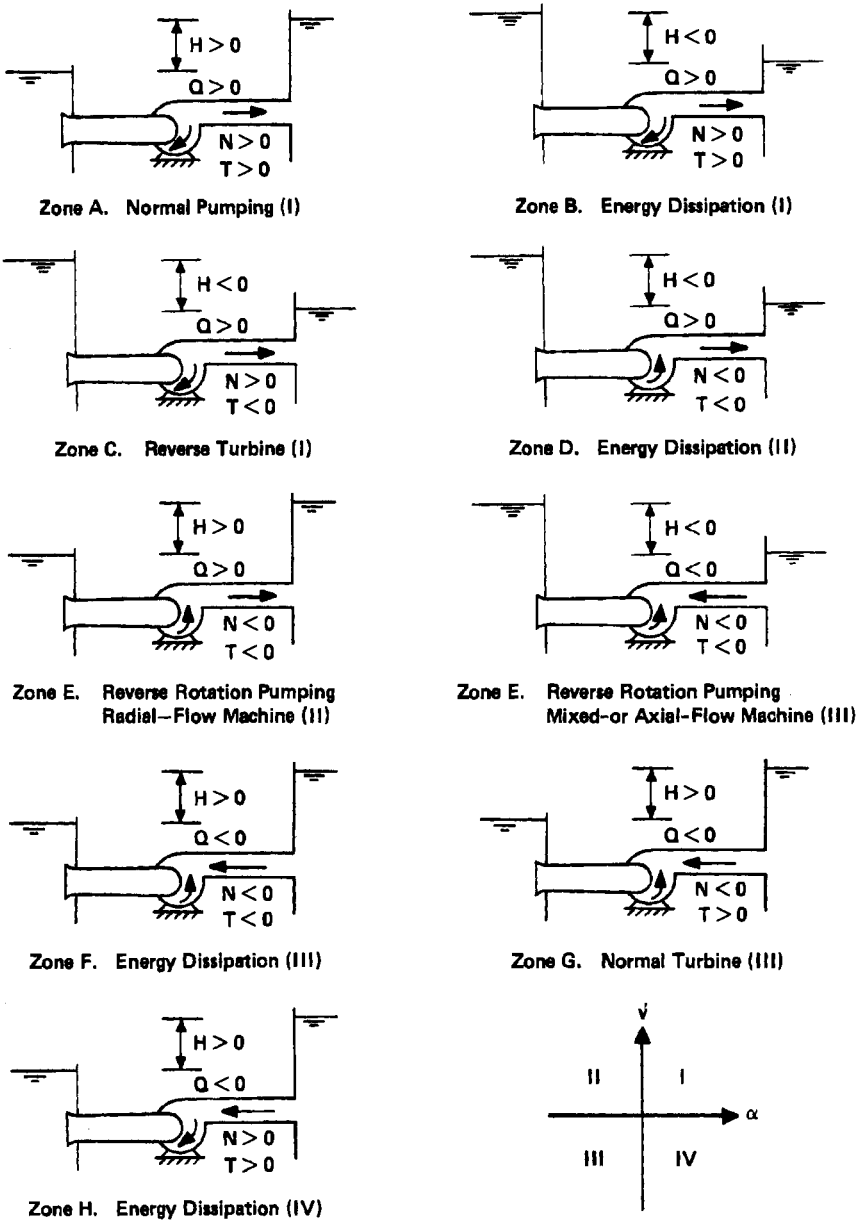


FIGURE 6 Four quadrants and eight zones of possible pump operation (after Martin, Reference 8)

condition. A machine could operate in Zone B by (1) being overpowered by another pump or by a reservoir during steady operation, or (2) by a sudden drop in head during a transient caused by power failure. It is possible, but not desirable, for a pump to generate power with both the flow and rotation in the normal positive direction for a pump, Zone C (reverse turbine), caused by a negative head, and resulting in a positive efficiency because

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of the negative torque. The maximum efficiency would be quite low due to the bad entrance flow condition and unusual exit velocity triangle.

QUADRANT IV Zone H, labeled energy dissipation, is often encountered shortly after a trip out or power failure of a pump, as illustrated in Figure 5. In this instance the combined inertia of all the rotating elements—motor, pump and its entrained liquid, and shaft—has maintained pump rotation positive but at a reduced value at the time of flow reversal caused by the positive head on the machine. This purely dissipative mode results in a negative or zero efficiency. It is important to note that both the head and fluid torque are positive in Zone H, the only zone in Quadrant IV.

QUADRANT III A machine that passes through Zone H during a pump power failure will then enter Zone G (normal turbining) provided that reverse shaft rotation is not precluded by a mechanical ratchet. Although a runaway machine rotating freely is not generating power, Zone G is the precise mode of operation for a hydraulic turbine. Note that the head and torque are positive, as for a pump but that the flow and speed are negative, opposite to that for a pump under normal operation (Zone A).

Subsequent to the trip out or load rejection of a hydraulic turbine or the continual operation of a machine that failed earlier as a pump, Zone F (energy dissipation) can be encountered. The difference between Zones F and G is that the torque has changed sign for Zone F, resulting in a braking effect, which tends to slow the freewheeling machine down. In fact, the real runaway condition is attained at the boundary of the two zones, for which torque $T = 0$.

QUADRANT II The two remaining Zones, D and E, are very unusual and infrequently encountered in operation, with the exception of pump/turbines entering Zone E during transient operation. Again it should be emphasized that both zones can be experienced by a pump in a test loop, or in practice in the event a machine is inadvertently rotated in the wrong direction by improper wiring of an electric motor. Zone D is a purely dissipative mode that normally would not occur in practice unless a pump, which was designed to increase the flow from a higher to lower reservoir, was rotated in reverse, but did not have the capacity to reverse the flow (Zone E, mixed or axial flow), resulting in $Q > 0, N < 0, T < 0$, for $H < 0$. Zone E, for which the pump efficiency > 0, could occur in practice under steady flow if the preferred rotation as a pump was reversed. There is always the question regarding the eventual direction of the flow. A radial-flow machine will produce positive flow at a much reduced capacity and efficiency compared to $N > 0$ (normal pumping), yielding of course $H > 0$. On the other hand, mixed and axial-flow machines create flow in the opposite direction (Quadrant III), and $H < 0$, which corresponds still to an increase in head across the machine in the direction of flow.

Representation of Pump Data for Numerical Analysis It is conventional in transient analyses to represent h/α^2 and β/α^2 as functions of v/α, as shown in Figures 7 and 8 for a radial-flow pump. The curves on Figure 7 are only for positive rotation ($\alpha > 0$), and constitute pump Zones A, B, and C for $v > 0$ and the region of energy dissipation subsequent to pump power failure (Zone H), for which $v < 0$. The remainder of the pump characteristics are plotted in Figure 8 for $\alpha < 0$. The complete characteristics of the pump plotted in Figures 7 and 8 can also be correlated on what is known as a Karman-Knapp circle diagram, a plot of lines of constant head (h) and torque (β) on the coordinates of dimensionless flow (v) and speed (α). Figure 9 is such a correlation for the same pump. The complete characteristics of the pump require six curves, three each for head and torque. For example, the h/α^2 curves from Figures 7 and 8 can be represented by continuous lines for $h = 1$ and $h = -1$, and two straight lines through the origin for $h = 0$. A similar pattern exists for the torque (β) lines. In addition to the eight Zones A-H illustrated in Figure 6, the four Karman-Knapp quadrants in terms of v and α, are well defined. Radial lines on Figure 9 correspond to constant values for v/α in Figures 7 and 8, allowing for relatively easy transformation from one form of presentation to the other.

In computer analysis of pump transients, Figures 7 and 8, while meaningful from the standpoint of physical understanding, are fraught with the difficulty of $|v/\alpha|$ becoming

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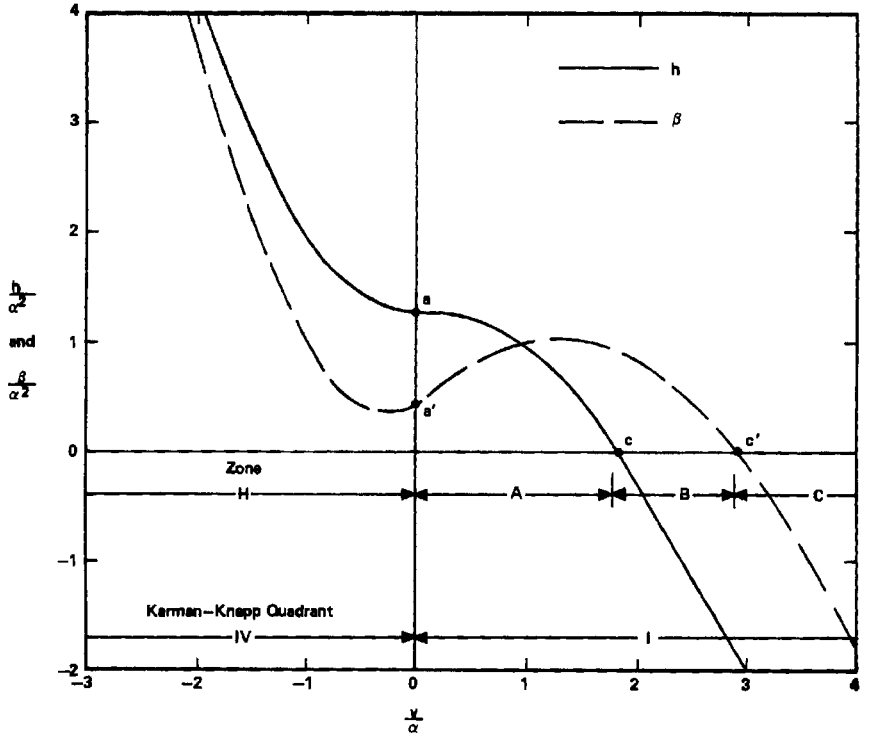
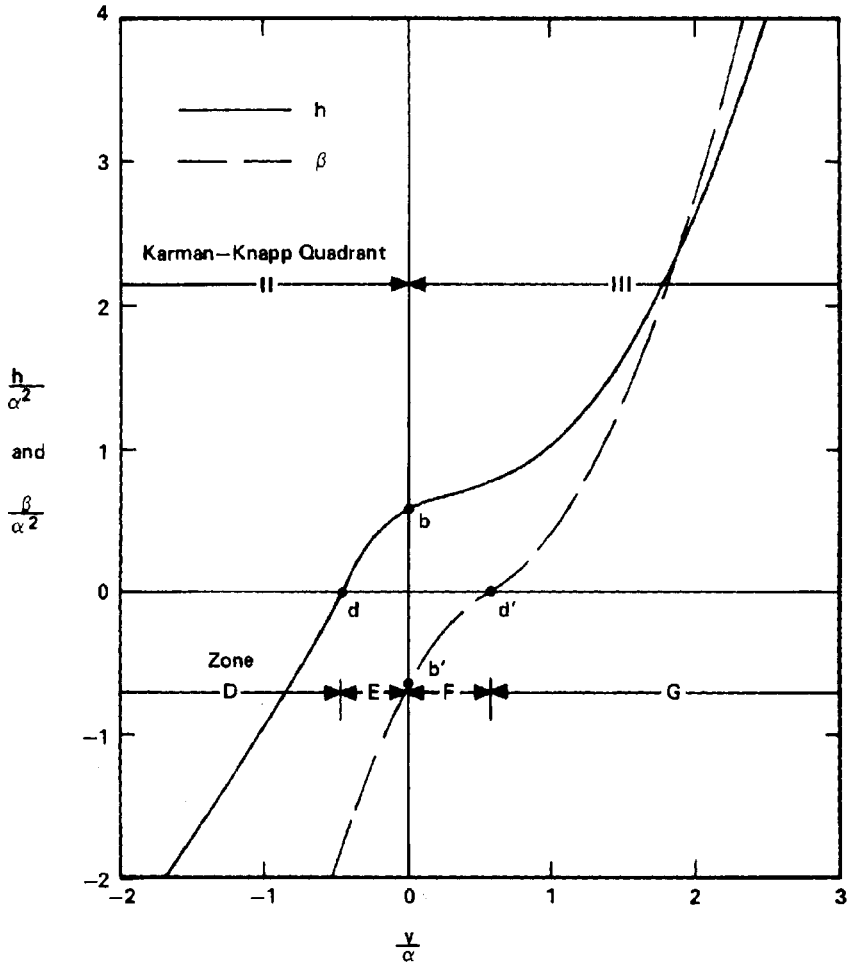


FIGURE 7 Complete head and torque characteristics of radial-flow pump for positive rotation (after Martin, Reference 8)

infinite as the unit passes through, or remains at, zero speed ($\alpha = 0$). Some have solved that problem by switching from h/α^2 versus v/α to h/v^2 versus α/v, and likewise for β, for $|v/\alpha| > 1$. This technique doubles the number of curves on Figures 7 and 8, and thereby creates discontinuities in the slopes of the lines at $|v/\alpha| = 1$, in addition to complicating the storing and interpolation of data. Marchal, Flesh, and Suter devised a useful transformation that allowed the complete pump characteristics to be represented by two single curves, as shown for the same pump in Figure 10.[9] The difficulty of v/α becoming infinite was eliminated by utilizing the function $\tan^{-1}(v/\alpha)$ as the abscissa. The eight zones, or four quadrants can then be connected by the continuous functions. Although some of the physical interpretation of pump data has been lost in the transformation, Figure 10 is now a preferred correlation for transient analysis using a digital computer because of function continuity and ease of numerical interpolation. The singularities in Figures 7 and 8 and the asymptotes in Figure 9 have now been avoided.

Critical Data Required for Hydraulic Analysis of Systems with Pumps Regarding data from manufacturers such as pump curves (normal and abnormal), pump and motor inertia, motor torque-speed curves, and valve curves, probably the most critical for pumping stations are pump-motor inertia and valve closure time. Normal pump curves are usually available and adequate. A few examples of abnormal pump curves have been given in this section, in Section 2.1.3 and in Stepanoff[10] for radial- and axial-flow pumps; however, such information describes only how those particular pumps behave in such

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Homologous Head and Torque Characteristics for Radial–Flow Pump
(Ω_s = 0.465 in Universal Units) for Negative Rotation ($\alpha < o$)

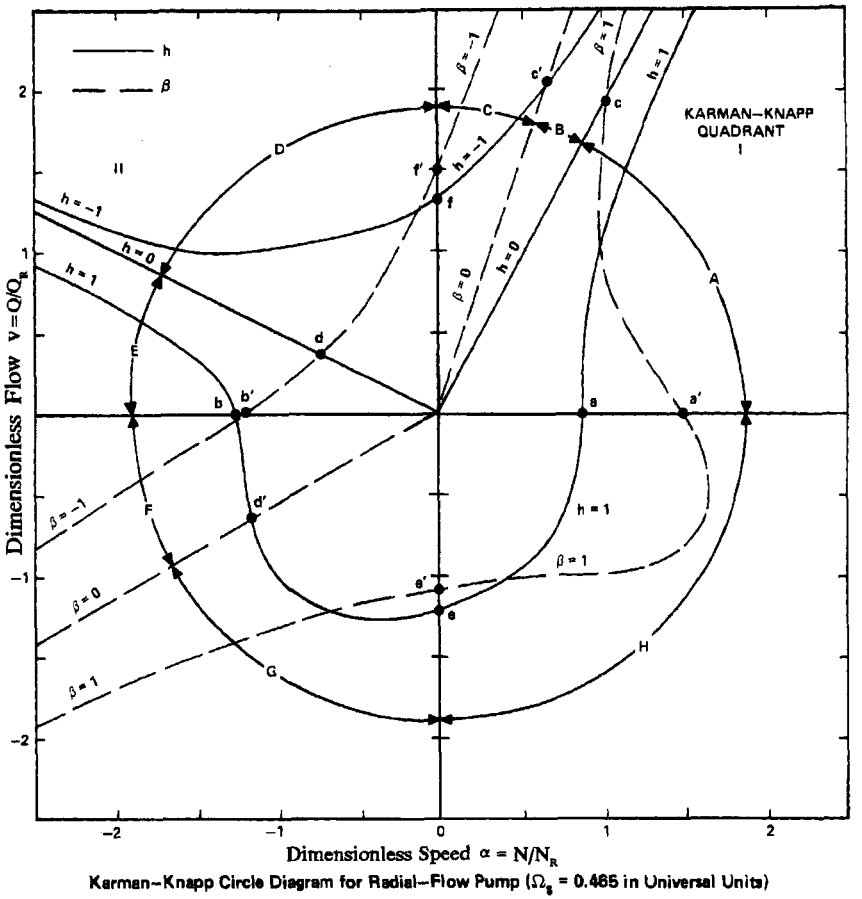
FIGURE 8 Complete head and torque characteristics of radial-flow pump for negative rotation (after Martin, Reference 8)

circumstances. For any other pump, actual data of this kind for the machine in question is essential for reliable hydraulic analysis of a pump system. Motor torque-speed curves are only needed when evaluating pump start-up. For pump trip the inertia of the combined pump and motor is quite important.

SURGE PROTECTION AND SURGE CONTROL DEVICES

There are numerous techniques for controlling transients and waterhammer, some involving design considerations and others the consideration of surge protection devices. There

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Karman–Knapp Circle Diagram for Radial–Flow Pump (Ω_R = 0.485 in Universal Units)

FIGURE 9 Complete four-quadrant head and torque characteristics of radial-flow pump (after Martin, Reference 8)

must be a complete design and operational strategy devised to combat potential waterhammer in a system. The transient event may either initiate a low-pressure event (downsurge) as in the case of a pump power failure, or a high-pressure event (upsurge) caused by the closure of a downstream valve. It is well known that a downsurge can lead to the undesirable occurrence of water column separation, which itself can result in severe pressure rises following the collapse of a vapor cavity. In some systems negative pressures are not even allowed because of (1) possible pipe collapse or (2) ingress of outside water or air.

The means of controlling the transient will in general vary, depending upon whether the initiating event results in an upsurge or downsurge. For pumping plants the major cause of unwanted transients is typically the complete outage of pumps due to loss of electricity to the motor. For full pipelines pump start-up, usually against a closed pump discharge valve for centrifugal pumps, does not normally result in significant pressure transients. Indeed, the majority of transient problems in pumping installations are associated with the potential (or realized) occurrence of water column separation and vapor-pocket collapse, resulting from the trip out of one or more pumps, with or without valve action. The pump-discharge valve, if actuated too suddenly, can even aggravate the downsurge problem. To combat the downsurge problem there are a number of options, mostly

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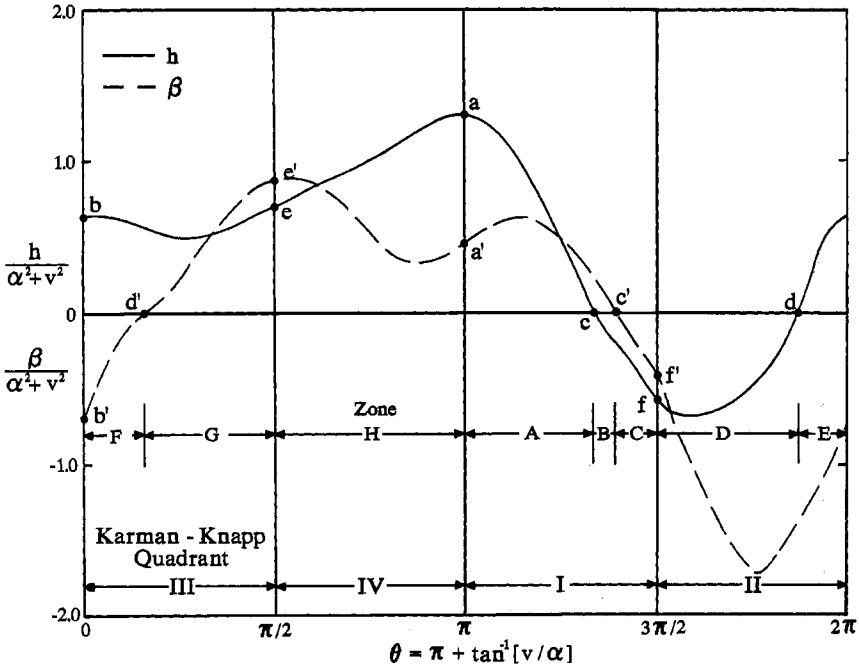


FIGURE 10 Complete head and torque characteristics of radial-flow pump in Suter diagram (after Martin, Reference 8)

involving the design and installation of one or more surge protection devices. In this section various surge protection techniques will be discussed, followed by an assessment of the virtue of each with respect to pumping systems in general.

Critical Parameters for Transients Before discussing surge protection devices, some comments will be made regarding the various pipeline, pump and motor, control valve, flow rate and other parameters that affect the magnitude of the transient. For a pumping system the four main parameters are (1) pump flow rate, (2) pump and motor WR^2, (3) any valve motion, and (4) pipeline characteristics. The pipeline characteristics include piping layout—both plan and profile—pipe size and material, and the acoustic velocity. So-called short systems respond differently than long systems. Likewise, valve motion and its effect, whether controlled valves or check valves, will have different effects on the two types of systems. In this regard both the Main Zone and High Zone are long systems.

The pipeline characteristics—item number (4)—relate to the response of the system to a transient such as pump power failure. Clearly, the response will be altered by the addition of one or more surge protection device or the change of (1) the flow rate, or (2) the WR^2, or (3) the valve motion. Obviously, for a given pipe network and flow distribution there are limited means of controlling transients by (2) WR^2 and (3) valve actuation. If these two parameters cannot alleviate the problem then the pipeline response needs to be altered by means of surge protection devices.

Critique of Surge Protection For pumping systems, downsurge problems have been solved by various combinations of the procedures and devices mentioned above. In many instances local conditions and preferences of engineers have dictated the choice of methods and/or devices. On-line devices such as accumulators and simple surge tanks are quite effective, albeit expensive, solutions. Examples of the various types and deployment of these devices are shown in Figures 11 and 12. One-way surge tanks can also be effective

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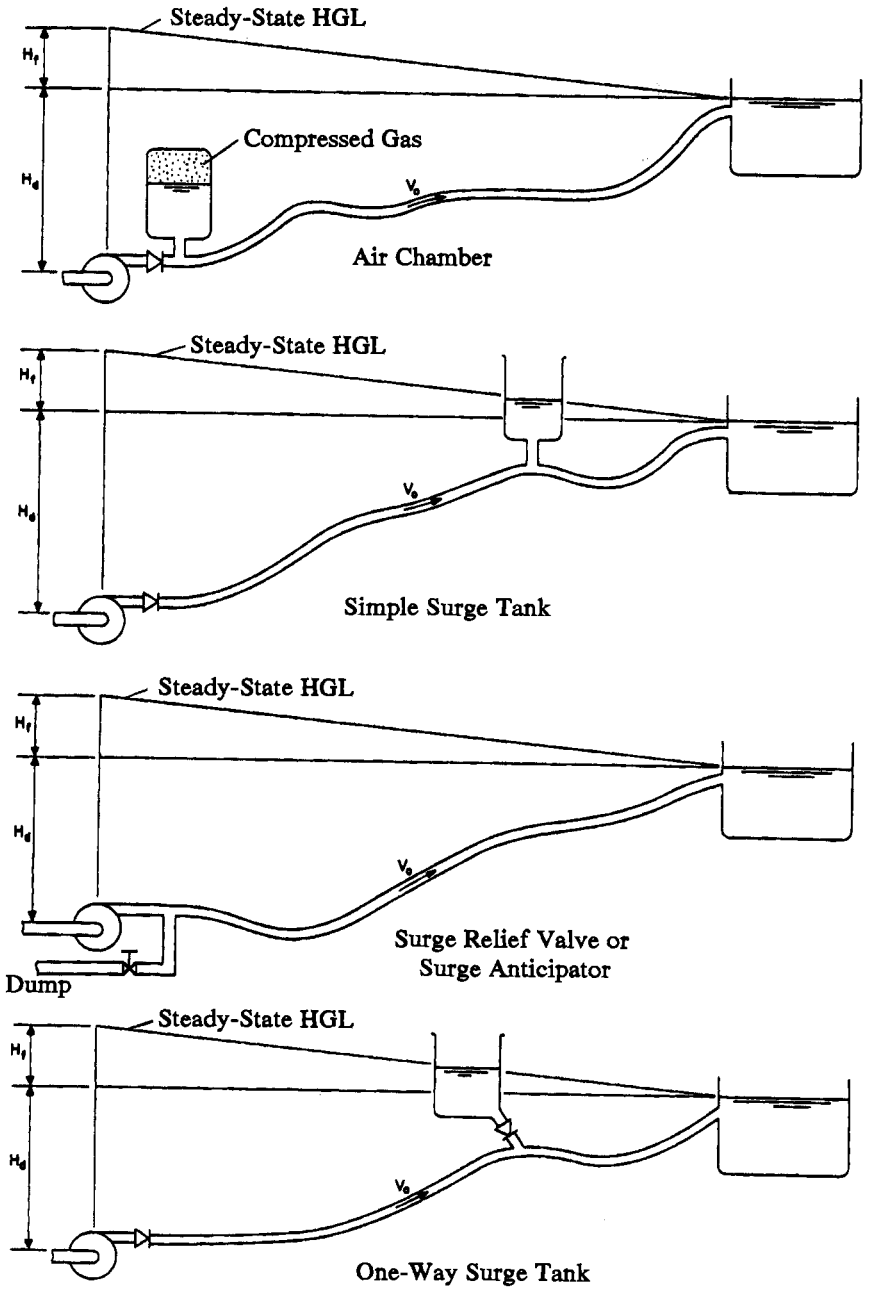


FIGURE 11 Schematic of various surge protection devices for pumping installations

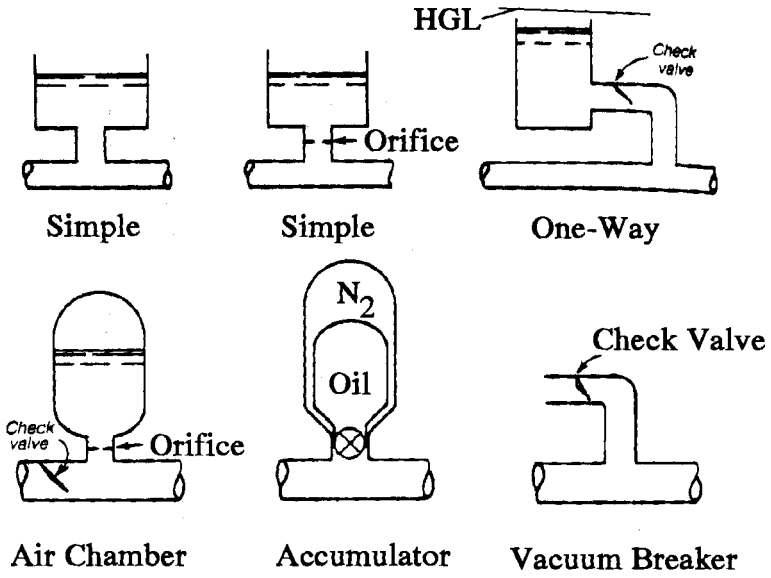


FIGURE 12 Cross-sectional view of surge tanks and gas-related surge protection devices

when judiciously sized and sited. Surge anticipation valves should not be used when there is already a negative pressure problem. Indeed, there are installations where surge anticipation functions of such valves have been deactivated, leaving only the surge relief feature. Moreover, there have been occasions for which the surge anticipation feature aggravated the low-pressure situation by an additional downsurge caused by premature opening of the valve.

Regarding the consideration and ultimate choice of surge protection devices, subsequent to calibration of analysis with test results, evaluation should be given to simple surge tanks or standpipes, one-way surge tanks, and hydropneumatic tanks or air chambers. A combination of devices may prove to be the most desirable and most economical.

The admittance of air into a piping system can be effective, but the design of air-vacuum-valve location and size is critical. If air may be permitted into pipelines careful analysis would have to be done to ensure effective results. The consideration of air-vacuum breakers is a moot point if specifications such as the Ten State Standards limit the pressures to 20 psig.

Surge Protection Control and Devices Pump discharge valve operation. In gravity systems the upsurge transient can be controlled by an optimum valve closure—perhaps two stage, as mentioned by Wylie and Streeter.[11] As shown by Fleming, an optimized closing can solve a waterhammer problem caused by pump power failure if coupled with the selection of a surge protection device.[12] For pump power failure a control valve on the pump discharge can often be of only limited value in controlling the downsurge, as mentioned by Sanks.[13] Indeed, the valve closure can be too sudden, aggravating the downsurge and potentially causing column separation, or too slow, allowing a substantial reverse flow through the pump. It should also be emphasized that an optimum controlled motion for single-pump power failure is most likely not optimum for multiple-pump failure. The use of microprocessors and servomechanisms with feedback systems can be a general solution to optimum control of valves in conjunction with the pump and pipe system. For pump discharge valves the closure should not be too quick to exacerbate downsurge, nor too slow to create a substantial flow back through the valve and pump before closure.

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CHECK VALVES Swing check valves or other designs are frequently employed in pump discharge lines, often in conjunction with slow acting control valves. As indicated by Tullis, a check valve should open easily, have a low head loss for normal positive flow, and create no undesirable transients by its own action.[14] For short systems, a slow responding check valve can lead to waterhammer because of the high reverse flow generated before closure. A spring- or counterweight-loaded valve with a dashpot can (1) give the initial fast response followed by (2) slow closure to alleviate the unwanted transient. The proper selection of the load and the degree of damping is important, however, for proper performance.

Check valve slam is also a possibility from stoppage or failure of one pump of several in a parallel system, or resulting from the action of an air chamber close to a pump undergoing power failure. Check valve slam can be reduced by the proper selection of a dashpot.

SURGE ANTICIPATOR VALVES AND SURGE RELIEF VALVES A surge anticipation valve (Figure 11), frequently installed at the manifold of the pump station, is designed to open initially under (1) pump power failure, or (2) the sensing of underpressure, or (3) the sensing of overpressure, as described by Lescovitch.[15] On the other hand, the usual type of surge relief valve opens quickly upon sensing an overpressure, then closes slowly, as controlled by pilot valves. The surge anticipation valve is more complicated than a surge relief valve in that it not only embodies the relief function at the end of the cycle, but also has the element of anticipation. For systems for which water column separation will not occur, the surge anticipation valve can solve the problem of upsurge at the pump due to reverse flow or wave reflection, as reported in an example by White.[16] An example of a surge relief valve only is provided by Weaver.[17] For systems for which water column separation will not occur, Lundgren provides charts for simple pipeline systems.[18]

As reported by Parmakian surge anticipation valves can exacerbate the downsurge problem inasmuch as the opening of the relief valve aggravates the negative pressure problem.[19,20,21] Incidents have occurred involving the malfunctioning of a surge anticipation valve, leading to extreme pressures because the relief valve did not open.

PUMP BYPASS In shorter low-head systems a pump bypass line can be installed in order to allow water to be drawn into the pump discharge line following power failure and a downsurge. As explained by Wylie and Streeter, there are two possible bypass configurations.[11] The first involves a control valve on the discharge line and a check valve on the bypass line between the pump suction or wet well and the main line. The check valve is designed to open subsequent to the downsurge, possibly alleviating column separation down the main line. The second geometry would reverse the valve locations, having a control valve in the bypass and a check valve in the main line downstream of the pump. The control would open on power failure, again allowing water to bypass the pump into the main line.

OPEN (SIMPLE) SURGE TANK A simple on-line surge tank or standpipe (Figures 11 and 12) can be an excellent solution to both upsurge and downsurge problems. These devices are quite common in hydroelectric systems where suitable topography usually exists. They are practically maintenance-free, available for immediate response as they are on line. For pumping installations open simple surge tanks are rare because of height considerations and the absence of high points near most pumping stations. As mentioned by Parmakian, simple surge tanks are the most dependable of all surge protection devices.[19] One disadvantage is the additional height to allow for pump shutoff head. Overflowing and spilling must be considered, as well as the inclusion of some damping to reduce oscillations. As stated by Kroon *et al.* (1984) the major drawback to simple surge tanks is their capital expense.[22]

ONE-WAY SURGE TANK The purpose of a one-way surge tank (Figures 11 and 12) is to prevent initial low pressures and potential water column separation by admitting water into the pipeline subsequent to a downsurge. The tank is normally isolated from the pipeline by one or more lateral pipes in which there is one or more check valves to allow flow into the pipe if the HGL is lower in the pipe than the elevation of the water in the open tank. Under normal operating conditions the higher pressure in the pipeline keeps the check

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valve closed. The major advantage of a one-way surge tank over a simple surge tank is that it does not have to be at the HGL elevation as required by the latter. It has the disadvantage, however, on only combating initial downsurges, and not initial upsurges. One-way surge tanks have been employed extensively by the U. S. Bureau of Reclamation in pump discharge lines, principally by the instigation of Parmakian, the originator of the concept.[19] Another example of the effective application of one-way surge tanks in a pumping system is reported by Martin.[23]

Considerations for design are (1) location of high points or knees of the piping, (2) check valve and lateral piping redundancy, (3) float control refilling valves and water supply, and other appurtenances. Maintenance is critical in order to insure the operation of the check valve(s) and tank when needed.

AIR CHAMBER (HYDROPNEUMATIC SURGE TANK) If properly designed and maintained, an air chamber (Figures 11 and 12) can alleviate both negative and positive pressure problems in pumping systems. They are normally located within or near the pumping station where they would have the greatest effect. As stated by Fox[24] and others, an air chamber solution may be quite effective in solving the transient problem, but quite expensive. Air chambers have the advantage that the tank, sometimes multiple, can be mounted either vertically or horizontally. The principal criteria are available water volume and air volume for the task at hand.

For design, consideration must be given to compressed air supply, water level sensing, sight glass, drains, pressure regulators, and possible freezing. Frequently, a check valve is installed between the pump and the air chamber. Since the line length between the pump and air chamber is usually quite short, check valve slamming may occur, necessitating the consideration of a dashpot on the check valve to cushion closure.

The assurance of the maintenance of air in the tank is essential, usually 50% of tank volume, otherwise the air chamber can be quite ineffective. An incident occurred at a raw water pumping plant where an air chamber became waterlogged due to the malfunctioning of the compressed air system. Unfortunately, pump power failure occurred at the same time, causing water column separation and waterhammer, leading to pipe rupture.

AIR-VACUUM AND AIR-RELEASE VALVES Another method for preventing subatmospheric pressures and vapor cavity formation is the admittance of air from air-vacuum valves (vacuum breakers, Figure 12) at selected points along the piping system. Proper location and size of air-vacuum valves can prevent water column separation and reduce waterhammer effects, as calculated and measured by Martin.[25] The sizing and location of the valves are critical, as stated by Kroon et al.[22] In fact, as reported by Parmakian the inclusion of air-vacuum valves in a pipeline did not eliminate failures.[20,21] Unless the air-vacuum system is properly chosen substantial pressures can still occur due to the compression of the air during resurge, especially if the air is at very low pressures within the pipeline when admitted. Moreover, the air must be admitted quickly enough to be effective.

As shown by Fleming, vacuum breakers can be a viable solution.[12] The advantage of an air-vacuum breaker system, which is typically less expensive than other measures such as air chambers, must be weighed against the disadvantages of air accumulation along the pipeline and its subsequent removal. Maintenance and operation of valves is critical in order for assurance of valve opening when needed. Air removal is often accomplished with a combined air release air-vacuum valve. For finished water systems the admittance of air is not a normal solution and must be evaluated carefully. Moreover, air must be carefully released so that no additional transient is created.

FLYWHEEL Theoretically, a substantial increase in the rotating inertia (WR^2) of a pump-motor unit can greatly reduce the downsurge inasmuch as the machine will not decelerate as rapidly. Typically, the motor may constitute from 75 to 90% of the total WR^2. Additional WR^2 by the attachment of a flywheel will reduce the downsurge. As stated by Parmakian, a 100% increase in WR^2 by the addition of a flywheel may add up to 20% to the motor cost.[19] He further states that a flywheel solution is only economical in some marginal cases. Flywheels are usually an expensive solution, mainly useful only for short

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systems. A flywheel has the advantage of practically no maintenance, but the increased torque requirements for starting must be considered.

UNINTERRUPTED POWER SUPPLY (UPS) The availability of large uninterrupted power supply systems are of potential value in preventing the primary source of waterhammer in pumping; that is, the generation of low pressures due to pump power failure. For pumping stations with multiple parallel pumps, a UPS system could be devised to maintain one or more motors while allowing the rest to fail, inasmuch as there is a possibility of maintaining sufficient pressure with the remaining operating pump(s). The solution usually is expensive, however, with few systems installed.

DESIGN CONSIDERATIONS

Any surge or hydraulic transient analysis is subject to inaccuracies due to incomplete information regarding the systems and its components. This is particularly true for a water distribution system with its complexity; presence of pumps, valves, tanks, and so on; and some uncertainty with respect to initial flow distribution. The ultimate question is how all of the uncertainties combine in the analysis to yield the final solution. There will be offsetting effects and a variation in accuracy in terms of percentage error throughout the system. Some of the uncertainties are as follows.

The simplification of a pipe system, in particular a complex network, by the exclusion of pipes below a certain size and the generation of equivalent pipes surely introduces some error, as well as the accuracy of the steady-state solution. However, if the major flow rates are reasonably well known, then deviations for the smaller pipes is probably not too critical. As mentioned earlier incomplete pump characteristics, especially during reverse flow and reverse rotation, introduce calculation errors. Valve characteristics that must be assumed rather than actual are sources of errors, in particular the response of swing check valves and pressure-reducing valves. The analysis is enhanced if the response of valves and pumps from recordings can be put in the computer model.

For complex pipe network systems it is difficult to assess uncertainties until much of the available information is known. Under more ideal conditions that occur with simpler systems and laboratory experiments, one can expect accuracies when compared to measurement on the order of 5 to 10%, sometimes even better. The element of judgment does enter into accuracy. Indeed, two analyses could even differ by this range because of different assumptions with respect to wave speeds, pump characteristics, valve motions, system schematization, and so on. It is possible to have good analysis and poorer analysis, depending upon experience and expertise of the user of the computer code. This element is quite critical in hydraulic transients. Indeed, there can be quite different results using the same code.

Computer codes, which normally are based upon the method of characteristics (MOC), are invaluable tools for assessing the response of systems to changes in surge protection devices and their characteristics. Obviously, the efficacy of such an approach is enhanced if the input data and network schematization is improved via calibration. Computer codes have the advantage of investigating a number of options as well as optimizing the sizing of surge protection devices. The ability to calibrate a numerical or analysis code to a system certainly improves the determination of the proper surge protection. Otherwise, if the code does not reasonably well represent a system surge protection devices can either be inappropriate or under- or oversized.

Computer codes that do not properly model the formation of vapor pockets and subsequent collapse can cause considerable errors. Moreover, there is also uncertainty regarding any free or evolved gas coming out of solution. The effect on wave speed is known, but this influence cannot be easily addressed in an analysis of the system. It is simply another possible uncertainty.

Even for complicated systems such as water distribution networks hydraulic transient calculations can yield reasonable results when compared to actual measurements provided that the entire system can be properly characterized. In addition to the pump,

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motor, and valve characteristics there has to be sufficient knowledge regarding the piping and flow demands. An especially critical factor for a network is the schematization of the network; that is, how is a network of thousands of pipes simplified to one suitable for computer analysis, say hundreds of pipes, some actual and some equivalent. According to Thorley, a network with loops tends to be more forgiving regarding waterhammer because of the dispersive effect of many pipes and the associated reflections.[26] On the other hand, Karney and McInnis show by a simple example that wave superposition can cause amplification of transients.[27] Since water distribution networks themselves have not been known to be prone to waterhammer as a rule, there is meager information as to simplification and means of establishing equivalent pipes for analysis purposes. Large municipal pipe networks are good examples wherein the schematization and the selection of pipes characterizing the networks need to be improved in order to represent the system better.

NEGATIVE PRESSURES AND WATER COLUMN SEPARATION

For finished water transmission and distribution systems the application of 20 psig (1.38 barg) as a minimum pressure to be maintained under all conditions should prevent column separation from occurring provided analytical models have sufficient accuracy. Although water column separation and collapse is not common in large networks, it does not mean that the event is not possible. The modeling of water column separation is clearly difficult for a complicated network system. Water column separation has been analytically modeled with moderate success for numerous operating pipelines. Clearly, not only negative pressures, but also water column separation, are unwanted in pipeline systems, and should be eliminated by installation of properly designed surge protection devices.

If the criterion of a minimum pressure of 20 psig (1.38 barg) is imposed then the issue of column separation and air-vacuum breakers are irrelevant, except for prediction by computer codes. Aside from research considerations, column separation is simulated for engineering situations mainly to assess the potential consequences. If the consequences are serious, as they often are in general, either operational changes or more likely surge protection devices are designed to alleviate column separation. For marginal cases of column separation the accuracy of pressure prediction becomes difficult. If column separation is not to be allowed and the occurrence of vapor pressure can be adequately predicted, then the simulation of column separation itself is not necessary.

Some codes do not simulate water column separation, but instead only maintain the pressure at cavity location at vapor pressure. The results of such an analysis are invalid, if indeed an actual cavity occurred, at some time subsequent to cavity formation. This technique is only useful to know if a cavity could have occurred, as there can be no assessment of the consequences of column separation. The inability of any code to model water column separation has the following implications (1) the seriousness of any column separation event, if any, can not be determined, and (2) once vapor pressure is attained, the computation model loses its ability to predict adequately system transients. If negative pressures below 20 psig (1.38 barg) are not to be allowed the inability of a code to assess the consequences of column separation and its attendant collapse is admittedly not so serious. The code need only flag pressures below 20 psig (1.38 barg) and negative pressures, indicating if there is a need for surge protection devices.

The ability of any model to properly simulate water column separation depends upon a number of factors. The principal ones are:

- Accurate knowledge of initial flow rates.
- Proper representation of pumps, valves, and piping system.
- Allowing a vapor pocket to form, grow, and collapse.
- Maintenance of vapor pressure within cavity while it exists.
- Determination of volume of cavity at each time step.
- Collapse of cavity at the instant the cavity volume is reduced to zero.

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TIME CONSTANTS FOR HYDRAULIC SYSTEMS

- Elastic Time Constant : $$t_e = \frac{2L}{a} \quad (16)$$
- Flow Time Constant : $$t_f = \frac{LV_o}{gH_o} \quad (17)$$
- Pump and Motor Inertia Time Constant: $$t_m = \frac{I\omega_R}{T_R} = \frac{I\omega_R^2}{\rho g Q_R H_R \eta_{RT}} \quad (18)$$
- Surge Tank Oscillation Inelastic Time Constant: $$t_s = 2\pi\sqrt{\frac{L_T}{g}\frac{A_t}{A_T}} \quad (19)$$

CASE STUDIES—EXPERIENCES WITH SURGE PROTECTION DEVICES

For three large water pumping systems with various surge protection devices waterhammer analyses and site measurements have been conducted. The surge protection systems in question are (1) one-way and simple surge tanks, (2) an air chamber, and (3) air-vacuum breakers.

Case Study with One-Way and Simple Surge Tanks A very large pumping station has been installed and commissioned to deliver water over a distance of over 30 km. Three three-stage centrifugal pumps run at a synchronous speed of 720 rpm, with individual rated capacities of 1.14 m^3/s, rated heads of 165 m, and rated power of 2090 kW. Initial surge analysis indicated potential water column separation. The surge protection system was then designed with one-way and simple surge tanks as well as air-vacuum valves strategically located.

The efficiency of these various surge protection devices was assessed from site measurements. Measurements of pump speed, discharge valve position, pump flow rate, and pressure at seven locations were conducted under various transient test conditions. The site measurements under three-pump operation allowed for improvement of hydraulic transient calculations for future expansion to four and five pumps. Figure 13 illustrates the profile of the ground and the location of the three pairs of surge tanks. The first and second pairs of surge tanks are of the one-way (feed tank) variety, while the third pair are simple open on-line tanks.

Pump trip tests were conducted for three-pump operation with cone valves actuated by the loss of motor power. For numerical analysis a standard computer program applying the method of characteristics was employed to simulate the transient events. Figure 14 shows the transient pressures for three-pump power failure. The transient pressures agree reasonably well for the first 80s. The minimum HGLs in Figure 13 also show good agreement, as well as the comparison of measured and calculated pump speeds in Figure 14. The test program and transient analysis clearly indicated that the piping system was adequately protected by the array of surge tanks inasmuch as there were no negative pressures.

Case Study with Air Chamber Hydraulic transients caused by simultaneous tripping of pumps at the pumping station depicted on Figure 15a were evaluated to assess the necessity of surge protection. Without the presence of any protective devices such as accumulators, vacuum breakers, or surge suppressors, waterhammer with serious consequences was shown to occur due to depressurization caused by the loss of pumping pressure following sudden electrical outage. In the case of no protection a large vapor cavity would occur at the first high point above the pumping station, subsequently collapsing after the water column between it and the reservoir stops and reverses. This phenomenon, called water column separation, can be mitigated by maintaining the pressures above vapor pressure.

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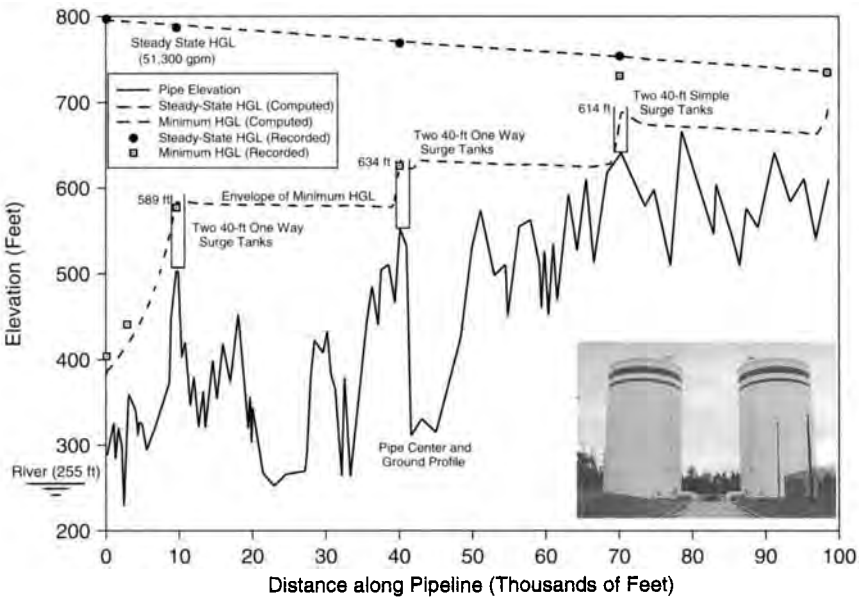


FIGURE 13 Case study of pump power failure at pumping station with three pairs of surge tanks—two pairs one way and one pair simple surge tanks (after Martin, Reference 23)

The efficacy of the 11.6 m (38 ft) diameter air chamber shown in Figure 15a was investigated analytically and validated by site measurements for three-pump operation. The envelope of the minimum HGL drawn on Figure 15a shows that all pressures remained positive. Figure 15b compares the site measurement with the calculated pressures obtained by a standard waterhammer program utilizing MOC.

Case Study with Air-Vacuum Breaker Air-inlet valves or air-vacuum breakers are frequently installed on liquid piping systems and cooling water circuits for the purpose of (1) eliminating the potential of water column separation and any associated waterhammer subsequent to vapor pocket collapse, (2) protecting the piping from an external pressure of nearly a complete vacuum, and (3) providing an elastic cushion to absorb the transient pressures.

A schematic of the pumping and piping system subject to the field test program is shown in Figure 16a. This system provides the cooling water to a power plant by pumping water from the lower level to the upper reservoir level. There are five identical vertical pumps in parallel connected to a steel discharge pipe 1524-mm (60 in) in diameter. On the discharge piping of each pump there are 460-mm (18 in) diameter swing check valves. Mounted on top of the 1524-mm (60 in) diameter discharge manifold is a 200-mm (8 in) diameter pipe, in which is installed a swing check valve with a counter weight. Air enters the vacuum breaker through the tall riser, which extends to the outside of the pump house.

Transient pressures were measured in the discharge header for simultaneous trip out of three, four, and five pumps. The initial prediction of the downsurge caused by pump power failure was based on the method of characteristics with a left end boundary condition at the pumps, junction boundary condition at the change in diameter of the piping, and a constant pressure boundary condition at the right end of the system.

The predicted pressure head variation in the pump discharge line is shown in Figure 16b for a simulated five-pump trip out. The predicted peak pressure for the five-pump trip out

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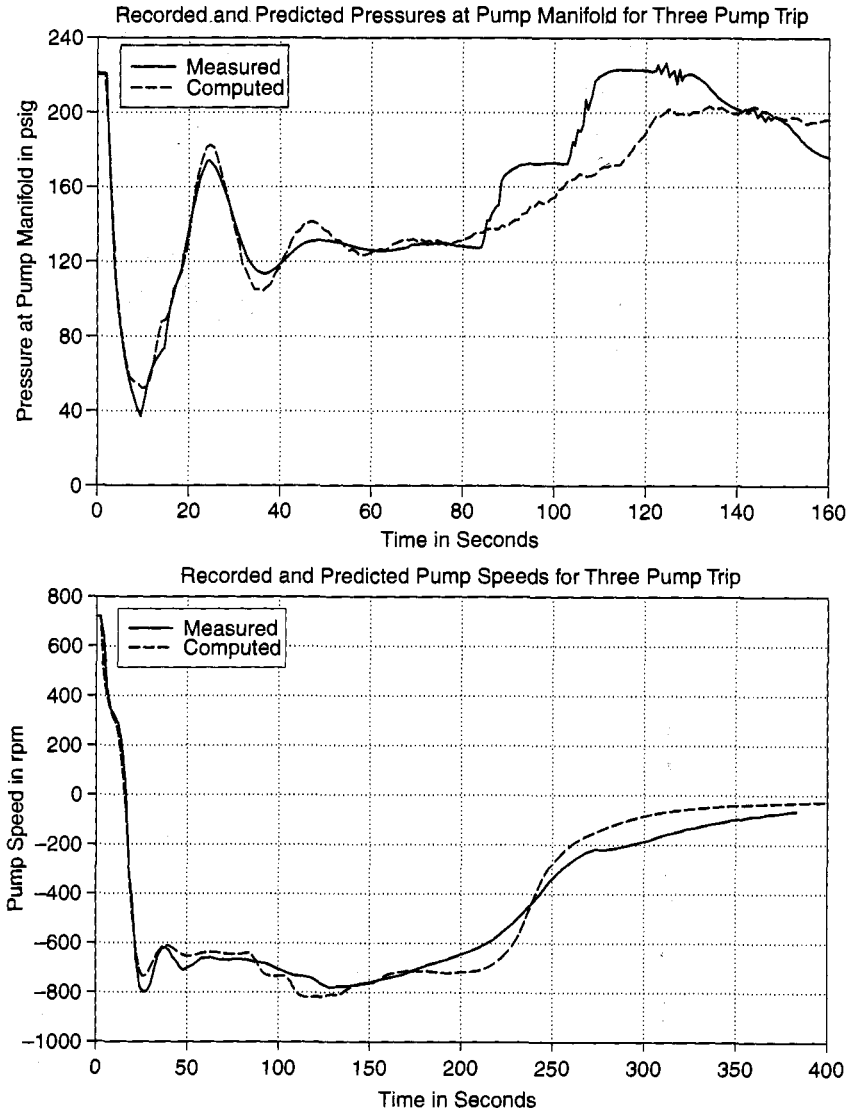


FIGURE 14 Case study of pump power failure at pumping station with three pairs of surge tanks—two pairs one way and one pair simple surge tanks (after Martin, Reference 23)

compares favorably with the corresponding measured peak, but the time of occurrence of the peaks and the subsequent phasing vary considerably. Analysis without a vacuum breaker or other protective device in the system predicted waterhammer pressure caused by collapse of a vapor pocket to exceed 2450 kPa (355 psi). The vacuum breaker effectively reduced the peak pressure by 60%. Peak pressures can be adequately predicted by a simplified liquid column, orifice, and air spring system. Water column separation can be eliminated by air-vacuum breakers of adequate size.

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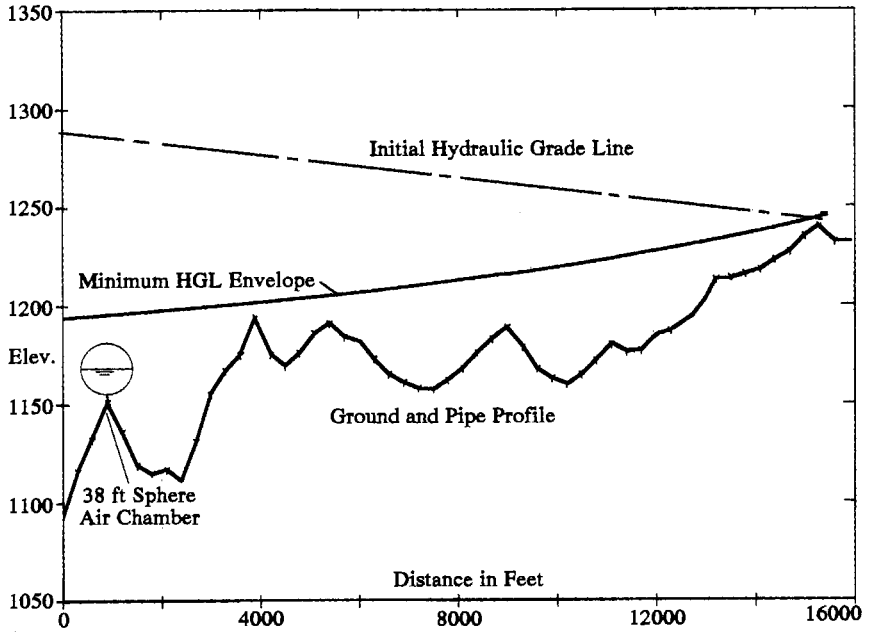


FIGURE 15A Pipe profile and HGL for case study of air chamber performance for raw water supply.

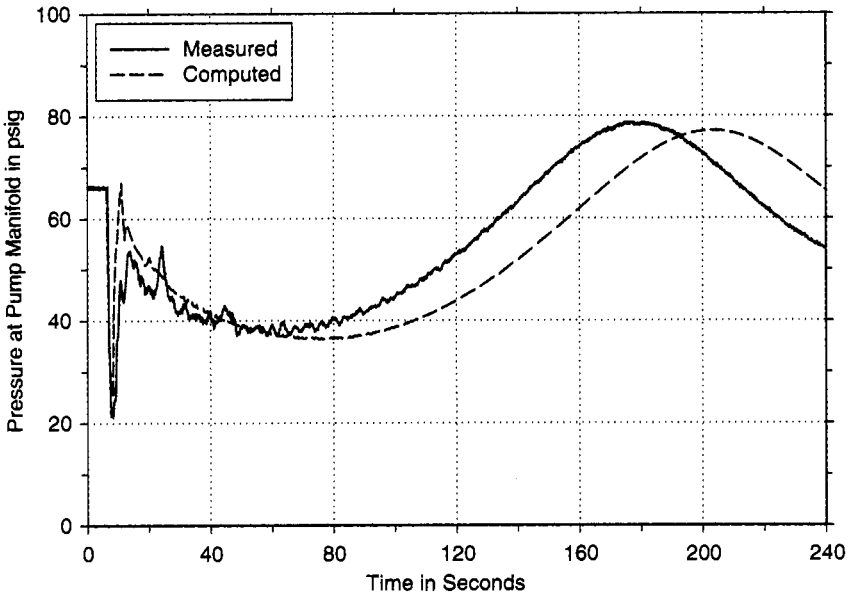


FIGURE 15B Comparison of computed and recorded pressures for case study of air chamber performance for raw water supply

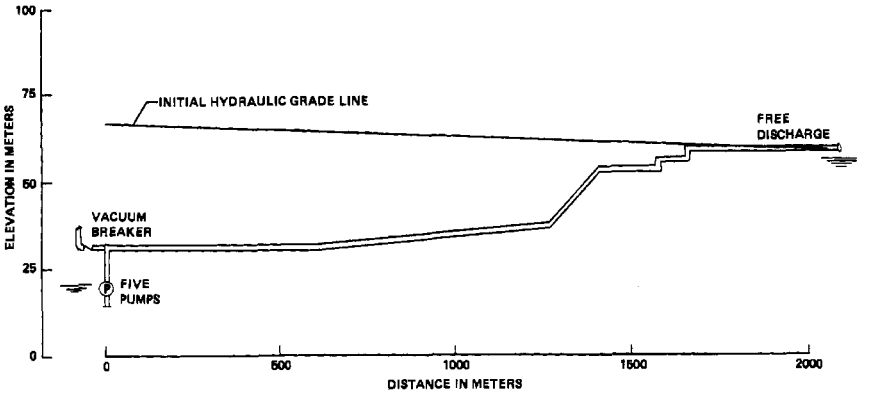


FIGURE 16A Pipe profile for case study of vacuum breaker air chamber performance for raw water supply (after Martin, Reference 25)

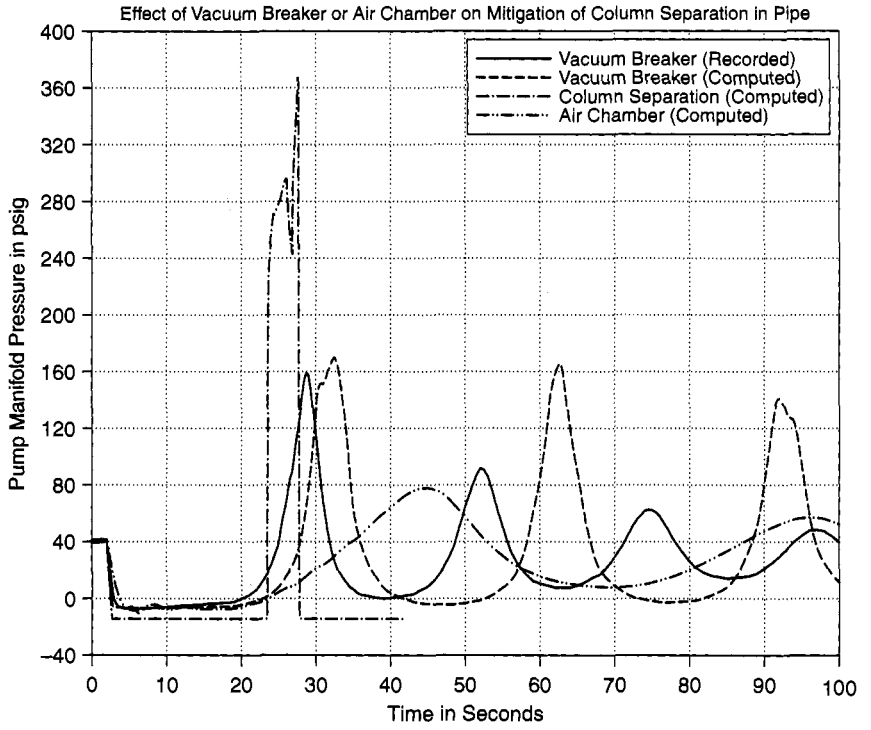


FIGURE 16B Comparison of computed and recorded pressures for case study of vacuum breaker air chamber performance for raw water supply (after Martin, Reference 25)

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SECTION 11.4
MINIMUM FLOW CONTROL SYSTEMS

HORACE J. MAXWELL
DAVID A. KALIX

In many cases, pumps are required to operate over a range of flow rates from a maximum well above the design value down to the minimum that is possible for a given pump size, type, and speed. Guidelines for establishing minimum flow are discussed in Sections 2.1.1 and 2.1.3 for centrifugal pumps, which can be throttled at the discharge to reduce the flow rate. Positive displacement pumps (both reciprocating and rotary), on the other hand, cannot be throttled; so, for the safety of the system and the user, these machines are usually equipped with pressure-actuated relief valves. However, both centrifugal and positive displacement pumps can be equipped with flow-bypass systems (and sometimes variable-speed capability) in order to accommodate varying flow requirements. For centrifugal pumps, which are the focus of this section, such systems act to protect against flow rates falling below the minimum allowable.

In designing centrifugal pumps, engineers strive to develop specific internal geometry that will produce head and flow with low energy loss. Each pump is designed for a specific head versus flowrate for the given impeller speed. The head and flowrate on this family of "characteristic" curves where the energy loss is minimum is known as the Best Efficiency Point (BEP).

In application, pumps spend a significant portion, if not all of their life, operating at conditions other than BEP. This is normal and is due to a combination of system design conditions, available pump designs, and actual plant operating requirements. Low flow related problems occur when pumps continuously operate in or repetitively cycle into flow regions that are significantly below the BEP (for example, 50% of BEP). In these low-flow regions, pump and system component performance and longevity can be adversely affected.

Low flow problems are known to be worst for large high-energy pumps (for example, boiler feedwater), for pumps handling hot liquids, for pumps that handle liquids that have solid particles, and for pumps for low net positive suction head (NPSH) service. A well-designed minimum flow control system can establish an environment that will substantially improve pump and system performance.

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FACTORS AFFECTING LOW FLOW RATE PUMP OPERATION

Collectively, the following broadly classed factors have been recognized as the major contributors to low flow pump problems.

Thermal Factors The increase in temperature of the liquid within the pump is directly related to the pump's efficiency. The energy that is available to heat the flowing liquid and the pump casing is basically the difference between the power input to the pump (brake horsepower) and the useful work done by the pump (liquid horsepower). At low flow conditions, centrifugal pumps are very inefficient and a significant amount of input energy is lost and heats the liquid and the pump assembly. Refer to Section 2.1.3 and Chap. 15 for more discussion on thermal effects.

Hydraulic Instabilities When the pump is operating significantly below the BEP, flow streamlines (i.e., patterns) within the pump change considerably from the rated design streamlines. Fluid eddies are most likely to develop at the inlet and discharge of the impeller resulting in flashing, cavitation, and shock waves that often produce vibration and serious component erosion. This phenomenon is classically known as internal recirculation. It can occur at the pump inlet (suction) and discharge. Refer to Section 2.1.1 and 2.1.3 for a more in-depth discussion on this topic.

Mechanical Loads As the flowrate through the pump decreases, steady state loads increase and superimposed dynamic cyclic loads appear radially and axially on the impeller and shaft. The dynamic cyclic component increases significantly when recirculation within the pump occurs. Bearing damage, shaft and impeller breakage, and rubbing wear on casing, impeller and wear rings can occur. See Sections 2.1.3 and 2.1.4 for discussion of this subject.

Axial-flow and mixed-flow pumps with high specific speed produce comparatively higher head and take comparatively more power at low flow. A bypass system may be necessary not only to reduce component loading and stress but also to prevent motor overload. See Section 2.1.3 and Section 11.1 for discussion.

Abrasive Fluids Liquids containing a large amount of abrasive particles, such as sand or ash, must flow continuously through the pump. If flow decreases, the particles can circulate inside the pump passages and quickly erode the impeller casing, wear rings, and shaft.

ESTABLISHING MINIMUM PUMP FLOW REQUIREMENTS

The bypass system designer must know the minimum pump flow specified by the pump manufacturer in order to properly design a bypass system. The four previously discussed topics should be evaluated in detail by the pump manufacturer to establish the minimum flowrate specification. Minimum flowrate specifications are generally established through a combination of analytical and experimental techniques coupled with field performance data.

Thermal Considerations The maximum allowable temperature rise of the pump is primarily based on two points: the permissible pump casing and shaft thermal growth, and the flash point temperature of the pumpage. Pump manufacturers use analytical, laboratory and field data to validate their thermal analysis to ensure that pumps do not seize within the allowable temperature operating ranges. Refer to Section 2.1.3 for temperature rise calculations.

Applications involving extremely high or low temperature fluids may require more in-depth analysis to determine if individual component thermal growth is the limiting factor in determining minimum flowrate. Additionally, certain chemicals, which polymerize or solidify at particular temperatures, may establish the minimum flowrate specification. Chap. 15 and Section 2.1.3 provide more detail on this subject.

Hydraulic Considerations The minimum bypass flow requirement for most pumps is based on minimum continuous stable flowrate, a hydraulic criterion, rather than a

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temperature rise. Pump internal recirculation will occur at both the impeller inlet and impeller outlet as flowrates are reduced. Internal recirculation will occur at flowrates well above those that cause temperature concerns. Refer to Section 2.1.3 for a detailed evaluation of this topic.

Mechanical Considerations It is necessary to know how head, radial thrust, axial thrust, and power vary with capacity before deciding on minimum allowable flow. Bearing capacity, motor rating, and stresses in drive and driven components are important influences.

Abrasive Wear Considerations Relatively high bypass flowrates may be required to protect the pump against abrasives in the liquid. Heavy wear can occur at flows below 85% of the best efficiency point. The designer must establish the minimum pump flow specification using the pump manufacturer's recommendation and his experience with comparable pumps and liquid/solid mixtures.

MINIMUM FLOW CONTROL SYSTEM DESIGN FACTORS

Pump Size Capacity, power, specific speed, and suction specific speed are all factors that must be examined when designing a bypass system. These factors have a direct impact on the cost of building and operating the bypass system. Use of a continuous bypass system will require an even larger pump and driver to supply both the process and bypass flow requirements simultaneously.

Discharge Pressure High discharge pressures result in high head loss in bypass valves, components, and lines. Liquids that can flash and cavitate demand special precautions to minimize damage in valving, orifices, and piping.

Available Heat Sink Bypass flow must be reintroduced into the system far enough upstream to prevent progressive temperature buildup or flow disturbance in the pump suction. This may mean a simple discharge back to an uninsulated inlet line or discharge to a receiving tank or cooler with enough area and enough inflow of cool liquid to handle the thermal load. Bypass flow can discharge into a deaerator storage tank, a condenser, a flash tank, or a cooling pond. Elevation, distance, and pressure inside the receiving tank are also factors, as is the fact that the interior must be available for inspection and for repair of spargers, spray or distribution pipes, orifices, and backpressure regulators.

Pump Design A pump's design and materials of construction often affect the minimum allowable percentage of flow. With thermal effects, pumps vary in the length of time that they will tolerate shut-off or low flow. This is important in designing the bypass system valves, instrumentation, and controls.

Hydraulic effects at low flow are most apparent in high-energy pumps. The pump manufacturer should state the continuous minimum hydraulically acceptable flow for a given pump—and how it was determined. With axial-flow pumps, the shape of the pump head curve may be a factor in selecting the required bypass flow percentage. If feasible, a witness shop test of the pump should be specified to demonstrate and verify minimum flow recommendations.

Liquid Pumped Liquids that flash and cavitate generally required a high bypass flow percentage. Examples are liquids near the boiling point or at high pressure. Abrasives in the liquid may require more bypass than would be needed for thermal reasons alone.

Energy Costs A high energy cost to operate the pump requires careful consideration of bypass system design. The evaluation should compare equipment installation costs, maintenance, and energy costs for various bypass configurations. Figure 1 shows the annual pumping costs for a continuous bypass type system based on bypass flow, pressure, and energy rates. The example shows a pump with a discharge head of 500 ft (152 m)

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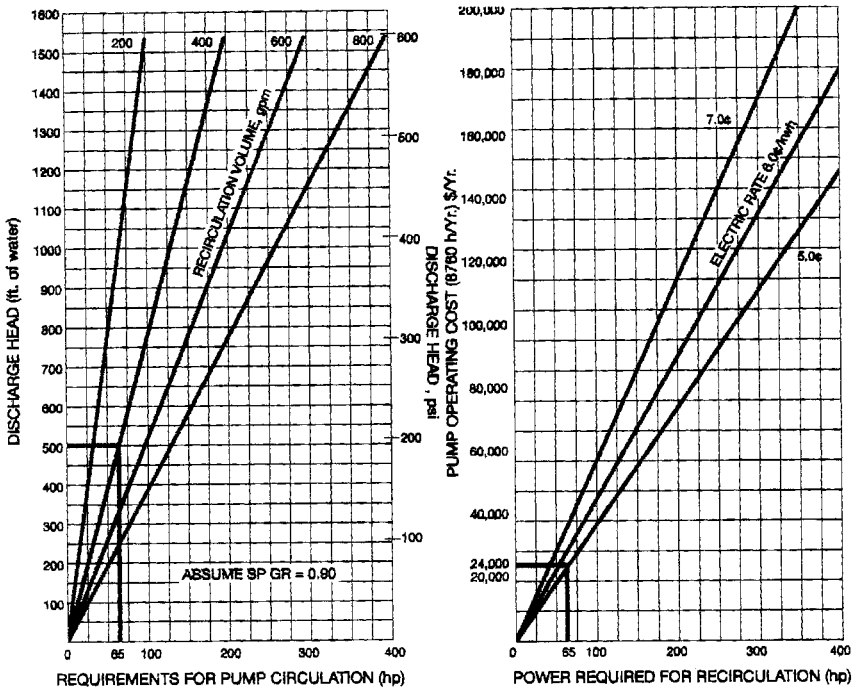


FIGURE 1 Annual pumping cost estimate for continuous bypass systems (metric conversions: ft × 0.3048 = m, gpm × 0.277 = m^3/h, hp × 0.746 = W)

and a bypass flow requirement of 400 gpm (91 m^3/h). Based on an energy cost of 5.0 cents/KWH, the annual cost for continuous bypass is $24,000. An automatic bypass system will only open the bypass when process flow demand is low. The total design life energy costs of a continuous bypass system can easily exceed the hardware costs of an automatic bypass system.

Noise Considerations Bypass systems can easily exceed OSHA requirements for occupied spaces if not designed properly. High-pressure drops and high-fluid velocity increase noise. Multistage pressure reduction, heavy wall pipe, insulation, and silencers will all combine to reduce noise to acceptable levels. See Chap. 10 for a further discussion of this topic.

Process System Design and Operational Expectations Bypass system design will depend on the plant design life and the expected process operational requirements. For example, a swing-loaded electric power plant will have far different pump operating requirements than a low-pressure emergency fire water system. The bypass system designer must evaluate installation, maintenance, and operating costs for the life of the process system with the expected utilization. If the actual operating conditions differ significantly from the design, the configuration of the bypass system should be reevaluated. For example, a process designed for normal operation with one pump at 75% of maximum capacity may put severe demands on the bypass system if the pump is operated continuously at 20% of maximum capacity.

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LOW FLOW PROTECTION SYSTEMS

Bypass Systems—Types/Design Considerations

CONTINUOUS BYPASS SYSTEMS As the title implies, continuous bypass systems provide continuous flow whenever the pump is running, regardless of the process demand. Figure 2 illustrates a simple system, with a bypass line branching off the pump discharge upstream of the main line check valve and containing a fixed orifice dimensioned by analysis to provide minimum required pump flow. The bypass line discharges into a reservoir that is at a lower pressure than the pump discharge. The bypass line can also discharge directly into the pump supply line. However, the piping system design must ensure that the bypass liquid temperature does not increase to an unacceptable level. Additionally, vapor bubbles formed in the bypass by the pressure reduction process may be introduced into the pump. This will affect pump performance and longevity. Locating the bypass branch-off before the discharge check valve as shown keeps backflow from the process or from a parallel operating pump from going back to the receiving tank or back through the pump during a pump shutdown. Consideration should also be given to installing a check valve in the bypass line.

The size of the bypass pipe depends on flow and piping configuration. If the pressure drop through the orifice results in flashing flow, the orifice should be located at the end of the bypass piping and should discharge directly into the larger receiving tank or larger downstream piping. The discharge flow should be directed so it does not impinge on the walls of the receiving tank but rather into the liquid to absorb the force of cavitation implosion. Valves and fittings located immediately downstream of the orifice may be damaged by cavitation. Full ported gate or ball valves are less susceptible to damage. Waterhammer and erosion are not problems in well-designed continuous bypass systems.

The pump and prime mover must be sized to simultaneously supply both the bypass flow and the maximum process flow at the required pump discharge pressure.

AUTOMATIC BYPASS SYSTEMS Automatic recirculation systems control the bypass flow in relation to the process flow. The sum of the process and bypass flowrates will always exceed the minimum flowrate specified to protect the pump. Two essential elements of these systems are a device to measure the process flow and a device to control the bypass flow. Compared with continuous bypass, these systems reduce energy consumption and pump horsepower requirements.

The bypass flow can be regulated by either "Modulated" or "On-Off" control. When bypass flow is "Modulated," the bypass flow is inversely proportional to the process flow.

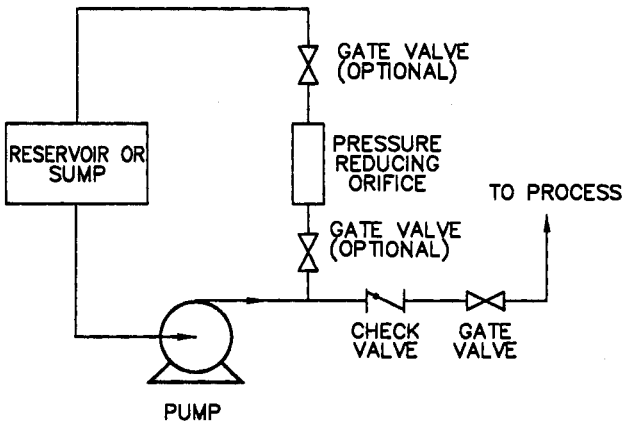


FIGURE 2 Typical continuous bypass system

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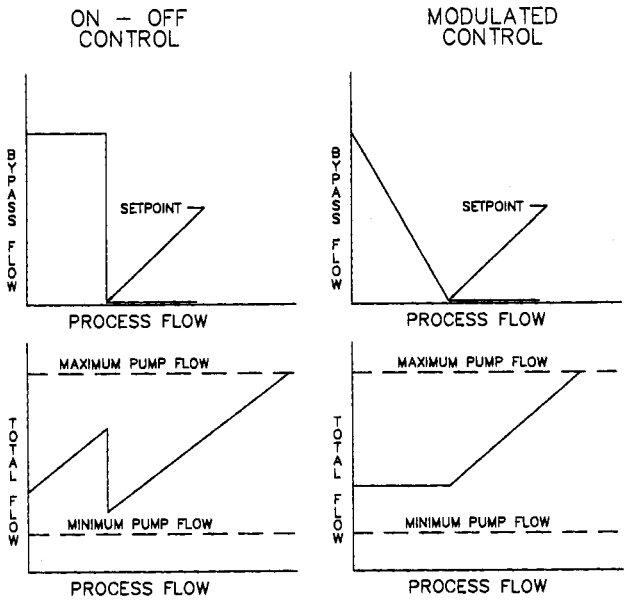


FIGURE 3 Comparison of On-Off versus Modulated automatic bypass flow control

The bypass is shut when process flow is greater than the control setpoint (i.e., minimum flowrate required), opens more as the process flow is reduced, and is full open when the process flow is zero. With "On-Off" bypass control, the bypass is open fully when process flow is below the setpoint and is shut when the process flow is above the setpoint.

Figure 3 illustrates these control methods and the effect on total pump flow. The total pump flow is the sum of the process and bypass flows; it is the flow that enters the pump suction. The "On-Off" control graph shows that there is a step change in the total flow when the bypass opens or closes. The "Modulated" control regulates the total flow smoothly when the process flow is below the setpoint. Either method will protect the pump from damage due to low flow. However, "On-Off" control may produce large pump discharge pressure variations depending on the shape of pump head curve. Rapid pump flow changes may produce damaging hydraulic or mechanical shocks. This will reduce pump life and increase maintenance. Modulated bypass control is required when the minimum flowrate specification exceeds approximately 40% of the rated pump capacity. If "On-Off" bypass control is used above this point, the pump may exceed its capacity rating with resultant pressure pulsation and unsteady flow.

CONTROL LOOP SYSTEMS A typical automatic bypass system, based on an instrumented flow control loop, is shown in Figure 4. This system requires a flow meter, a check valve, a bypass control valve, a valve controller and a pressure-reducing orifice.

The flow meter can be placed at the pump inlet or discharge before the bypass tee (as opposed to it having to be placed after the bypass tee in an "On-Off" system). The controller compares the flow meter signal with the setpoint (minimum pump flow required) and operates the bypass control valve.

Single stage control valves are commonly used in low-pressure systems with an orifice for pressure reduction located downstream. However, these are not suitable for high-pressure differentials or for the controlling of cavitation or flashing. Pressure reducing valves with multistage trim, good seat tightness, and low-pressure recovery characteristics are widely used in this service.

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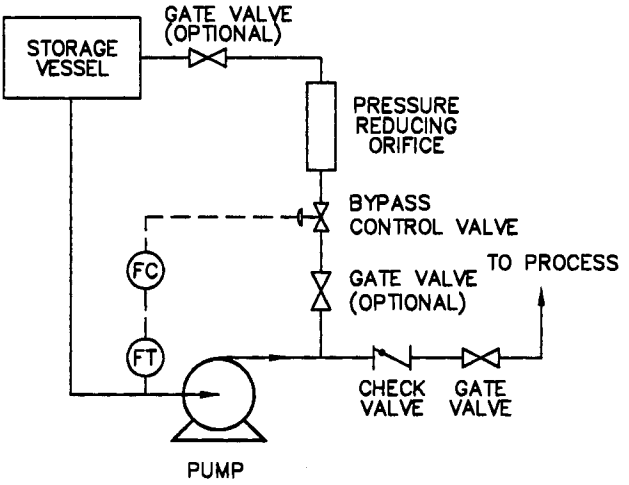


FIGURE 4 Typical automatic bypass system based on flow control

The bypass pressure reducing orifice handles the high velocity and erosive forces of the bypass flow. However, the handling of cavitation and flashing by this fixed restriction is limited. The orifice is sized to operate properly at one bypass flow condition. At low flows, the orifice does not provide the correct pressure reduction as designed.

The check valve is placed in the main line, as shown in Figure 4, to prevent reverse flow through the pump. The pump and motor bearings can be damaged if rotated in reverse.

A typical automatic bypass system based on pump differential head is shown in Figure 5. This system is identical to the flow controlled bypass system except that pump differential head is measured instead of flow. This system can only be used with continuously rising pump head characteristic curves. The system will have to be adjusted for each individual pump characteristic and later adjusted if this characteristic changes over the life of the pump. Refer to Section 11.1, "Throttling Orifices" and Section 11.2, "Centrifugal Pump Bypass."

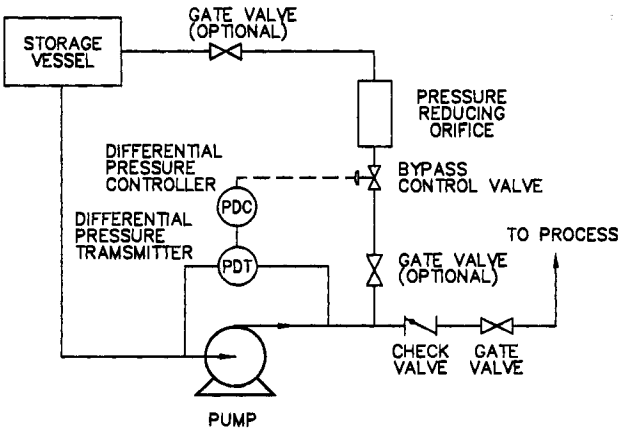


FIGURE 5 Typical automatic bypass system based on pump differential head

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Depending on the bypass valve and control instrumentation selected, these systems can provide either "Modulating" or "On-Off" bypass control. Valves, instrumentation, and controls must be individually sized for the service conditions. All components must be integrated together to provide bypass flow control that meets the pump and process operational design criteria.

The Automatic Recirculation Control (ARC®) Valve This valve provides bypass flow control, pressure reduction, and reverse flow pump protection all within a single unit. This single valve combines the functions of the check valve, pressure reducing orifice, pipe tee, control instrumentation flow meter, and bypass control valve that are all required in the instrumented control loop bypass system. Valves can be designed to provide either "Modulated" or "On-Off" bypass control. The operation of all ARC valves is fundamentally the same. The flow sensing element (check valve disc) responds to the process flow demand and opens or closes the self-contained bypass control valve. Differences in ARC valve design center around bypass valve actuation method, repairability, serviceability, and adjustability. Figure 6 illustrates a typical ARC valve bypass system.

Pilotless Trim Design The simplest of ARC valve designs operate the bypass flow control valve directly by the flow sensing element. The construction of a typical ARC for low-pressure service is shown in Figure 7. The basic operation of this valve is shown in Figure 8. At zero main flow ("A"), the bypass provides full recirculation flow. As main flow increases ("B"), the bypass modulates proportionally closed. At full main flow ("C"), the bypass is shut. The pressure reduction from the pump discharge to the bypass is accomplished by the characterized orifices in the bypass element that is attached to the disc. However, the differential pressure from inlet to bypass imposes static and dynamic forces on the disc assembly. These unbalanced forces interfere with the normal disc motion and limit the maximum differential pressure for pilotless trim. There is an upper differential pressure limit at which this design will not operate satisfactorily and pilot operated bypass control must be used.

Pilot Operated Trim Pilot operated bypass control valves utilize a combined hydraulic-mechanical force to control the bypass flow. Only a relatively small mechanical force is necessary to hydraulically generate the large forces required to operate a bypass flow control valve in a high-pressure system. Figure 9 illustrates a pilot-operated bypass that utilizes a lever connected to the check valve disc to actuate the pilot. Figure 10 illustrates a pilot operated multistage bypass operated by an in-line pilot valve. Pilot-operated valves require more components and seals than unbalanced valves and are therefore more costly.

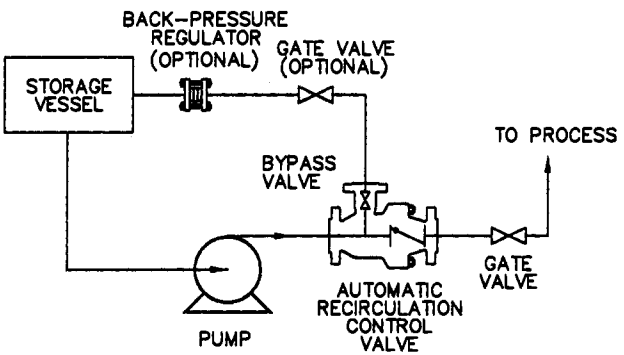


FIGURE 6 Typical ARC valve bypass system

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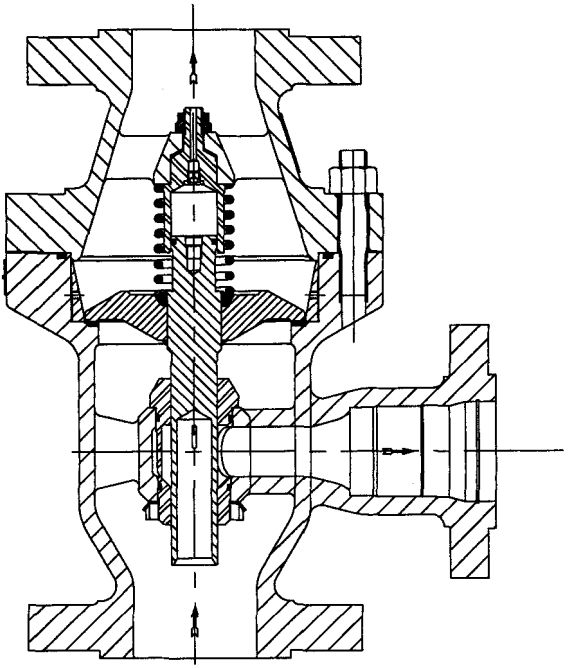


FIGURE 7 Construction of a typical ARC valve for low-pressure service

Serviceability ARC valves can be designed for in-line serviceability. Access is provided to the valves internal components for inspection or repair without disturbing the process or bypass piping. Figure 11 illustrates a valve designed to be serviced in-line.

SAFETY SHUTDOWN SYSTEMS

These controls are normally provided as backup pump protection for the continuous bypass and automatic bypass control systems. They should only operate if the normal system fails due to mechanical or operator causes. In each case, the protection system trips the pump off.

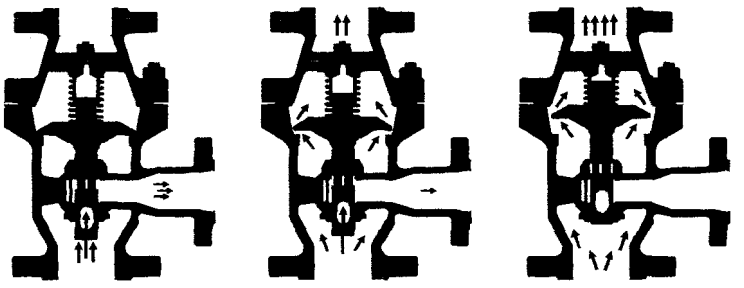


FIGURE 8A through C Basic ARC valve operation at three process flow conditions

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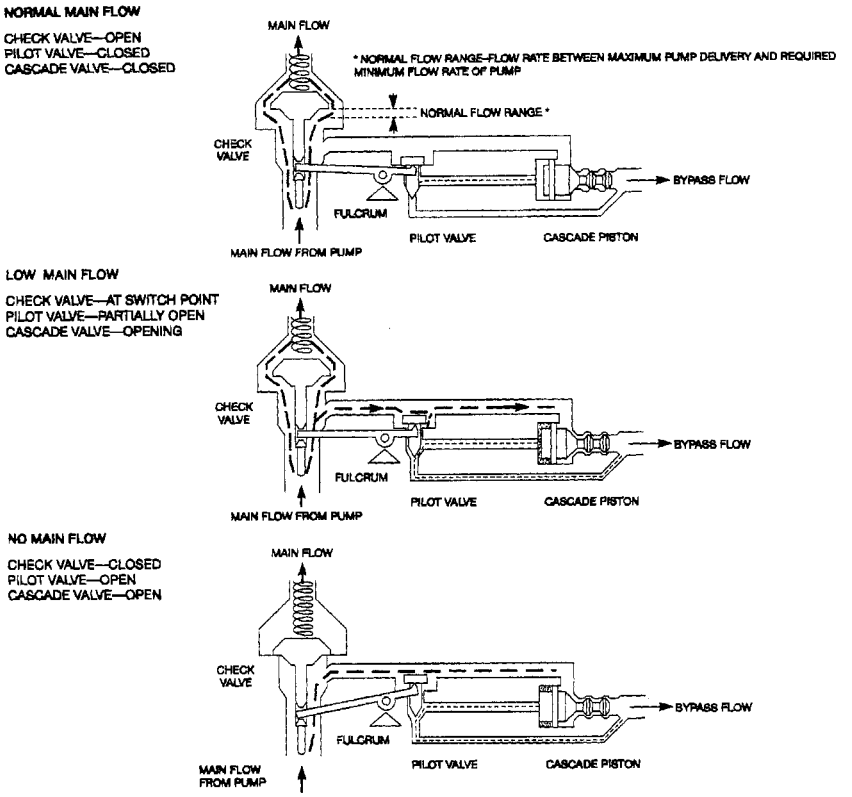


FIGURE 9 Operation of a typical pilot-operated ARC bypass valve actuated through a lever mechanism

Head This requires only an accurate pressure switch, set for a discharge pressure corresponding to the total pump head when the pump is close to the shutoff capacity. For conservative results, calculate based on minimum suction pressure. To use this method, the pump characteristic must rise continuously to shutoff. Control circuitry may require delays to reject spurious pressure surges.

Temperature A temperature sensor mounted on the pump casing or in the discharge and close to the pump can signal for a shutdown of all or some of the operating pumps. The trip can be on either a high-temperature sensing or a high differential temperature sensing between inlet and discharge. This method can protect against large temperature rises (occurring at perhaps 5% or less of capacity) but is less sensitive to smaller rises at higher percentages of capacity where adverse hydraulic effects begin for large and high-power pumps.

Low-Flow Systems These protection systems can be configured in various ways. A low-flow switch can trip the pump off at a predetermined setpoint. A flow meter can signal a controller to shut down the pump. If this protection circuit is installed, the pump start circuit must temporarily bypass this trip in order to start the pump. This is normally accomplished by a spring-loaded switch that bypasses the low-flow trip when held in the "Start" position, but activates the trip protection in the "Run" position.

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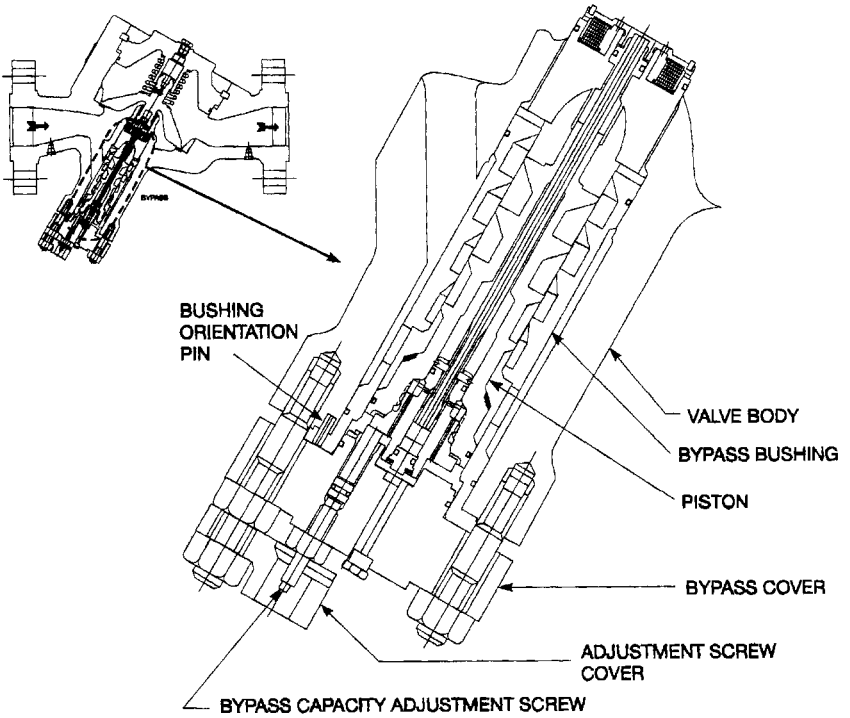


FIGURE 10 Pilot operated multistage bypass trim actuated by an in-line pilot valve

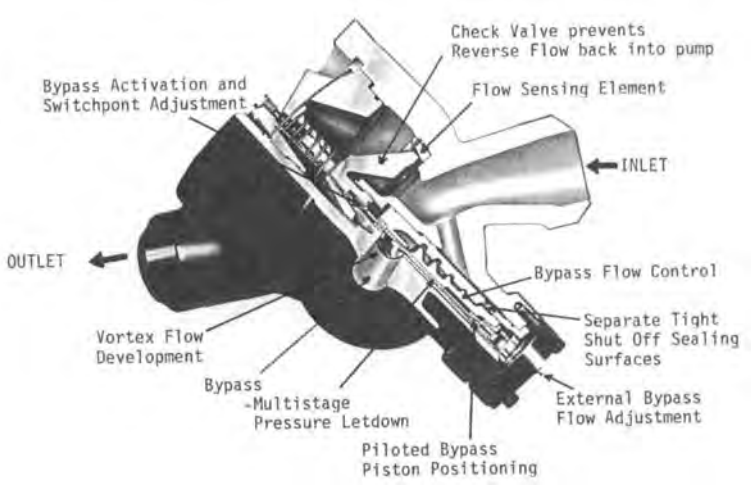


FIGURE 11 Typical in-line serviceable ARC valve

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OTHER BYPASS SYSTEM HARDWARE

Orifices In high-pressure applications, the system often does not provide adequate pressure in the bypass line to prevent cavitation or flashing. Either of these conditions is undesirable because it can cause damage to both valves and the pipe system or cause a reduction in flow below the minimum desired, jeopardizing the pump protection system. All pressure-reducing valves will experience a velocity-induced recovery effect that will limit the amount of pressure drop a valve can take and cause a reduction in flow capacity.

The requirement of backpressure is generic to all pressure-reducing applications. Pressure reduction even by multiple stage cascading can minimize the requirement; however, no valve design will redefine a fluid's physical properties. This becomes especially important in modulating systems. A fixed orifice will not provide the proper backpressure at all flow levels.

As the flow in the bypass line is reduced, the orifice becomes less effective. Proper system design should be used to optimize valve pressure reduction and consider all fluid dynamic effects downstream of any pressure-reducing device. When adequate backpressure is not available downstream of a pressure reducing valve, vapor bubbles will form in the zone just downstream of the valve last stage control surface. This zone is defined as the "Vena Contracta" and represents the point of highest fluid velocity and lowest pressure.

The potential for 1) damage to downstream piping components, and 2) flow reduction exists from this point. When line pressure remains below the fluid vapor pressure, any existing bubbles will remain and expand as piping friction further reduces line pressure. This can be defined as *"flashing condition"* and is characterized by a polished appearance on affected surfaces. When the line pressure drops below the fluid vapor pressure and then recovers, any entrapped vapor bubbles will collapse (implode). This is defined as a *"cavitating condition"* and is characterized by a cinder-like appearance on affected surfaces. The resolution of either condition is best addressed by eliminating vapor formation. This can be assured by the provision of adequate back pressure through the use of a fixed or variable orifice.

Fixed Orifice Simple, easily replaced orifices that reduce the pressure are an effective way to reduce bypass head and provide adequate backpressure in bypass systems. Several stages may be necessary, however, to break down high-pressure drops without flashing. For calculations of flow through standard-shaped orifices, see Sections 11.1 and 11.2.

Coefficients of discharge for oddly shaped multistage orifices are difficult to calculate. However, manufacturers of these specialties can furnish curves of delivery as a function of pressure.

Variable Orifice In modulating systems, a fixed orifice will not provide the proper backpressure over a wide flow range. A backpressure regulator (BPR) has a variable orifice with a spring-loaded plunger that is designed to open at a specified differential pressure. If flow and differential pressure increase, it opens further to maintain the differential pressure and backpressure constant. Figure 12 illustrates a typical BPR construction. Figure 13 shows a standard BPR installation. The BPR is normally located as close to the receiver vessel as possible so that the correct backpressure is maintained in the entire bypass line.

Valves, Piping, and Fittings For cold water at low pressure, a simple power-actuated, globe-type bypass valve is often adequate. In modulating bypass systems, the bypass valve must resist throttling damage, particularly if the water is hot. Staging the pressure drop in the valve is the most common way to reduce or eliminate flashing and cavitation damage to the valve trim or body. Figure 14 illustrates a typical multistage pressure reducing valve (PRV).

Pressure is reduced in stages to ensure that the pressure never decreases below the fluid vapor pressure. This prevents cavitation and the resultant valve damage and noise. Figure 15 illustrates typical calculations for reducing pressure in sequential stages. Refer to Chap. 7 for detailed information regarding valve sizing and selection.

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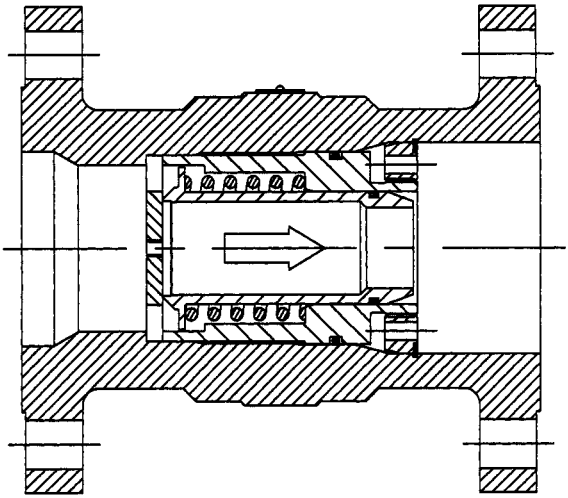


FIGURE 12 Construction of a typical backpressure regulator

Pipe material like that used for the main discharge is adequate for nearly all of the bypass line. Near the orifice and control valve outlets, however, heavier wall or higher chrome content will lengthen life. Welded piping is common for high pressures. To prevent erosion, pipe fittings (especially elbows) should not immediately follow an orifice.

Flow Meters Flow rate is the variable that must be measured for most automatic bypass multicomponent control systems. The meter may have any type of primary element that will produce an accurate signal at the process flow for which the bypass must be controlled.

A simple orifice meter or venturi tube is commonly used. The user must have the required straight upstream and downstream pipe lengths or use flow straighteners to obtain an accurate reading. The device must be properly sized to provide both accurate indication at relatively low process flows and satisfactory pressure drop at maximum process flow conditions.

Flow meters can be located either upstream or downstream of the pump. Meters located upstream are at lower internal pressure but are larger in diameter with larger flanges. In addition, pressure drop at rated flow is important to avoid low pump NPSH. Meters located downstream are normally smaller but must be rated for the maximum pump discharge pressure.

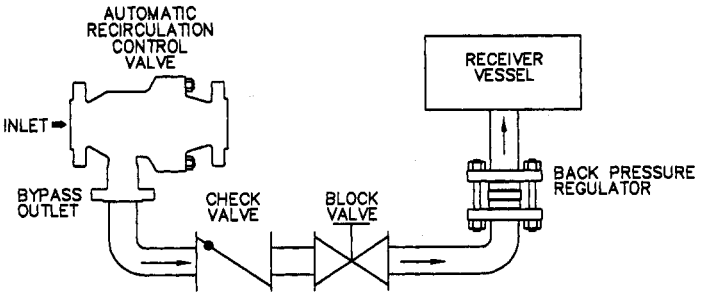


FIGURE 13 Backpressure regulator installation

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FIGURE 14 Construction of a typical multistage pressure reducing valve (Flowserve Corporation)

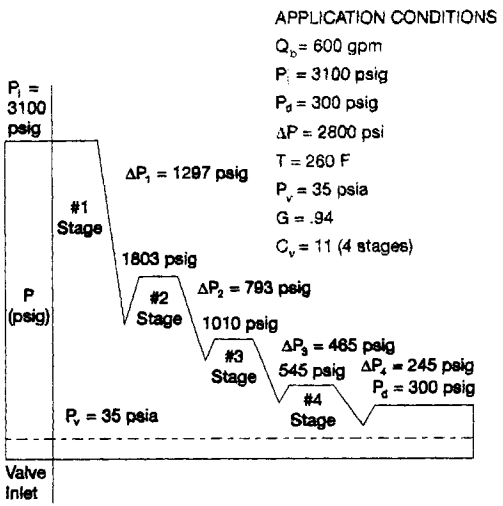


FIGURE 15 Calculations for pressure reduction through a multistage valve

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Controls Controls range in complexity from a simple pressure switch that stops a pump drive motor to modulating systems that meter flow and maintain a selected minimum flow through the pump. The control system may have to provide a deadband near the opening point and be sufficiently stable to hold bypass flow nearly constant in spite of erratic flow during upsets and startup.

For example, when the primary flow-measuring element is upstream of the bypass branch off, sufficient decease in flow will cause the bypass control to open in an "On-Off" bypass system. The main control valve, farther downstream, will then also open to maintain flow to the boiler or process. The primary element will sense the added flow and, in a simple bypass control system, close the bypass valve, initiating hunting in flow and valve action. To prevent this, the setpoint for bypass valve closing must be greater than twice the minimum bypass flow. If minimum flow is in the 30–50% range, this wastes energy during bypass. This control problem is avoided if a modulating system is installed.

SPECIAL BYPASS SYSTEMS

Pumps in Parallel It is possible to provide a single bypass valve for two or more pumps, as shown in Figure 16. The bypass line and valve must be able to handle simultaneously the total bypass flow of all pumps. The pumps must operate together and they must have identical head-capacity curves. If they do not, the bypass flow will differ between pumps. The pump with the lower head may have insufficient bypass flow or could be completely shut-off.

In the system of Figure 16 (left), the main line check valve prevents backflow through an idle pump. If the branch off were installed as shown in Figure 16 (right), an additional check valve in each bypass connection would be necessary to stop the backflow. Therefore, this latter configuration is not recommended.

An alternative arrangement that provides better operational flexibility and pump protection is illustrated in Figure 17. Each pump is individually protected by its own automatic recirculation control (ARC) valve. Pumps can be operated individually or in any combination and the pump head characteristic curves do not have to be identical.

Pumps in Series A single bypass line and valve can protect two pumps in series (see Figure 18). The designer must provide for the larger of the two separately evaluated bypass flows and must take into account the heating effect of the upstream pump. The pumps cannot operate singly unless additional piping and controls are installed.

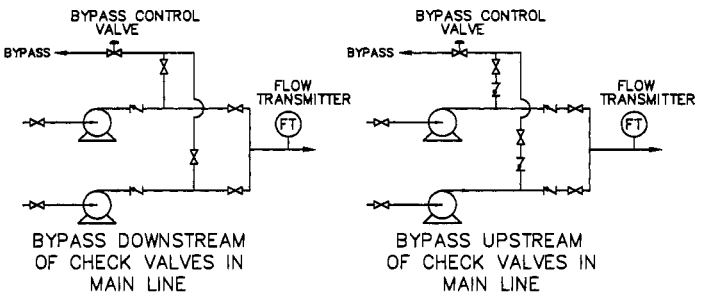


FIGURE 16 Examples of a single bypass valve that provides minimum flow control for two parallel pumps

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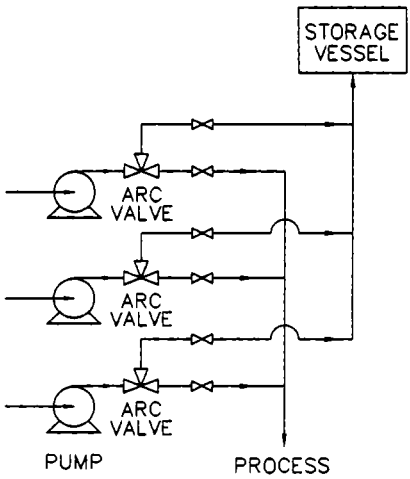


FIGURE 17 Multiple parallel pumps individually protected by ARC valves

VARIABLE SPEED PUMP BYPASS CONSIDERATIONS

Large horsepower pumps are often powered by steam turbines or with motors connected through a variable speed coupling. The effect of decreased speed and discharge head is to reduce the minimum flow requirement for the pump. Therefore, if the minimum required bypass flow is calculated and sized with the pump at normal operating speed and pressure, the bypass flow will be adequate when the pump is operated at lower speeds.

Warning If a constant backpressure device such as a backpressure regulator is used to control bypass pressure, precautions must be taken to ensure that the pump is never operated at pressures below the BPR setting. If the pump is operated below the BPR setting, the pump will have no bypass flow and may be damaged in a short time.

In a typical boiler system, a constant speed booster pump, requiring less minimum flow than the boiler-feed pump, could be protected by the bypass flow for the variable speed boiler feed pump. The designer must determine if the booster pump bypass flow is sufficient when the high-pressure feed pump is running at minimum discharge pressure.

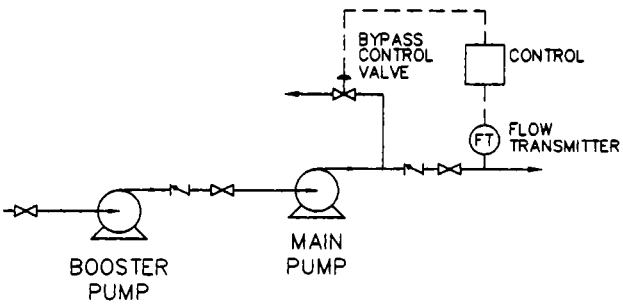


FIGURE 18 Example of a single bypass line that protects two pumps in series

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C • H • A • P • T • E • R • 12

PUMP SERVICES

SECTION 12.1
WATER SUPPLY

JIM BIRD

"Water is the driving force of all nature." (Leonardo da Vinci)

INTRODUCTION

The supply of water to households and industry in sufficient quantities and of a high enough quality to make it something that is taken for granted is one of the hallmarks of the United States' infrastructure. In the years just following the turn of the twenty-first century many retrospective surveys were conducted and one prominent engineering organization determined that the development of reliable and safe water supply and distribution systems ranked fourth on a list of the greatest engineering achievements of the twentieth century.[1] Arguably the key element in all these systems is the ability to draw water from the source, transport it to treatment facilities, process it through the treatment and finally send it out to consumers where it is used with little consideration of the effort required to supply such a basic need. Standing as the most important piece of machinery in each of these steps is the pump, in a wide variety of the classifications described in Chap. 1 of this book.

WATER RESOURCES

"In one drop of water are found all the secrets of all the oceans." (Kahlil Gibran)

The Water Cycle The first step in the full utilization of any resource is an understanding of its nature. When the earth and its atmosphere are viewed as a whole it defines a closed system not gaining or losing matter, including water. What this means is that the water present at the beginning is still the water that is here today. When taking a closer look into the earth's system it is evident that water is always in the process of change. Readily recognized are the many forms of water that exist at any one time and how freely water moves between them. This movement from phase to phase is a combination of the filtration, distillation, and purification processes that are combined into the steps of the *water cycle* (see Figure 1).

The water cycle has no starting point. But, one can begin in the oceans, since that is where most of earth's water exists. The sun, which drives the water cycle, heats water in the oceans. Some of it evaporates as vapor into the air. Ice and snow can sublimate directly

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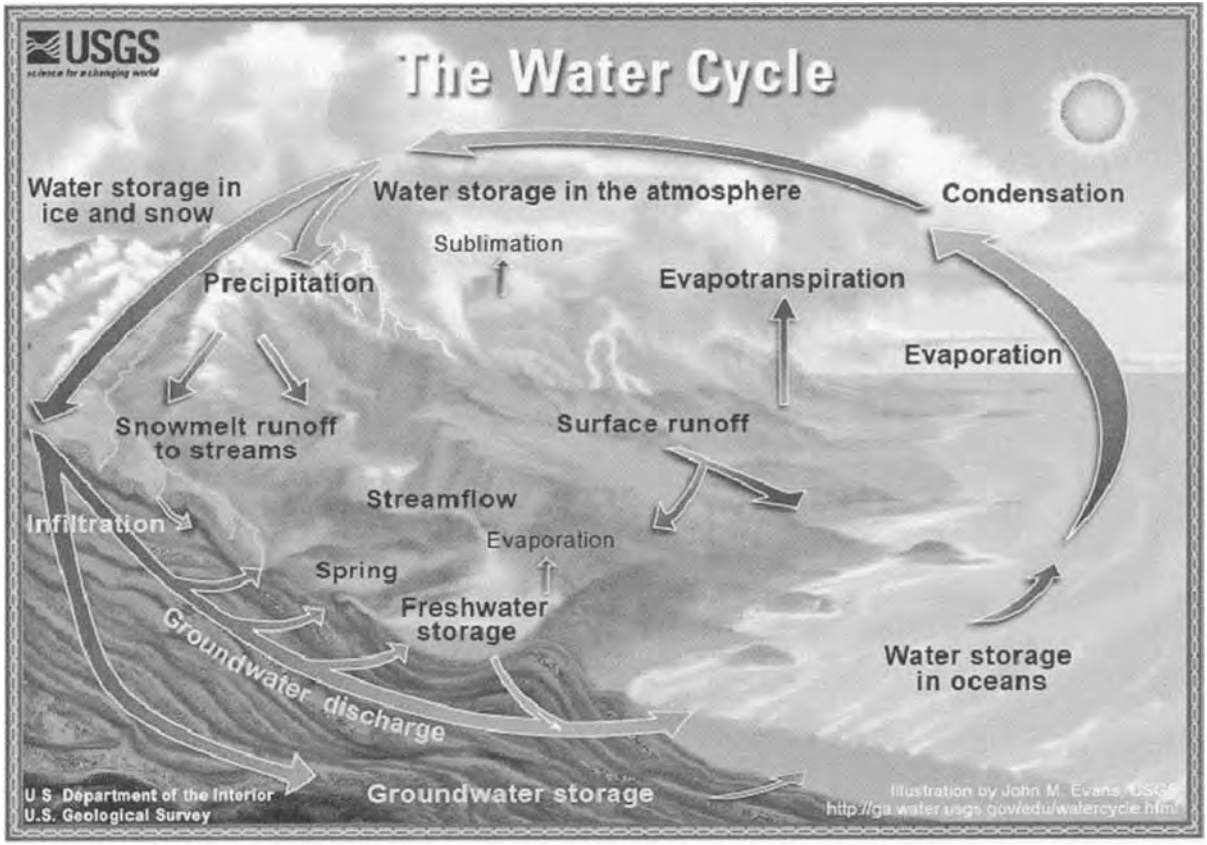


FIGURE 1 The hydrologic cycle (U.S. Geological Survey, U.S. Department of the Interior)

into water vapor. Rising air currents take the vapor up into the atmosphere, along with water from transpiration, which is water transpired from plants and evaporated from the soil. The vapor rises into the air where cooler temperatures cause it to condense into clouds. Air currents move clouds around the globe, cloud particles collide, grow, and fall out of the sky as precipitation. Some precipitation falls as snow and can accumulate as ice caps and glaciers, which can store frozen water for thousands of years. Snowpacks in warmer climates often thaw and melt when spring arrives, and the melted water flows overland as snowmelt. Most precipitation falls back into the oceans or onto land, where, due to gravity, the precipitation flows over the ground as surface runoff. A portion of runoff enters rivers in valleys in the landscape, with streamflow moving water toward the oceans. Runoff and groundwater seepage accumulate and are stored as freshwater in lakes. Not all runoff flows into rivers, though. Much of it soaks into the ground as infiltration. Some water infiltrates deep into the ground and replenishes aquifers (saturated subsurface rock), which store huge amounts of freshwater for long periods of time. Some infiltration stays close to the land surface and can seep back into surface-water bodies (and the ocean) as groundwater discharge, and some groundwater finds openings in the land surface and emerges as freshwater springs. Over time, though, all of this water keeps moving, some to reenter the ocean, where the water cycle is continued.[2]

Water Supply Although the earth's water, in various forms, is constantly progressing through the hydrologic cycle the quantity present in each step along the way remains relatively constant. Table 1 lists the storage locations and the amount that they hold.

The majority of these locations present a usable source of water that can be retrieved and used to meet the daily demands of societies. The sources exhibit great variations in the form, quality, and accessibility of the water. A cursory analysis of the percentages in the table produces the following observations:

- Freshwater resources comprise less than 3% of the total available water supply.
- Water in its solid form represents about two-thirds of the available freshwater.
- Groundwater storage contains the vast majority of the liquid freshwater.
- Lakes store only 1% of the water.
- Streams and rivers hold only 0.013% of the available freshwater.

In general, the analysis supports one general conclusion that the ease of access to available water supplies is inversely proportional to the amount of water contained in the specific resource. The challenge to water-supply professionals is how to manage the resources available in their location, supply a reliable product to their consumers and make certain that there remains a supply for those downstream or next door.

Water Usage Total withdrawals of available water in the United States for the year 2000 were estimated to be 408,000 million gallons per day (MGD) (= 17,875,500 L/s). This usage was apportioned as shown in Figure 2. Withdrawals of the available water supply for

TABLE 1 Earth's water inventory

Location	Volume (million cubic km)	Percent of Total
Oceans	1370	97.25
Ice caps and glaciers	29	2.05
Groundwater	9.5	0.68
Lakes	0.125	0.01
Soil moisture	0.065	0.005
Atmosphere	0.013	0.001
Streams and rivers	0.0017	0.0001
Biosphere	0.0006	0.00004

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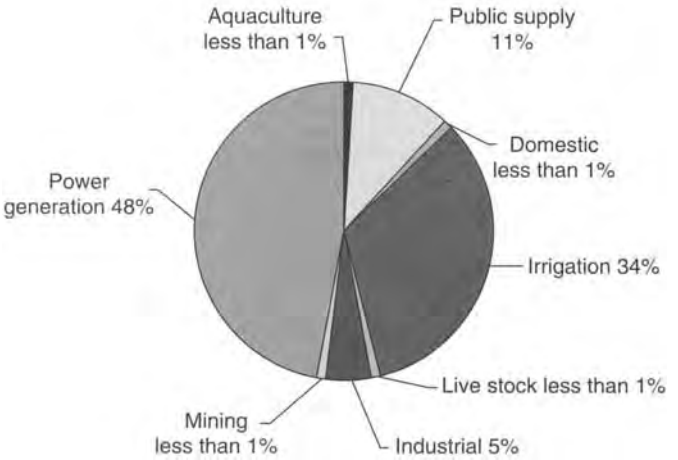


FIGURE 2 Water usage for the United States—2000 (U.S. Geological Survey, U.S. Department of the Interior)

public supply accounted for about 11% of this total or about 44,000 MGD (1,927,800 L/s). The public-supply category includes water withdrawn by public and private water suppliers that furnish water to at least 25 people or have a minimum of 15 connections. Public-supply water may be delivered to users for domestic, commercial, industrial, or thermoelectric-power purposes. Some public-supply water may be delivered to other public suppliers or used in the processes of water and wastewater treatment. Public-supply water is used for such public services (public uses) as pools, parks, and public buildings; or be unaccounted for (losses) because of system leaks or such nonmetered services as firefighting or the flushing of water lines. Some public suppliers treat saline water before it is distributed. However, all public-supply withdrawals in this report are considered freshwater.[3]

Figure 3 shows the steady increase of withdrawals with the growth of the population. Growth in demand without a growth in supply challenges the water-supply professionals

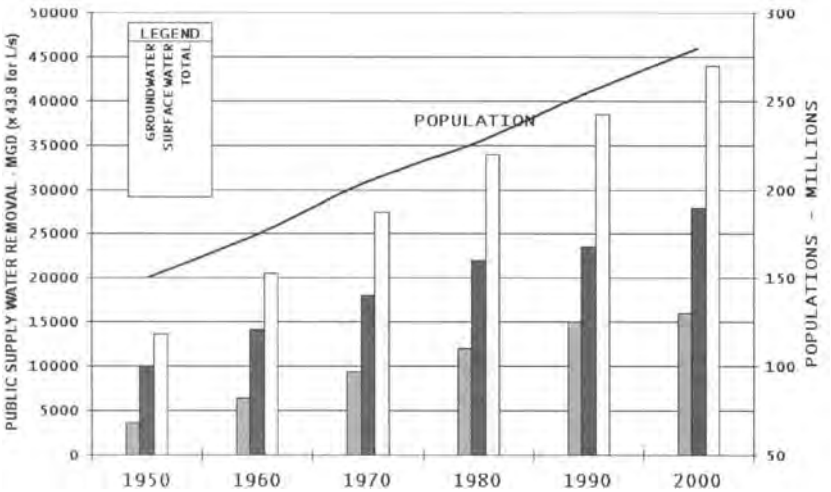


FIGURE 3 Growth of public water usage for the United States—1950 to 2000 (U.S. Geological Survey, U.S. Department of the Interior)

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to both better manage traditional sources and look to nontraditional sources to satisfy users.

The withdrawal of water for public supply still relies heavily on the most convenient source, surface water as shown in Figure 3. Since 1950 the percentage of the total usage for public supply being taken from surface water sources has dropped from 75 to 64%, showing an increased reliance on groundwater. However, the actual withdrawal amount from surface water still has grown from 10,000 to 28,000 MGD (438,100 to 1,226,800 L/s).

Traditional Water Sources The primary source of freshwater for public water supply is the watershed. This is all the water in the hydrologic cycle from the precipitation deposited on the ground until the water flows out into the sea. A watershed is defined as an area of land that drains all the streams and rainfall to a common outlet such as the outflow of a reservoir, mouth of a bay, or any point along a stream channel. The watershed consists of surface water—lakes, streams, reservoirs, and wetlands—and all the underlying groundwater.[3] The Chesapeake Bay Watershed, shown in Figure 4, drains a large portion of the northeast United States to the Atlantic Ocean. But this watershed is made up of many smaller ones each feeding the streams and rivers that eventually feed the bay. Within the watershed the precipitation is carried along in various ways made up of two principle divisions, surface water and groundwater.

SURFACE WATER The most available water in the watershed is that on the surface. Surface water is made up of all the streams, rivers, ponds, and lakes within the watershed.



FIGURE 4 Chesapeake Bay Watershed (Chesapeake Bay Program)

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Reservoirs may be built along the flow path to retain supplies close to points of use. Due to the relative ease of removal surface water is the most relied upon source for freshwater.

GROUNDWATER A significant amount of the precipitation falling into a watershed will infiltrate the ground, where it will enter the local aquifer, a geological formation or structure that stores and/or transmits water. Some water that infiltrates will remain in the shallow soil layer forming an unconfined aquifer, where it will gradually move downhill, through the soil, and eventually enters the stream by seepage into the stream bank. Some of the water may infiltrate much deeper, recharging confined groundwater aquifers. Water may travel long distances or remain in storage for long periods before returning to the surface (see Figure 5).

At any point in the watershed there can be interchange between the surface and groundwater. Surface water is absorbed into the ground as it flows in streams or sits in water in an unconfined aquifer can come to the surface where the land drops below the local water table. Water in a confined aquifer can come to the surface through springs or artesian wells.

EFFECTS OF SOURCE ON WATER QUALITY Although the quality of water obtained from both groundwater and surface supplies varies greatly according to local climatological, hydrological, and geological conditions, some general comparisons can be made between groundwater and surface supplies. Surface supplies generally contain more bacteria, algae, and suspended solids than do groundwater sources and thus require specific treatment to eliminate the resulting turbidity, colors, and tastes. Groundwater supplies generally contain few bacteria and in some instances are pure enough for domestic use without chemical treatment. Frequently, however, groundwater supplies contain significant amounts of dissolved minerals. Depending upon the mineral type, the resulting supply may exhibit such characteristics as exterme hardness, toxicity or undesirable color, odor or taste. Special treatment such as aeration, lime softening, and disinfection may be required to remove such objectionable characteristics.

Developing Sources of Water A popular pump journal placed on its cover the question, "Water: The Next Oil?" The lead article began with the statement, "Take this to the bank: water will eventually replace oil as our planet's core commodity."[4] Also, Ismail

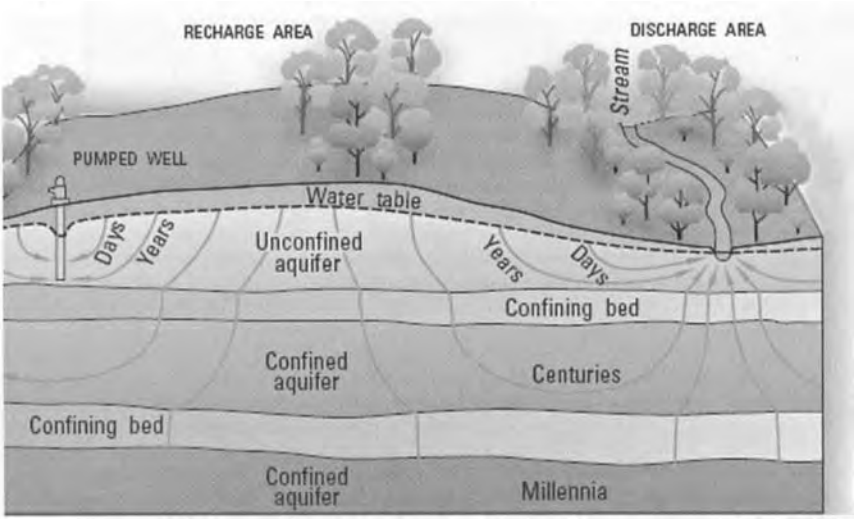


FIGURE 5 Groundwater storage and flow (U.S. Geological Survey, U.S. Department of the Interior)

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Serageldin, World Bank Vice President for Environmental Affairs in 2000, was quoted as saying, "The wars of the twenty-first century will be fought over water." With this outlook on the growth of demand for water and the relatively constant supply in each of the steps of earth's water inventory, the need to develop resources other than the watershed has caused a look to the larger but lesser used sources.

OCEANS At over 97% of the available water, the oceans represent the promise of an inexhaustible water supply. The problem is to make the water potable at a cost and quality acceptable to the market. Currently, two methods exist for the separation of the dissolved salts from the feedwater, thermal, and membrane technology.

In the thermal process, essentially distillation, the feedwater is evaporated and then condensed. The energy cost of distillation is one of its major drawbacks. The most common methods of distillation include vapor compression (VC), multistage flash (MSF), and multieffect distillation (MED). The vapor compression process involves evaporating the input water and then compressing the vapor. The vapor is then condensed, and the heat released is used to evaporate the input water. In the multistage flash process, saline feedwater is heated and the pressure is lowered so that the water "flashes" into steam. This process constitutes one stage; then comes a number of stages in series, each of which is at a lower pressure. In multieffect distillation, there is a number of evaporators in series, and vapor from one series is used to evaporate water in the next effect.

In membrane technologies pressure is applied to force a solution through a membrane. The membrane is semipermeable, meaning it is designed to allow the passage of water while trapping the impurities. The most common membrane technology is reverse osmosis. The name comes from the fact that it reverses the natural process of osmosis where the solution passes through the membrane from low to high concentration. The reversal of the natural process is accomplished by applying pressure to the high concentrate side of the membrane above the natural osmotic pressure. Pretreatment of the feed solution is necessary to eliminate solids and microorganisms that would otherwise impede the process.

SOLID WATER Where accessible, ice in glaciers and ice packs, representing two-thirds of the earth's supply is an attractive source of freshwater. In the United States, Alaska represents the most promising resource. There are an estimated 100,000 glaciers in Alaska covering approximately 28,800 mi^2, or 3% of Alaska. This is 128 times the area covered by glaciers in the rest of the United States, storing three-fourths of all freshwater in Alaska. Many of the state's 100,000 glaciers are inaccessible, practically speaking, but where they can be accessed the State of Alaska regulates the removal of ice. People or companies wanting to harvest Alaska's glaciers must apply for a permit if they plan to take away more than 20 tons a year. Glacier ice is now harvested from glaciers that calve into tidewater. The state permit grants a right to the water that makes up the ice. Not every glacier is fair game, and a permit isn't to be granted if the harvest would interfere with the rights of other users, such as wildlife, fish, and humans.[5] Glacier ice harvesting is a successful business in Alaska, a 1-ton block of clean glacial ice sells for $500 wholesale.[6]

WATER DEMAND

"We never know the worth of water 'til the well is dry." (Thomas Fuller)

Water Consumption The primary design parameter in the development of a public water supply system is the capacity of the system needed to meet the demand of the service area. This is dependent on many variables including the population to be served, usage habits and growth estimates for the utility. The consumption rate is normally expressed as gpcd (gallons per capita per day) in U.S. customary units or lpcd (liters per capita per day) in SI units. The world water consumption per inhabitant varies greatly; in the United States the average is 132 gpcd (500 lpcd), in western Europe 40 gpcd (150 lpcd) and in Africa 13 gpcd (50 lpcd).[7] Even within a country the rates can vary greatly. Using the data from Figure 3 the per capita consumption curve shown in Figure 6 can be drawn.

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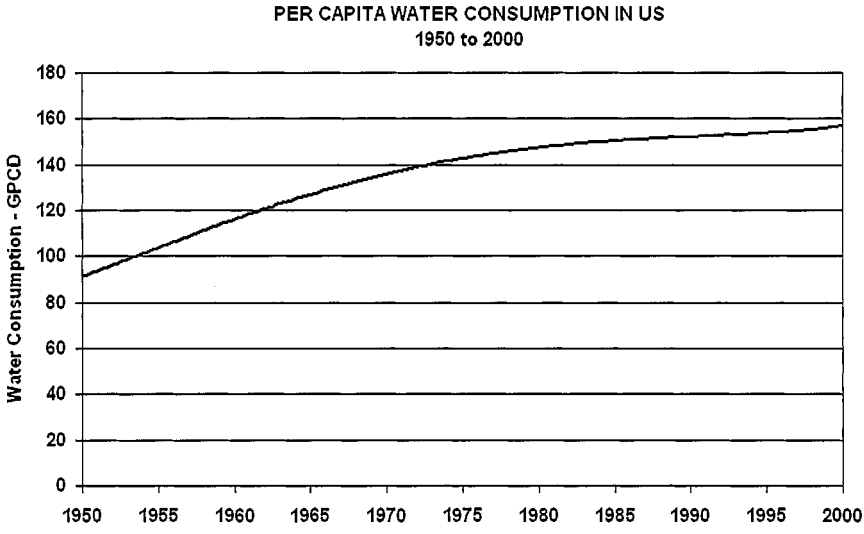


FIGURE 6 Per capita water consumption in the United States (U.S. Geological Survey, U.S. Department of the Interior) (liters = gallons × 3.78)

The consumption rate typically used for design purposes is 175 to 200 gpcd (670 to 760 lpcd). This range is supported by the chart above; however, this should be compared to the historical data for the City of New York shown in Figure 7. This demonstrates the effects of an effective water conservation education effort resulting in not only a reduction in per capita usage but in real total usage even with a population increase.

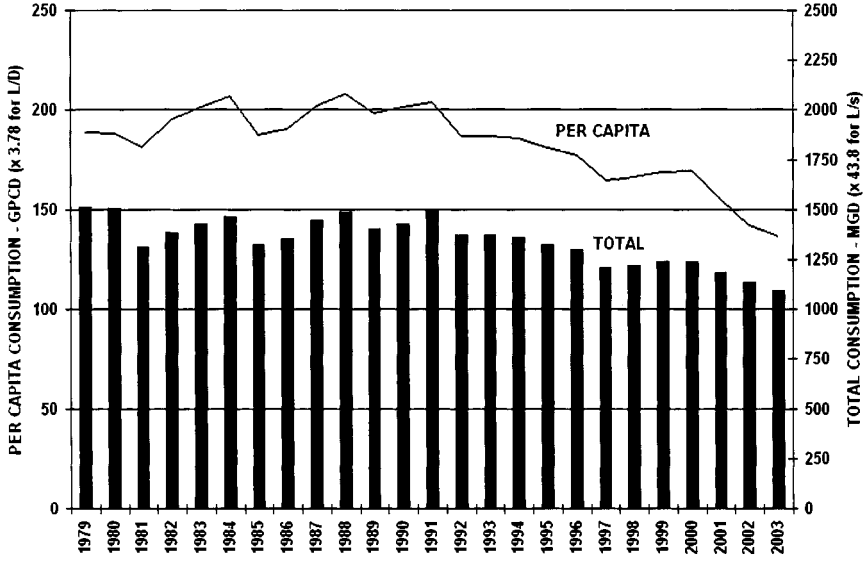


FIGURE 7 Total and per capita water consumption in NYC (The City of New York, Department of Environmental Protection)

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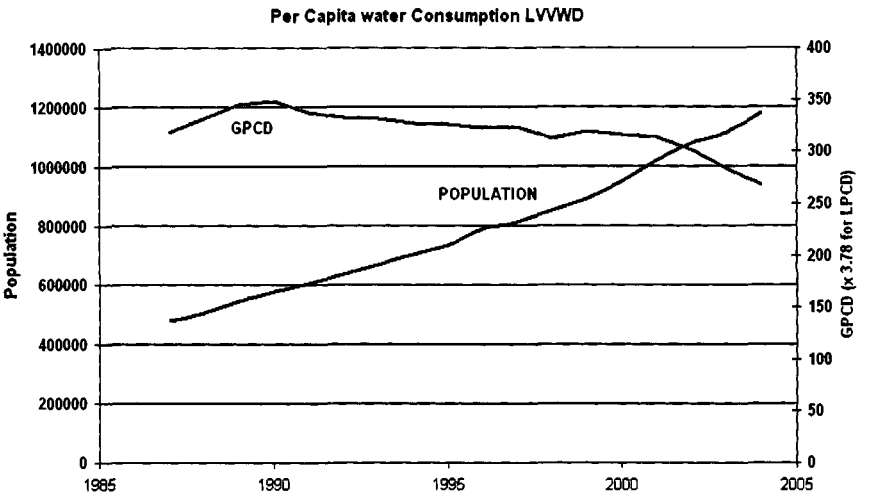


FIGURE 8 Per capita water consumption Las Vegas (Las Vegas Valley Water District)

Geographical and climatic effects also exert a strong influence in usage habits. In this regard, the data for the Las Vegas Valley Water district shown in Figure 8 should be considered. This utility serves a growing population in an arid location with higher per capita usage than New York City. This also shows the effect of conservation efforts as the total water usage has only doubled over the recorded period while the population has tripled.

Location-specific effects and usage trends like these are difficult to predict and show the extensive nature of the preliminary studies on plant and system size.

Demand and Daily Fluctuation The average consumption rate is the primary sizing factor but does not provide the full set of parameters for the system design. There are large fluctuations in the demand rate based on season, month, day of the week and hour of the day. The relationship of these parameters to the base average is critical to the ability to meet all demand possibilities. The usage and demand rates are as follows:

1. **Average daily demand** It may be expressed as gallons (liters) per capita per day or million gallons (liters) per day (the total year's usage divided by 365). The average daily demand rate is generally used as the yardstick by which all other demand rates are measured.
2. **Maximum monthly demand** This is the average daily use during the month of greatest consumption. It is determined by dividing that month's usage by the number of days in the month.
3. **Maximum daily demand** This is the total amount of water used during the day of heaviest consumption in the year. Experience shows this demand rate occurs on from three to five consecutive days during the year.
4. **Maximum hourly demand** This is the rate of use during the hour of peak demand on the day of maximum demand. This demand rate normally establishes the highest rate of design for distribution systems and peak pumpage.

Of these rates, the most difficult to determine accurately is the hourly demand because it is a nearly instantaneous snapshot of all the components of the system including pumping rates and storage levels. The development of sophisticated monitoring devices like Supervisory Control and Data Acquisition (SCADA) systems have made this a much simpler and accurate process. Figure 9 shows the hourly fluctuations for the Las Vegas Valley Water District.

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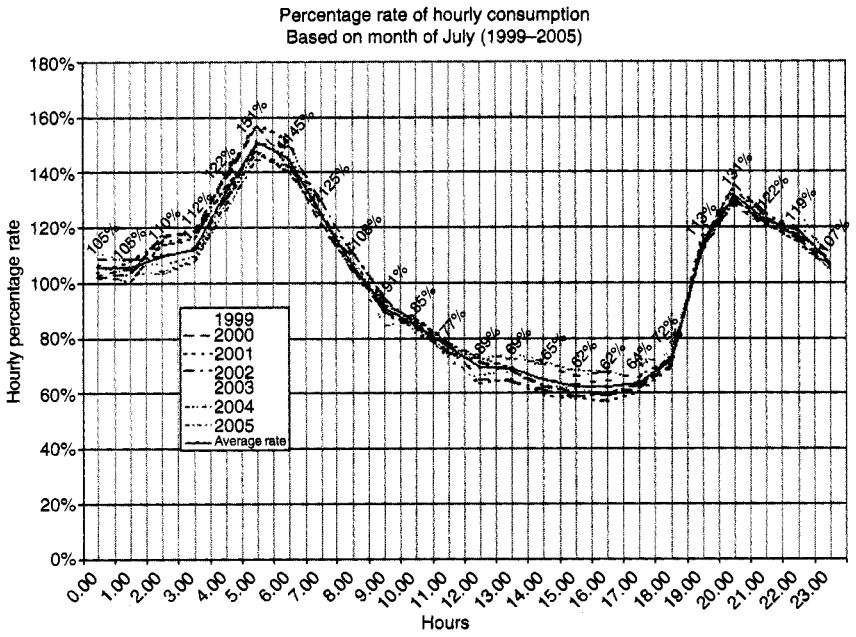


FIGURE 9 Hourly demand fluctuations (Las Vegas Valley Water District)

Demand Rates as Basis of Design The fluctuations of demand expressed as a percentage of some base value must be developed and used as the base for the system design. Performing a sort of worst-case analysis can do this. Historical data having been used to establish the four demand rates listed above, these can now be used for the process. In the example shown the base value chosen is the *average daily demand*. This is the most commonly used parameter although another may be used if shown by the data gathering activity to be more applicable to the usage pattern. The other three demand rates are then expressed as a percentage of this value. The analysis proceeds by manipulating the data as if the maximum day occurs during the maximum month and further as if the maximum hour occurs during the maximum day and month. The results of such an analysis are shown in Figure 10 based on data provided by the Las Vegas Valley Water District. The design rates suggested in the chart are averages of the results. The large variation in the hourly results may lead to a higher value than the 342% being selected by the designers.

WATER SUPPLY

"Pure water is the best of gifts that man to man can bring." (Anonymous)

Water Supply Basics Meeting the demand for water involves one or more steps in a set of processes that define the full scope of water supply. Not all systems will require every step and not all systems will utilize the same technology to achieve the end result of that step. The basic functions of a water supply system are these:

- *Screening*: Prior to entering the actual treatment process there may be a rough screening of the raw water to remove floating debris and large particles that could damage or clog equipment.

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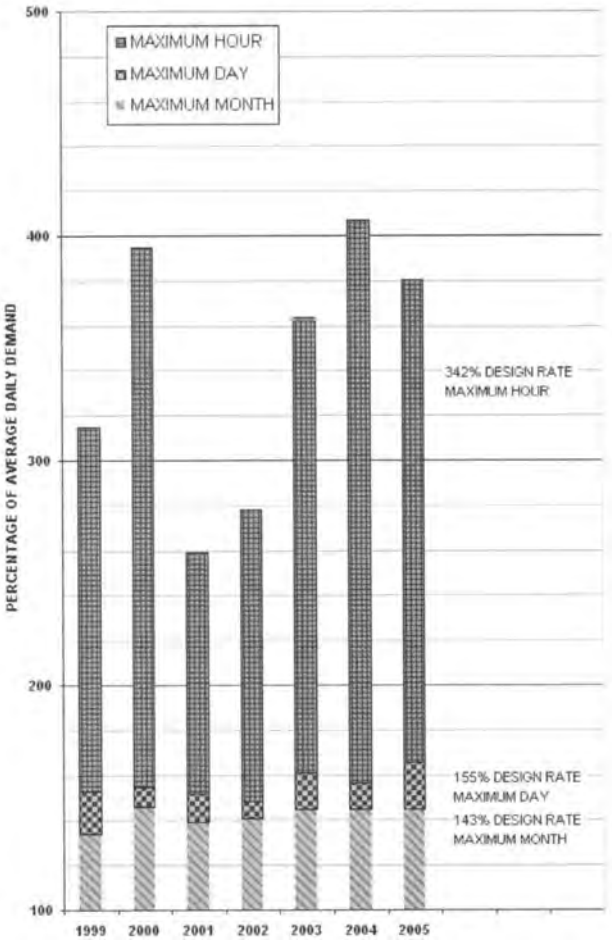


FIGURE 10 Comparison of maximum month, day, and hour to average daily demand (Las Vegas Valley Water District)

- *Acquisition*: This is the withdrawal of the feed supply, usually termed "raw water," from the source.
- *Transmission*: Depending on the location of the source it may be necessary to transport the raw water over significant distances to the plant.
- *Coagulation*: This is the collecting of minute particles of a solid dispersed throughout a liquid into a mass followed by the precipitation or separation of the solid mass from the liquid.
- *Clarification*: Also called *sedimentation*, this step requires the retention of the treatment flow in a large tank called the clarifier or sedimentation basin. This is to allow the coagulants to settle to the bottom where they can be removed as sludge. Exit from the clarifier is generally over a weir so only the water furthest from the sediment goes on into the system.
- *Filtration*: After clarification the water passes through a filter to remove unsettled sediment and any other particles. Several different filtration technologies exit including sand filters, pressure filters, and membrane "ultrafilters."

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- *Disinfection*: A final step in the purification process, disinfection is used to destroy any pathogens remaining after filtration. Forms of disinfection include chlorination, ozonation, and UV radiation.
- *Fluorination*: The addition of fluorides to the water is accomplished by the addition of dihydrogen hexafluorosilicate. This compound decomposes in water, releasing fluoride ions.
- *Storage*: Following the disinfection process the water is held in temporary storage to complete the action and then goes on to finished water storage.
- *Distribution*: The final step in the water supply system is the distribution of the water to the user.

Source of Water Quality The levels of treatment depend largely on the source of the raw water. The quality of the water can vary greatly based on the local climate, watershed and geology. Surface water problems influencing the treatment can include solids; suspended and dissolved; color, odor, microorganisms and taste. Groundwater may be pure enough for domestic use without chemical treatment; and domestic withdrawals amount to about 3000 MGD (131,000 L/s), with much being consumed as pumped. Groundwater problems could include hardness, acidity, and dissolved minerals that can cause objectionable taste and odor.

Pump Services for Water Supply The reliable supply of high quality water in sufficient quantities to support the lifestyle of the user is one of the engineering hallmarks of our society and the heart of the system, from source to destination, is the pump. At each step of the process, pumping machinery of all types transforms raw energy into the flow necessary to feed the system.

Raw Water The delivery of the primary feedstock into the treatment process begins with the extraction of the raw water from the source. The raw water system is highly dependent upon the source of the water, the style of the intake structure and the location relative to the treatment plant. The primary determinant of the raw water pump station design is source of the water, surface water from a lake, river or reservoir, or groundwater from a well. Surface water sources can have a further effect on the pump selection. Where the source of water is from a reservoir at an elevation above the pump station the normal selection would be a horizontal centrifugal pump of the general design, shown in Figure 11. A typical station layout for a pump of this type is shown in Figure 12. The basic reason for this style of pump being used is that the elevation of the source water above the pump provides the energy (net positive suction head) to fill the pump and support the energy requirements of the impeller. The station and the pump picture both show the bottom suction feature that simplifies the piping of the station.

Other surface water sources such as lakes or rivers place the water below the surrounding grade level where it would take a significant amount of costly excavation to place the pumps below the water level. For these sources pump stations are typically built along the shoreline or riverbank with an intake structure extending out into the water. If horizontal pumps were to be used then a priming system to evacuate the pump and lift the water into the impeller prior to starting would be required. To avoid the extra cost and maintenance of this system vertical pumps are used with the pump suspended down into the water.

Figure 13 shows a vertical raw water pump station where the design not only allowed for the pumps to be suspended in the suction source to provide the NPSHR for the impeller but also to accommodate a drop in the suction source water level. This installation was also designed to allow the number of stages in the liquid end pumps to be increased and the pumps lengthened if the level of the suction continued to drop due to climatic conditions and upstream usage.

Figure 14 shows a typical vertical raw water pump station. The only part of the pump in the picture, barely visible behind the piping, is the discharge head. This arrangement can save floor space in the pump station but can be more costly to maintain.

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FIGURE 11 Horizontal, axially-split, single-stage, bottom-suction sump (Flowserve Pump Division)

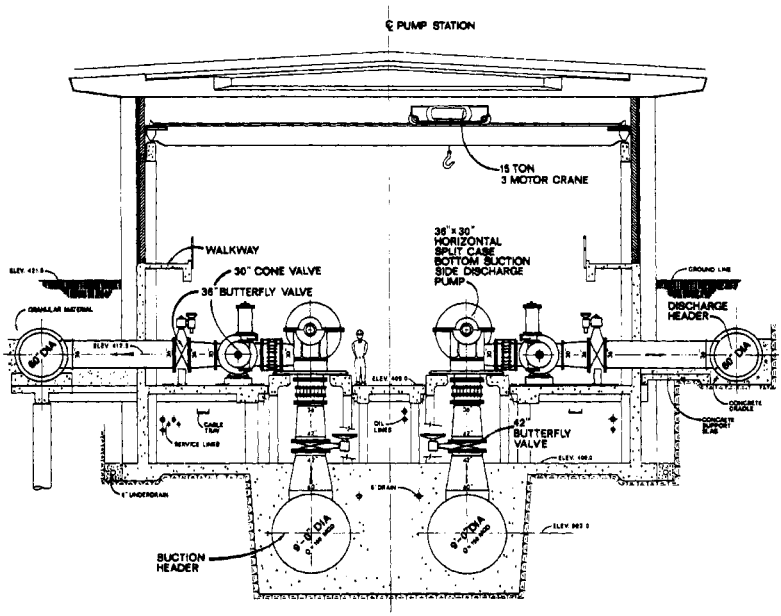


FIGURE 12 Typical section of the Forney raw water pumping station in Dallas, showing pump suction and discharge piping (1 in = 2.54 cm; 1 ft = 0.3048 m) (URS/Forrest and Cotton)

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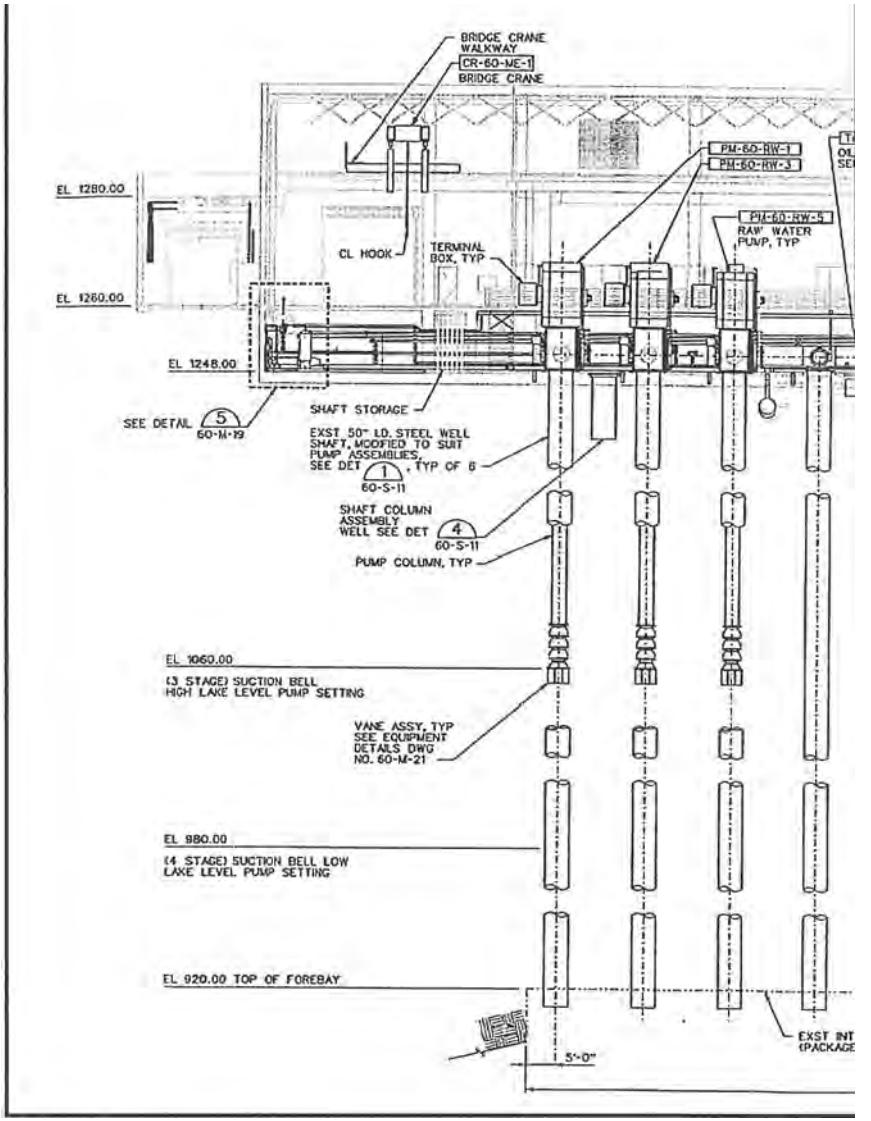


FIGURE 13 Typical section of a raw water pump station using vertical pumps (Southern Nevada Water Authority)

Groundwater is becoming a more attractive source for raw water as the supply of readily available surface water shrinks with demand. It will be recalled from the water inventory that fresh groundwater is much more abundant than surface water, yet it represents only 25% of the public water supply. Also, groundwater frequently requires less treatment than surface water. Although some groundwater sources are available at the surface in the form of artesian wells or springs, aquifers are chiefly accessed through wells drilled or dug down below the local water table. The pumping rate of wells is determined by the aquifer

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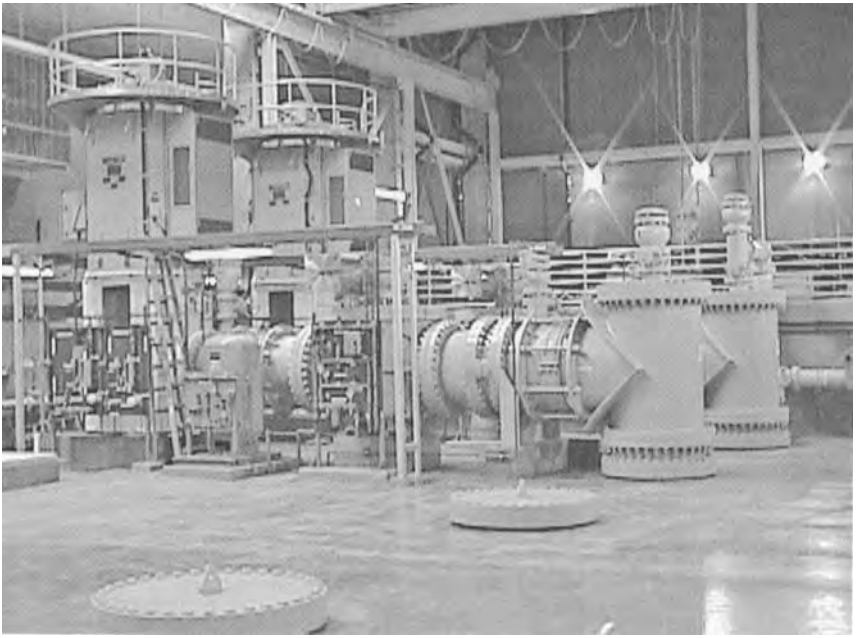


FIGURE 14 Typical raw water pump station with vertical pumps

activity, especially the recharge rate. If a well is pumped at a faster rate than the aquifer around it is recharged by precipitation or other underground flow, then water levels around the well can be lowered. For this reason, wells are drilled well below the "water table" to provide some protection against the lowering of the supply. Deeper wells also provide for some ready storage in the casing above the pump element.

Pumps for public supply groundwater are almost always of the multistage vertical turbine design. Smaller private wells may use the jet pump design, but the flow rates and lifts are limited (Chap. 7). The depth of public supply wells has been increasing recently with depths of 1000 ft (305 m) now becoming a more frequent occurrence. Depending on the geology of the field wells of this depth can present significant problems with straightness, both in drift (deviation from vertical) and spiral.

Vertical turbine pumps (VTPs) for deep wells come in two distinctly differing designs. The defining characteristic is the location of the drive motor. The actual pumping element of both types is essentially the same; a set of stages consisting of casings (bowls) and impellers mounted is series on a single shaft. The power can be delivered to the pump shaft either from below the element as in a submersible pump, so called because the motor is submerged, or from above as in a lineshaft pump, so called because the power is delivered from a motor mounted on the surface through a slender shaft.

Figure 15 shows a typical lineshaft design consisting of the submerged pumping element, the column pipe/shaft assembly, the discharge head and the surface mounted motor. The lineshaft pump design has been used for this application for many years, in fact the VTP design originated with well pump applications, but the deeper settings have associated problems that must be considered for each application. As with all pump designs the impeller creates an axial force that is applied to the pump shaft. In the case of lineshaft VTPs this axial force is transmitted to the surface and supported by a thrust bearing in the motor. The result of this force applied to a slender shaft of significant length is a growth in the length of the shaft. In deep-well applications this elongation can be several inches and special designs for the casings and impellers must be used to allow for this

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FIGURE 15 Vertical turbine, lineshaft-driven pump (Flowserve Pump Division)

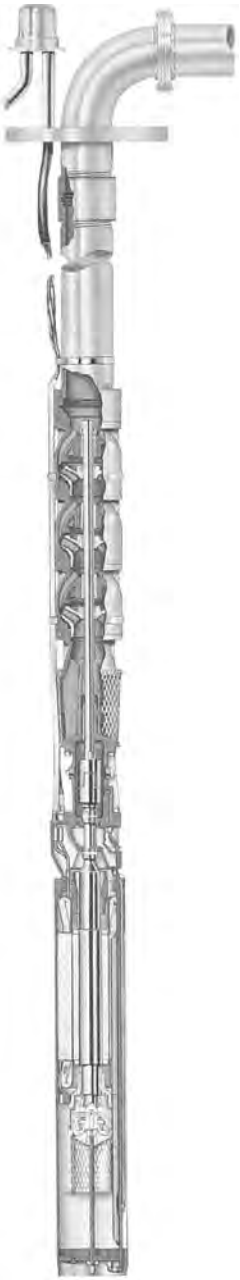


FIGURE 16 Vertical turbine, submersible motor driven pump (Flowserve Pump Division)

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movement of the rotor within the pumping element. The calculation of the shaft elongation is simple, as seen by the following formula:

$$Elong. = (TBH \times L_{col} \times K)/(A_{shaft} \times E)$$

where TBH = total "bowl head" = total developed head of multistage pump
L_{col} = length of column
K = impeller thrust constant [8,9]
A_{shaft} = shaft cross sectional area
E = shaft modulus of elasticity

This calculation will closely represent the elongation of the shaft itself, but other factors come into consideration. The column is subject to a portion of the weight of the water in the pump and will elongate some itself reducing the relative elongation of the shaft. Also, depending on the depth to the water level the head produced by the pump can vary affecting the value of the TBH.

The second variation of the VTP design and one that is growing in popularity with users is the submersible or submerged motor design. In this, as the name implies, the motor is located down in the well along with the pumping element, as shown in Figure 16. The motor is located at the bottom of the assembly with a special connector between it and the first stage pump case. A long cable from the surface transmits energy to the motor and the water is carried to the surface through standard, threaded pipe. This VTP variation, since it eliminates the lineshaft and absorbs the thrust in the motor bearing, does not require special case and impeller designs. The elimination of the lineshaft also reduces the number of maintenance parts and makes the unit less susceptible to bearing rubs and wear that can come from less than straight wells. As with any design choice, there are trade-offs that must be considered when deciding on the course to take. The location of the motor can cause higher maintenance costs if problems occur.

Hydraulic Gradient Whatever the source or pump style the primary task of the raw water pump is to withdraw the water from the source and deliver it to the treatment plant. Often this is a simple task of lifting the water to the plant where much of the process is performed using gravity flow. Raw water pumps in this service may be called *low lift* because the head required to feed the plant is not high. In other cases, such as river water intakes, the source may be at a lower elevation and the plant located at a higher elevation. In cases like this, the raw water system must be designed carefully to be certain the pumps will meet all the needs of the plant. The development of a hydraulic gradient plot (described in Section 11.1) is one method used to size these pump stations, sometimes referred to as high lift. A plot of hydraulic gradient for one system is shown in Figure 17. In this plot, the X-axis represents the distance from the source, a river pump station; and the Y-axis is the elevation of the transmission line from source to the delivery point, a reservoir at the treatment plant. Two lines are plotted on the chart. First is the profile of the raw water transmission line as it passes over the terrain. The upper line shows the head needed in the system to reach the reservoir with a positive head margin. The most important concept to be derived from this is that the head developed by the pumps must meet the difference between the minimum source level and the hydraulic gradient line for the selected line size at the entrance station to the pipeline. It cannot be stressed too strongly that the hydraulic gradient line must be above the system profile at all points. One pitfall of using a system head calculation that looks only at the source and delivery levels and the line friction without considering this chart was a real experience for one system designer. The pump station was complete and the day came to push the button. Everything seemed to go well, but no water was being delivered to the plant. The pumps were running, power was being consumed but the discharge gauge was reading much higher than expected. The problem was that between the pump station and the plant was a mountain, and the elevation of the pipeline as it passed over the mountain imposed a head on the pumps greater than that available, even at zero flow rate.

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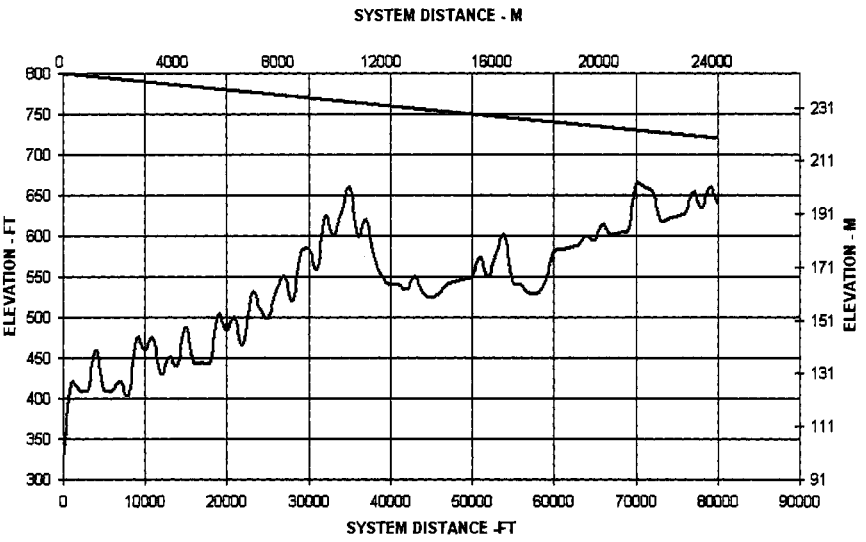


FIGURE 17 Typical hydraulic gradient plot

Transmission At first it was simple: population centers grew next to the best source of water, and when treatment plants were built they were also close to the water source. Cities grew and in many cases outgrew the capability of the local source to meet all the water needs. The need arose to locate, develop, and transmit water from new sources, often over long distances to the point of use. Many times the source of water was at a higher level than the point of use and gravity alone could deliver the flow. This is as old as the aqueducts of the Roman Empire and as new as New York City's Tunnel Number Three. Unfortunately, this is not always the case. Referring back to Figure 17, consider the profile being both longer and having multiple peaks. This is the prospect facing many utilities today, and the system needs far exceed the capability of a single pump station design. It becomes necessary to break up the raw water transmission line into two or more segments, adding a booster pump station at the junctions. Raw water booster pump stations can take many forms, but they commonly consist of a surge tank as a delivery point, a forebay to serve as the pump suction source, and the booster pumps. The layout of a typical raw water booster station using horizontal centrifugal pumps is shown in Figure 18.

Chemical Feed Pumps At the inlet to the treatment plant and continuing throughout the process, chemicals are added to the water for various purposes. Among these chemicals are the following:

POTASSIUM PERMANGANATE It is added to the water coming into the plant to remove iron, manganese, and objectionable tastes and odors. Because raw water is often drawn from below the surface of the source dissolved oxygen is low. When the oxygen level is low, iron and manganese are kept in solution. When iron and manganese are oxidized, they become insoluble in water and can then be removed by sedimentation and filtration. Potassium permanganate is also used to oxidize the organic contaminants that cause taste and odor problems.

ALUM Alum is added to the incoming raw water to support coagulation and flocculation reactions that bind together very fine suspended particles into larger particles that can be removed in the settling and filtration steps. The end result is to remove objectionable color and turbidity (cloudiness) from the drinking water.

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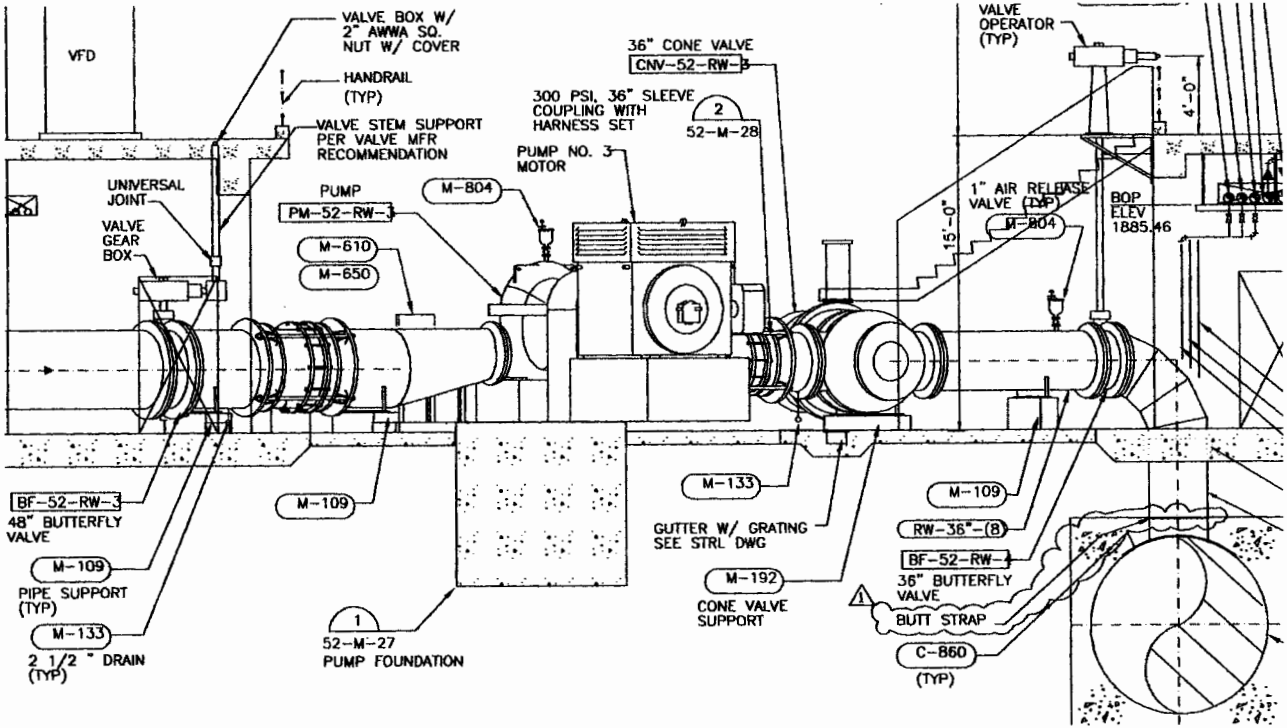


FIGURE 18 Typical section of a water booster pump station (Southern Nevada Water Authority)

HYDRATED LIME Hydrated lime is an alkaline compound that is added to the water for pH adjustment. Since filter alum is an acidic salt, it lowers the pH of the water, and lime is used to neutralize this effect.

CHLORINE It is added to water to disinfect it or improve its quality. Chlorine destroys pathogenic microorganisms, oxidizes undesirable elements, such as iron and manganese, and reduces some tastes and odors. After these needs have been met, some residual chlorine remains in the water to protect it from further contamination until it reaches the customer's tap.

PHOSPHATES Phosphates are added to the water to isolate iron, manganese, and calcium to keep them in solution. It also prevents corrosion of water lines by coating them as the water passes through. Some utilities use this to prevent the leaching of lead from older distribution systems and domestic plumbing.

FLUORIDE Fluoride is added to water following the disinfection process for the purpose of preventing tooth decay. Fluoridation is usually accomplished by the addition of dihydrogen hexafluorosilicate, which decomposes in water, yielding fluoride ions.

CARBON Carbon is added as slurry to remove objectionable tastes and odors. The tiny carbon particles are tremendously absorbent, like sponges. The longer they stay in the stream, the more thorough their dispersion; the better the job they do in making the water pleasant and palatable.

POLYPHOSPHATE Polyphosphate, the last additive, is a material that coats the surfaces of pipes as it passes through them, and prevents the lead in old plumbing from leaching into the drinking water.

All of these chemicals must be added at the correct step in the process and at a closely controlled rate. The pumps used for this purpose are normally of the positive displacement type, so that the discharge rate can be controlled by the speed and operating time. The specific type of displacement pump used is a matter of preference, the plunger, diaphragm, and peristaltic pumps being frequently used. Examples of each type can be found in Chap. 3.

Filter Pumps Conventional rapid gravity sand filters with their multilayer technology are a low-pressure process and depend only on gravity flow from the clarifier. Some treatment plants use a pressurized form of these filters, which work on the same principle as rapid gravity filters, except that the filter medium is enclosed in a vessel and the water is forced through it under pressure.

Newer filtration processes also use media that depend on pressure differential to force the flow through the media. The principal new methods are membrane technologies such as reverse osmosis, mentioned earlier. In a reverse osmosis system, pressure is applied to water on one side of a semipermeable membrane. The process is termed reverse because the natural migration of water in an osmotic cell would be from the area of lower to higher concentration. Pressure must be applied to the "dirty" side of the membrane to produce the reverse flow and produce clean water. For instance, the osmotic pressure of ocean water is about 27 atm or 27 bar (400 psi). Reverse osmosis desalinators use pressures around 50 atm (750 psi) to produce freshwater from ocean saltwater. Multistage pumps of either the horizontal-type or closed-suction vertical "can" pumps are used in the service.

Wash Water Pumps Conventional sand filters do rely on pumps for one important function, washing the trapped solids from the filter media. Wash water pumps may be either horizontal or vertical depending on the configuration of the wash water tank. During normal operation a portion of the filtered water is diverted to a wash water holding tank. At regularly scheduled intervals, the wash water pumps are energized and pump water through a distribution system in the bottom of the filter causing an upward flow through the media and out into a collection system. The backwash rate is determined by the filter

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manufacturer and is based on the type of media and area of the filter surface. Typical backwash rates are 15 to 20 gpm/ft² of bed area (37 to 49 m³/h per m²), and backwash should be carried out for 8 to 10 min. In some systems air is first injected into the lower layer of the filter to agitate and loosen the filter media to make the backwash process more effective. Pump selection is important to the stability of the filter media. Backwash services generally require high flow rate and low head; so, pump performance curves must be selected to prevent any unsteady hydraulic behavior that might lead to the upset and mixing of the layers of the filter media.

High Service Pumps High service pumps are the final pumps at a water treatment plant placing the water from the clearwell into the distribution system. Either horizontal or vertical pumps can be used, based on the system requirements and the placement of the clearwell within the treatment plant. Horizontal pumps are normally used when the required head is low enough to be supplied by a single-stage pump. Pump stations for horizontal pumps must be located so that a positive head is delivered to the pump inlet that is sufficient to meet the NPSH required by the pump. Often this means excavating so that the pumps are below the existing grade level as shown in Figure 19.

In this pump station the suction well is pressurized by the level of the water in the clearwell. The bottom suction horizontal pump allows for a simple piping arrangement on the station main floor with all maintenance performed in accessible areas.

Vertical pumps are typically used when heads required are more than that available from single-stage pumps. The flexibility of the vertical, multistage design adapts well to the varying demands placed on high service pumps. A typical vertical pump installation is shown in Figure 20.

Vertical pumps also allow the station designer to maintain a simple piping arrangement. The one drawback to this design is the removal of the pump for maintenance. The station must be designed with sufficient overhead space and crane capacity to remove the pump from the suction well. Costs related to maintenance can be higher than those for horizontal pumps.

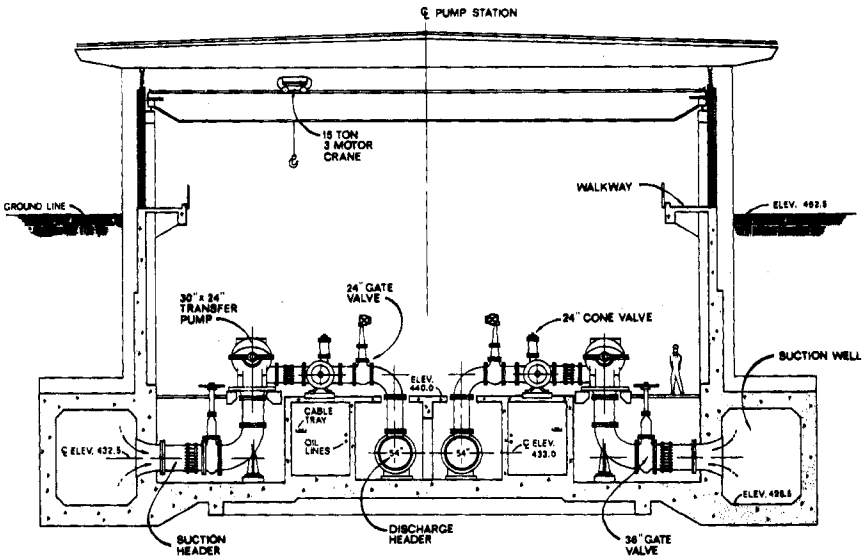


FIGURE 19 Typical section of the high service pumping station of the East Side water treatment plant in Dallas (1 in = 2.54 cm; ft = 0.3048 m) (URS/Forrest and Cotton)

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FIGURE 20 High service pump station with vertical pumps (Flowserve Pump Division)

The design conditions for high service pumps are not easily determined. In some smaller systems all the user needs can be met by a single set of pumps operating at a constant speed with demand variations being met by selecting the number of pumps on line and by elevated storage tanks and water towers in the system. As systems become more complex and user patterns more varied the nature of the analysis needed for the high service pump sizing becomes much more complicated. Meeting the peak hourly demand, which may exceed 350% of the average daily demand rate (see Figure 10) and be sustained for several hours (see Figure 9), calls for a balance of treatment plant sizing, system storage and pumping station capability. The primary choice required is the balance between the portion of the peak hourly demand to be met by direct pumping from the treatment plant and that supplied from storage. This choice not only affects the high service system sizing but, projected back upstream, the design capacity of the treatment plant also. The need for accurate input into this decision cannot be overstressed. The most common process for providing this information has been a system head analysis looking at all of the possible variables. (See Sections 11.1 and 11.2 in this book.) The result of one such analysis is shown in Figure 21. In this example, three system head curves are shown for the "equalization" hour, demand and design rates equal, "peak" hour, demand above design, and "refill" hour, demand below design. Superimposed on these three curves are additional curves reflecting the performance of a set of six pumps driven by two-speed electrical motors.

An upgrade to the pumping system to provide more complete system coverage would be to step up from two speed motors to a variable speed system. This could be done to one, two, or all of the pumps and makes the system much more flexible, allowing more than one selection to meet the critical operating conditions. By careful selection of the pump design conditions the designer can cover all of the possible combinations needed to meet the system demand. This type of analysis is good for one given design and, while the numbers of pumps can be varied with relative ease, requires a redevelopment of the system data for

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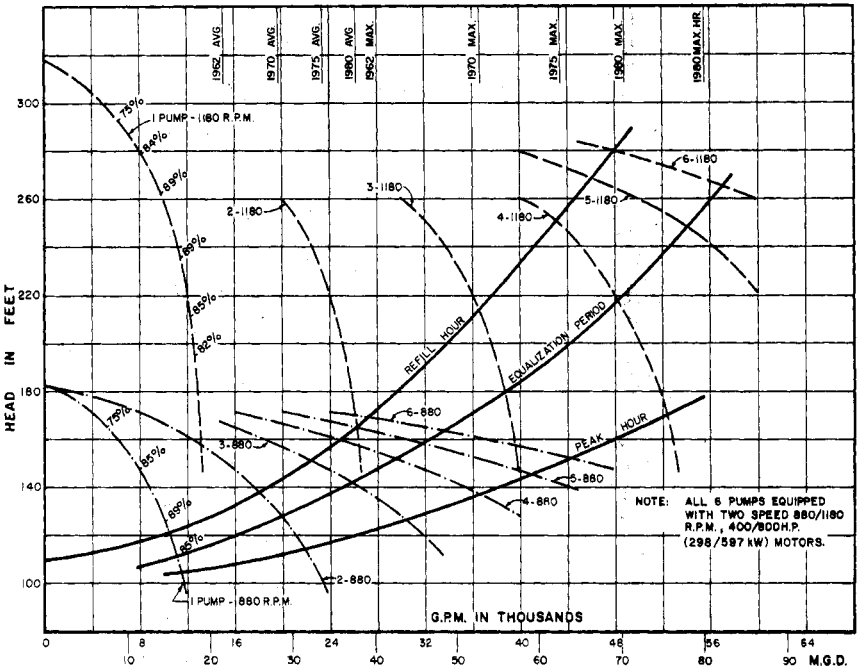


FIGURE 21 Typical system head curves for high service pumpage to water distribution system (1 ft = 0.3048 m; 1 gpm = 0.227 m^3/h)

any proposed system modifications. The importance of this information and the complexity of the analysis along with the need for flexibility in quickly evaluating system options have led to the development of computer software for distribution system design. As a quick search of the Internet will confirm there are many choices available to the system designer. Some programs are available free of charge and provide a basic analysis capable of an extended period hydraulic analysis package which can perform the computations described in Section 11.1, including the ability to:

- handle systems of any size
- compute friction head loss using the Hazen-Williams, Darcy-Weisbach, or Chezy-Manning formulas
- include minor head losses for bends, fittings, and so on
- model constant or variable speed pumps
- compute pumping energy and cost
- model various types of valves including shutoff, check, pressure regulating, and flow control valves
- allow storage tanks to have any shape (e.g., diameter can vary with height)
- consider multiple demand categories at nodes, each with its own pattern of time dependence
- model pressure-dependent flow issuing from emitters (sprinkler heads)
- base system operation on simple tank level or timer controls as well as on complex rule-based controls

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FIGURE 22 Exterior and interior of a typical community booster station (Flowserve Pump Division)

Evaluation of the various packages is beyond the scope of this document and left to the user. The basic functions appear to be the same with many options in the area of output and graphics.

Distribution Booster Pumps The addition of branches to a distribution system (cf. Section 11.2) as it reaches out to new housing developments often results in a loss of service pressure. The utilities goal should be to maintain normal service pressure delivery to the user is between 40 and 90 psig (2.8 and 6.2 barg). Where the existing system falls below this value a booster pump station must be installed. Typically booster stations include a storage reservoir to accept the incoming flow and a set of pumps to supply the new branch system. The pumps are controlled by the upstream pressure and meet the varying demand through variable speed or multiple pump operation. The design of these stations is often very sensitive to the location. Figure 22 shows the exterior design of one such station that was designed to conform to the general local architecture. An interior view of this building is also shown in Figure 22.

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SECTION 12.2
SEWAGE TREATMENT

H. H. BENJES, SR.
W. E. FOSTER
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Sewage is defined as the spent water of a community. Although it is more than 99.9% pure water, it contains wastes of almost every form and description. Raw sewage, when fresh, is gray and looks something like dirty dishwater containing bits of floating paper, garbage, rags, sticks, and numerous other items. If allowed to go stale, it turns black and becomes very malodorous. About 25% of the waste matter of normal domestic sewage is in suspension; the remainder is in solution.

Sewage contains many complex organic and mineral compounds. The organic portion of sewage is biochemically degradable and, as such, is responsible for the offensive characteristics usually associated with sewage. Sewage contains large numbers of microorganisms, most of which are bacteria. Fungi, viruses, and protozoa are also found in sewage, but to a lesser extent. Although most of the microorganisms are harmless and can be used to advantage in treating the sewage, the viruses and some of the bacteria are pathogenic and can cause disease.

Sewage flow generally averages between 50 and 200 gallons per capita per day (gpcd) (190 and 760 lpcd). In the absence of better information, an average figure of 100 gpcd (380 lpcd) is generally used for design purposes. The rate of flow usually varies from minimum in the early morning to maximum in the later afternoon. Minimum flow ranges from 50 to 80% and maximum dry-weather flow from 140 to 180% of average flow. The extent of variation decreases as the size of the system increases. Wet-weather flows can be 600 gpcd (2270 lpcd) or more because of the extraneous water entering sewers from roof drains, areaway drains, footing drains, and so on.

SEWAGE SYSTEMS

In most instances, sewage systems are divided into two parts: collection systems and treatment systems.

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Collection Collection systems consist of a network of sewers that collect and convey sewage from individual residences, commercial establishments, and industrial plants to one or more points of disposal. Pumping stations are often needed at various points in the system to pump from one drainage area to another or to the treatment plant. The judicious location of pumping stations enhances the economy of the overall design by eliminating the need for extremely deep sewers.

Small-system pumping stations are frequently built underground and may be factory-built. For larger stations, superstructures should be in keeping with surrounding development. It has been said that people smell with their eyes and their ideas as well as their noses, and for this reason aboveground structures should be attractive, with landscaped grounds, to overcome the popular prejudices against sewage works. Stations can be and have been designed and constructed in residential areas where the neighbors apparently are not aware that the stations are not homes.

Treatment Treatment facilities can be many and varied, with the extent and nature of the treatment determined to a large degree by the proposed use of the receiving stream and its ability to assimilate pollutants. Most conventional treatment plants being built today can be classified as either primary, biological, or advanced waste treatment. Other alternatives, such as physical-chemical or chemical-biological treatment, are also used on occasion but on a lesser scale. The treatment needs of smaller communities are sometimes satisfied by package treatment plants or by waste stabilization lagoons.

Primary treatment involves removal of a substantial amount of the suspended solids but little or no colloidal or dissolved matter. Primary treatment facilities normally include screening, grit removal, and primary sedimentation. The sewage is often chlorinated during primary treatment in order to sterilize the wastes.

Biological treatment uses bacteria and other microorganisms to break down and stabilize the organic matter. Trickling filters and the many variations of the activated sludge process are the most popular biological treatment concepts presently in use. Biological treatment is generally followed by final sedimentation of the solids produced by the microorganisms.

Advanced waste treatment is a very complex subject, and it can range from a limited objective, such as phosphate removal, to whatever additional treatment is necessary for water reuse purposes. Advanced waste treatment usually follows conventional primary and biological treatment and can include phosphate removal, nitrate removal, multimedia filtration, carbon absorption, and ion exchange. Where zero discharge is required, it may be necessary to follow advanced waste treatment with spray irrigation of the plant effluent or other methods of disposal.

Combined primary and biological treatment using the activated sludge process is perhaps the most commonly used treatment concept currently in use. A schematic drawing of a typical activated sludge treatment plant is shown in Figure 1. In the example, liquid treatment is accomplished by coarse screening, grit removal, fine screening (or communication), and primary settling, followed by aeration, final settling, and chlorination. Sludge processing includes thickening, dewatering, incineration, and liquid disposal of ash. There are many variations to this layout, but the one shown includes most of the pumping applications normally encountered in treatment plant design. Pumping requirements will of course vary from plant to plant, depending on the process used, the site size and topography, and the relative location of the various structures and equipment.

PUMP APPLICATIONS

Most of the pumping applications associated with the collection and treatment of sewage can be classified according to the general nature of the liquid to be handled. The primary classifications are (1) raw sewage, (2) settled sewage, (3) service water, and (4) sludge. There are also, however, a number of specialized applications involving the handling of abrasive materials, such as grit and ash. The types of pumps recommended for sewage applications are indicated in Table 1. Although included in the table, chemical pumps are not discussed in this section. They are covered in Section 12.7.

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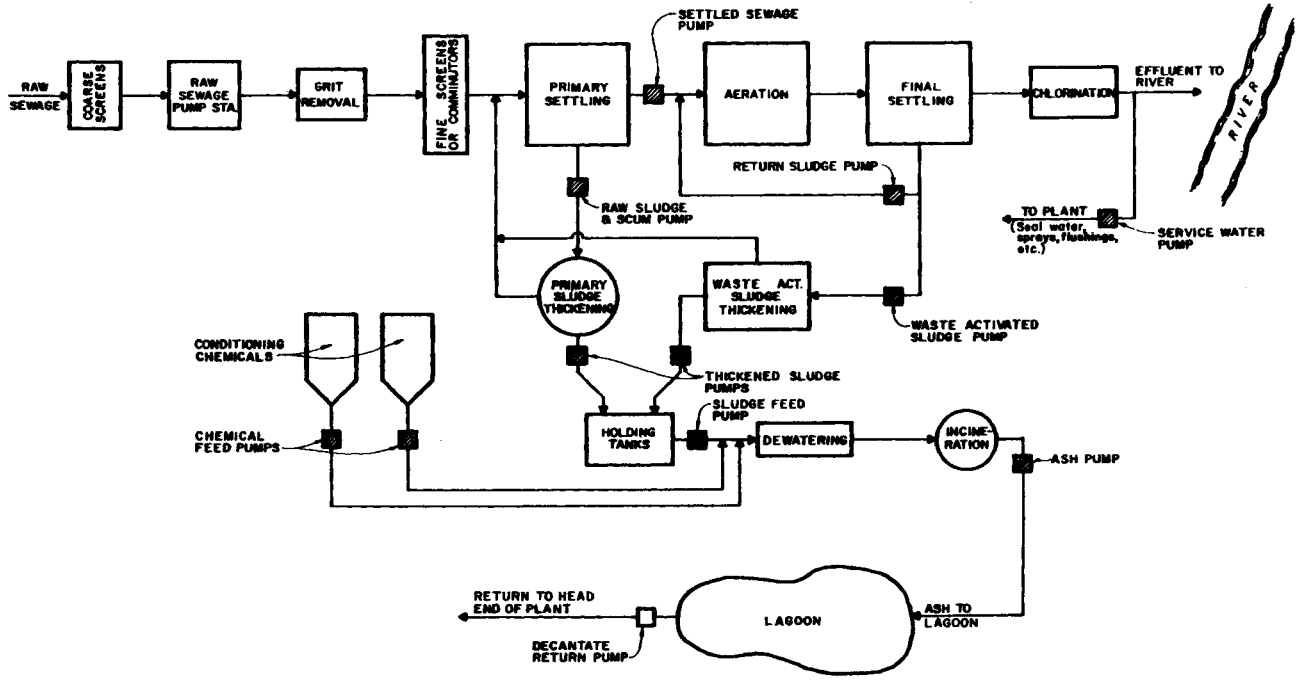


FIGURE 1 Typical activated sludge plant, with dewatering, incineration, and liquid disposal of ash

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TABLE 1 Types of pumps generally used in sewage applications

Application	Conventional sewage	Diffuser	Torque flow	Clear-water volute	Ash	Screw	Pneumatic ejector	Air lift	Positive displacement	Chemical
Raw Sewage	X	...	...	...	...	X	X	...	...	...
Grit	...	...	X	...	...	...	...	X	...	...
Primary sludge										
Less than 2% solids	X	...	X	...	...	...	...	...	...	...
More than 2% solids	...	...	...	...	...	...	...	...	X	...
Normal primary scum	...	...	...	...	...	...	...	...	X	...
Diluted scum	X	...	...	...	...	...	...	...	...	...
Biological sludge	X	X	...	...	...	X	...	X	...	...
Thickened biological sludge	...	...	...	...	...	...	...	...	X	...
Digested sludge, recirculation	X	...	X	...	...	...	...	...	...	...
Settled sewage	X	X	...	...	...	X	...	...	...	...
Plant effluent	X	X	...	...	...	X	...	...	...	...
Service or nonpotable water	X	X	...	X	...	...	...	...	...	...
Ash sluice	...	...	...	...	X	...	...	...	...	...
Decantate or supernatant liquor	X	X	...	...	...	...	...	...	...	...
Chemical solution	...	...	...	...	...	...	...	...	...	X

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Raw Sewage Raw sewage pumps are used to lift liquid wastes from one level of the collection system to another or to the treatment plant for processing. Regardless of where the pumps are located, the basic design considerations remain the same.

Even though the sewage is normally screened at larger installations before entering the suction wet well, it still contains a large quantity of problem material, such as grit, rags, stringy trash, and miscellaneous solids small enough to pass through the coarse screens. Screens are often omitted from smaller installations because large objects are not as much of a problem because of the smaller size of the incoming sewers.

Raw sewage pumping installations are usually sized so their firm capacity either is equal to a future maximum flow rate of the incoming sewers or can be expanded to accommodate this level. *Firm capacity* is defined as total station capacity with one or more of the largest units out of service.

Pneumatic ejectors are sometimes used where the required capacity is less than that provided by the smallest conventional sewage pump. This type of unit, however, should not be used where more than 50 connections are expected.

Conventional sewage pumps are, by far, the most common pumps used for the handling of raw sewage. A conventional sewage pump is more specifically described as an end-suction, volute-type centrifugal with an overhung impeller of either the nonclog (Figure 2) or the radial- or mixed-flow type (Figure 5), depending on capacity and head.

Nonclog pumps are all based on an original development by Wood at New Orleans. Actually, no pump has been developed that cannot clog, either in the pump or at its

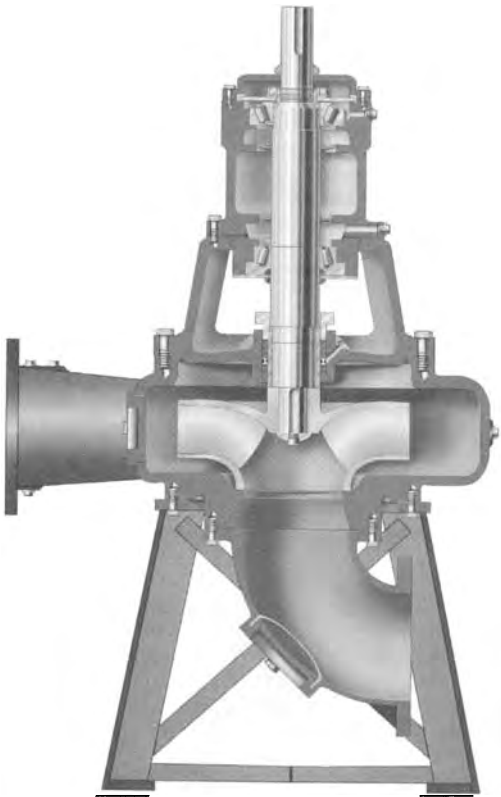


FIGURE 2 Nonclog dry pit sewage pump (Flowserve Corporation)

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appurtenances. Experience shows that rope, long stringy rags, sticks, cans, rubber and plastic goods, and grease are most conducive to clogging.

Nonclog impellers are used almost exclusively today for pumps smaller than 10 in (25 cm). These pumps differ from clear-water pumps in that they are designed to pass the largest solids possible for the pump size. The conventional nonclog impeller contains two blades, although some manufacturers are now offering a single-blade ("bladeless") impeller. The two-blade impeller has thick vanes with large fillets between the vanes and the shroud at the vane entrance. The bladeless impeller has no vane tips to catch trash. On the other hand, it is inherently out of balance because of its lack of symmetry.

The larger raw sewage pumps are equipped with either mixed-flow or radial-flow impellers, depending on head conditions. Both have two or more vanes, depending on pump size and the size of solids to be handled. The vane tips are sharper than for the nonclog impeller, resulting in a higher operating efficiency. The heavier vanes are not necessary because the vane openings can be larger than on the smaller pumps. Experience indicates that stringy trash will not clog an impeller with vane openings larger than 4 in (102 mm) in diameter.

Conventional solids handling volute-type sewage pumps may be of the dry pit type (Figure 2), wet pit type (Figure 3), or submersible type (Figure 4). Dry pit pumps are by far the most popular and widely used type due to their accessibility for observing mechanical operation and ease of maintenance when necessary.

Wet pit solids handling pumps are gaining in popularity especially on return activated sludge services. They are similar in design to a vertical turbine or mixed-flow diffuser pump except the bowl contains a limited number (usually two) of well-rounded diffusers,



FIGURE 3 Solids handling wet pit pump (Flowserve Corporation)

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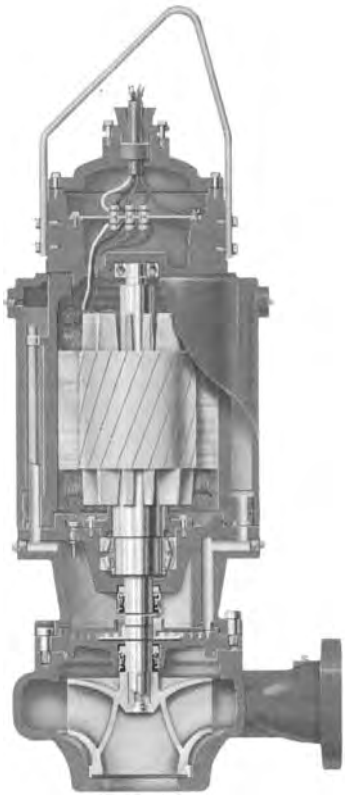


FIGURE 4 Nonclog submersible sewage pump (Flowserve Corporation)

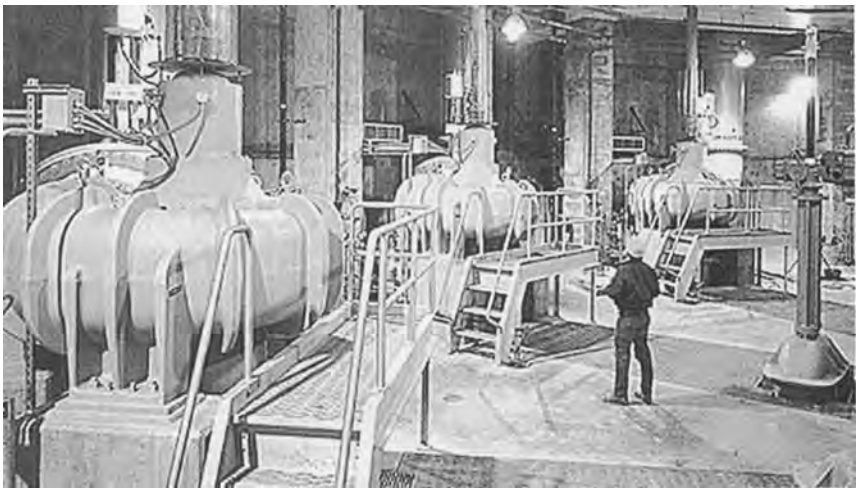


FIGURE 5 Vertical sewage pumping units at the South System Pump Station, Deer Island in Boston, MA, containing (8) 36 in (915 mm), 46,300 gpm (10,510 m^3/h) pumps driven by 1250 hp (932 kW) variable speed motors (Flowserve Corporation)

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the impeller is of the solids handling type, and there is no lower suction bell bearing. Therefore, they are not nearly as likely to clog on the debris normally found in raw sewage. Single-stage models available at the time of this publication are limited to operational total heads under 100 ft (30 m).

Submersible solids handling pumps are commonly used in small raw sewage lift stations. They are most widely used in sizes 12 in (305 mm) and smaller with total heads under 100 ft (30 m) and motor ratings under 150 HP (112 kW).

Although nonclog pumps 8 in (203 mm) and small are available with self-priming (Section 2.2.2), most conventional sewage pumps are located so the impeller is always below water level in the suction wet well. This eliminates the need for specialized priming systems.

Self-priming pumps have been used successfully to pump raw unscreened sewage, particularly in the southern part of the United States. The self-priming feature eliminates the dry-pit cost and gives the centrifugal pump the gas-handling advantage of positive displacement pumps. Operating costs are higher, though, because the design efficiencies generally run about 10 to 15% lower than for the conventional nonclog units.

Archimedean screw pumps (Figure 6) are occasionally used for raw sewage pumping applications. These units are advantageous in that they do not require a conventional wet well, and they are self-compensating in that they automatically pump the liquid received regardless of quantity as long as it does not exceed the design capacity of the pump. This is done without the need for variable speed drive equipment. Also, as shown by Figure 6, the total operating head of a screw pump installation is less than for those pumps that require conventional suction and discharge piping. Screw pumps, however, have a practical limitation as to pumping head. Generally speaking, they are not used for lifts in excess of 25 ft (7.6 m).

Settled Sewage Settled sewage pumps are used to lift partially or completely treated waste from one part of the plant to another or to the receiving stream. In Figure 1, these pumps are the settled sewage pump, service water pump, and decantate return pump.

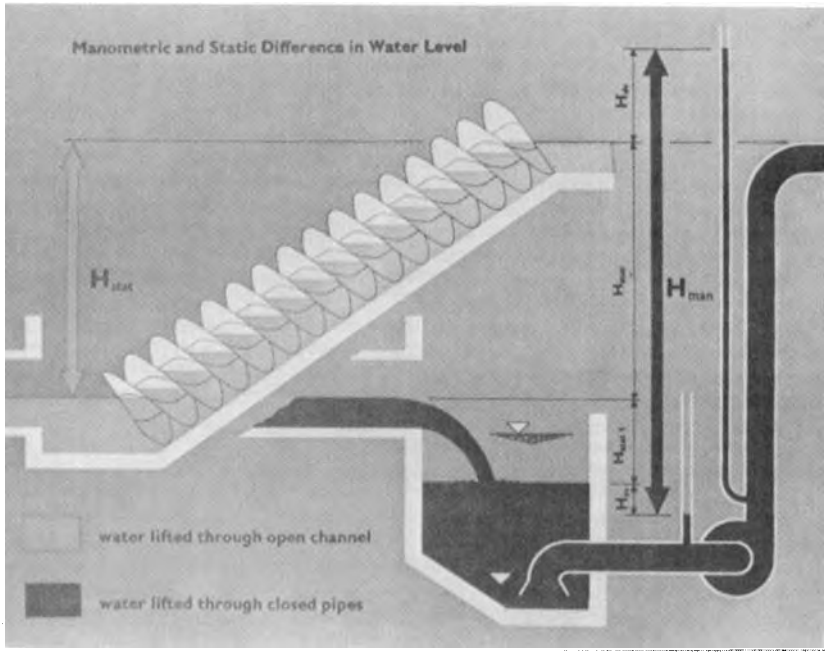


FIGURE 6 Archimedean screw pump (U.S. Filter/Zimpro)

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The liquid to be handled usually contains some solids, but grit and most of the rags and other stringy material have already been removed. Sufficient firm capacity should be provided to meet peak flow requirements. In no case should fewer than two units be provided.

Wet pit solids handling or diffuser pumps (Section. 2.2.1) are commonly used for the pumping of settled sewage. Depending on the total head conditions and degree of solids removal, a diffuser pump selection may be of either the propeller or mixed-flow design. However, solids-handling wet pit pumps are gaining in popularity due to their greater freedom from clogging. Although normally installed in wet-pit applications, these units are sometimes mounted on suction piping and installed in a dry pit. Either type of application is acceptable, although economics usually dictates a wet-pit installation. Head and capacity conditions will determine which type of unit is applicable.

Conventional sewage pumps may also be used to pump settled sewage. They may be of the dry pit, wet pit, or submersible type. It is not usually as economical to design a dry pit for this application, but it is acceptable as far as suitability of equipment is concerned.

Archimedean screw pumps can be used to pump settled sewage, provided the lift is not excessive. As previously noted, this type of pump has certain inherent advantages.

Service Water Plant effluent water is frequently used for flushing, gland seal, foam control sprays, chlorine injector operation, lawn sprinkling, fire protection, and various other services in a waste-water treatment plant. Except for the fact that some solids must be contended with, this application is much the same as that found in building-water supply and small distribution systems.

Screening of solids is normally required; this can be accomplished either before or after the pumps, depending upon various circumstances. Pipeline-type strainers are recommended as they are not only economical but require a minimum of space, can be automatically backflushed, and are much easier to operate than alternative equipment.

Any type of conventional volute or diffuser clear-water pump can be used on service water applications, provided the effluent water is screened prior to entering the pump. Pumps capable of handling some solids should be used in those instances where prescreening is not practical.

Sludge and Scum This classification is divided into two separate categories, based on the concentration of solids in the liquid to be handled. Specialized pumping equipment is required for more concentrated sludges, whereas pumping of dilute sludge and scum is somewhat comparable to the handling of settled sewage.

DILUTE SLUDGE OR SCUM For the purposes of this discussion, *dilute sludge* and *scum* is defined as having less than 2% solids. An exception is digested sludge recirculation, which generally exceeds the 2% limit. This is included along with the more dilute sludges because the same type of pumping equipment is used.

Normally, the handling of dilute sludge is limited to the transfer of biological sludge back to the treatment process or to some other point for further concentration or dewatering and disposal. When digesters are used as part of the treatment facilities, sludge is often recirculated through external heat exchangers in order to maintain temperatures conducive to anaerobic bacterial action. This recirculation also helps keep the contents of the digester mixed. Occasionally, primary sludge and scum are handled in diluted form.

The firm capacity of dilute sludge pumping facilities should be equal to anticipated peak loading. Biological sludge return pumps should have a capacity range from 25 to 100% of average design raw sewage flow to the plant. Digested sludge recirculation pumps should be sized to turn over the contents of the digester frequently enough to maintain the desired temperature. Diluted primary and waste biological sludge pumps should have sufficient capacity to handle peak sludge loading at conservative solids concentrations.

Conventional sewage pumps are suitable for handling dilute sludge and scum. Either the nonclog or mixed-flow impeller may be used, depending upon capacity requirements.

Diffuser pumps are particularly suitable for handling biological sludge that does not contain any appreciable amount of trash or stringy material. They are not recommended, however, for handling diluted scum or for recirculating digested sludge. Depending on capacity requirements, diffuser pumps may be of either the mixed-flow or propeller design.

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Wet-pit applications are most common, although dry-pit installations are occasionally used.

Torque flow (or vortex) pumps are often used to handle dilute sludges that contain some grit. These units are particularly suitable for this type of service because their design is such that close running tolerances are not required; this allows the use of specially hardened materials, such as high-nickel iron, which are not easily machined. The most common applications of torque flow pumps are for the pumping of nondegritted dilute primary sludge to gravity thickening and the recirculation of digested sludge.

Screw pumps can be used in certain instances for handling biological sludge. Use of screw pumps is generally limited to low to medium lifts and to those instances where the point of discharge is close to the sludge source.

Air-lift pumps are suitable for transferring biological sludge where the lift is small and the point of discharge nearby. A typical air-lift pump installation is shown in Figure 7. Total head should not exceed 4 to 5 ft (1.2 to 1.5 m). The ability of an air-lift pump to vary capacity is somewhat limited, ranging from about 60 to 100% of the rated amount. These pumps are inexpensive in first cost but have an operating efficiency of only about 30%. They are very easy to install, and maintenance is minimal because there are no moving parts. Air-lift pumps are commonly used to transfer sludge at package treatment plants.

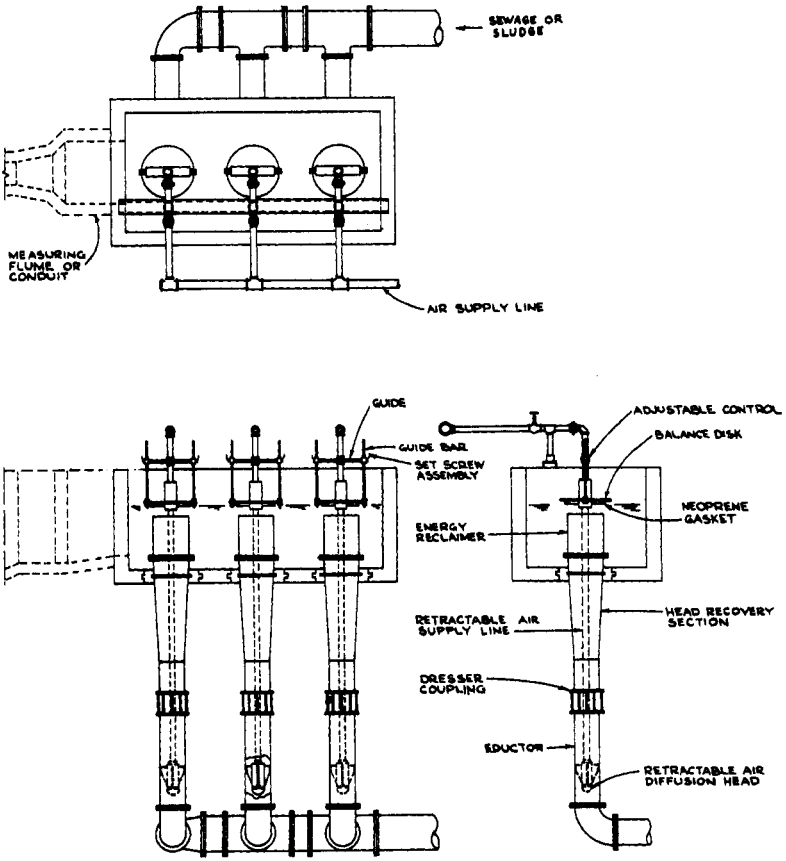


FIGURE 7 Typical air-lift pump installation (Walker Process Equipment Division, McNish Corp.)

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CONCENTRATED SLUDGE OR SCUM Concentrated sludge or scum is defined as having more than 2% solids. The single exception is in the case of the recirculation of digested sludge. As previously discussed, this has been included in the dilute sludge classification.

Each pumping installation should have enough firm capacity to handle peak design sludge quantities while operating part-time. The proportion of operating time at peak loading should vary from about 25% for primary sludge pumps to close to 80% for pumps feeding dewatering equipment.

Only positive displacement pumps are recommended for handling concentrated sludge and scum, mainly because they can pump viscous liquids containing entrained gas without losing prime. Also, these materials are thixotropic, and conventional formulas for frictional losses are not always valid. An arbitrary allowance of at least 25 lb/in^2 (170 kPa) should be added to the pumping head calculated by conventional methods to allow for changes in viscosity and partial clogging of pipelines. Positive displacement pumps are able to maintain a relatively constant capacity regardless of variations in discharge head.

For most applications, positive displacement pumps may be of either the plunger or the progressing cavity design (Figure 8). The performance of both depends upon close running clearances; consequently they have a high incidence of maintenance, especially where gritty substances are encountered. Even so, they represent the best pumping equipment currently available, and both designs have been used with success. Lobe-type gear pumps have been used for specialized applications. These are to be avoided, however, where there is any possibility that the material to be pumped will contain even a small amount of grit.

Plunger pumps should be of the heaviest design available and should be rated for capacity at about one-half of full stroke. The shorter the stroke, the more stable the operation and the less maintenance required. Heads as high as 80 to 100 lb/in^2 (550 to 690 kPa) are available and should be specified in order to give as much flexibility as possible.

Specially designed progressing cavity pumps are available for handling sewage sludges. Wear increases along with pump speed, and so excessive speed should be avoided. Ideally, the maximum speed of a progressing cavity pump should not exceed 350 rpm. These units are readily available with head capabilities up to 50 lb/in^2 (345 kPa) and should be so specified.

Certain of the newer sludge conditioning and dewatering processes, such as heat treatment and pressure filtration, require pumps having a head capability in excess of 500 lb/in^2 (3450 kPa). This is extremely difficult service, and special care should be taken in selecting the type of equipment to be used. So far, this area of application has received very little consideration from pump manufacturers.

Other Uses Grit may be handled with reasonable success with either a torque or an air-lift pump. Considerable flushing water is required with a torque flow pump, and to a lesser extent with an air-lift unit. A special ash pump is required where it is necessary to dispose of incinerator residue in a liquid form. These units are especially designed for ash sluicing service and are made of special hardened metals. No other pump should be considered for this service.

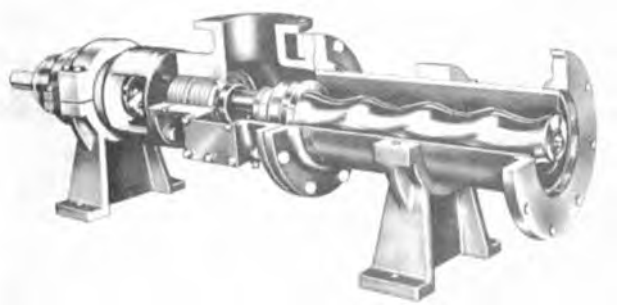


FIGURE 8 Progressing cavity sludge pump (Mono Pumps Ltd.)

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PUMP SELECTION

Various factors should be considered when selecting pumping equipment. These include the number of units to be installed, operating frequency, and station reliability requirements. After these factors have been fully evaluated, head-capacity curves should be prepared in order to match the pumps properly with system requirements. This is necessary because the capacity of most pumps varies with the total head at which the unit operates. When a pump is referred to as having a certain capacity, this capacity applies to only one point on the characteristic curve.

Number of Pumps The number of pumps to be provided at a particular installation depends largely on the required capacity and range of flow. In considering capacity, it is customary to provide a total pumping capability equal to the maximum expected inflow with at least one of the largest pumping units out of service. A minimum of two pumps should be installed in any installation except where pneumatic ejectors are used to serve fewer than 50 houses. Two pumps are customarily installed where the maximum inflow is less than 1.0 mgd (160 m³/h). At larger installations, the size and number of units should be such that the range of inflow can be met without starting and stopping pumps too frequently and without requiring excessive wet-well storage capacity. Variable-capacity pumps can be used to match pumping rate with inflow rate.

Where variable-capacity pumps are used, a minimum of two units should be installed. In those cases where more than one variable-capacity unit is required to handle peak flow, three units should be installed. In this manner, it is possible to maintain a reasonable rate of flow through each pump. Operation of a single variable-capacity pump in parallel with a constant-capacity pump requires the variable-speed unit to operate at almost no capacity whenever total inflow barely exceeds the rating of the constant-capacity unit. This is extremely difficult service and should be avoided. As a general rule, pumping rates of less than 20% of the rated capacity for which a pump is designed will result in excessive internal recirculation and unstable operation. Recirculation can occur in some pumps at more than 50% of rated capacity. See Section 2.1.3.

Operating Frequency Pump size should be coordinated with wet-well design in order to avoid frequent on-off cycling of pumps. Excessive starting will cause undue wear on the starting equipment. Also, standard motors should not be started more than six times an hour. Where more frequent starting is required, special motors should be provided. Inflow into the wet well without pumping should not exceed about 30 min if septicity is to be prevented.

Cycle time is defined as the total time between starts of an individual pump. It can be determined by comparing the volume between the on and off levels in the wet well with the pump capacity. Cycle time is computed as follows:

In USCS units
$$CT = \frac{V}{D - Q} + \frac{V}{Q}$$

In SI units
$$CT = 60\left(\frac{V}{D - Q} + \frac{V}{Q}\right)$$

where CT = cycle time, min
V = wet-well volume between on and off levels, gal (m³)
D = rated pump capacity, gpm (m³/h)
Q = wet-well inflow, gpm (m³/h)

With a given wet-well volume and pumps having a uniform pumping rate, minimum cycle time will occur when the rate of inflow is equal to one-half the discharge rate of the individual pump under consideration. The formula for cycle time simplifies to

in USCS units
$$CT = \frac{2V}{Q}$$

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in SI units $$CT = \frac{120V}{Q}$$

An effective wet-well volume of at least 2.5 times the discharge rate of the pump under consideration is required in order not to exceed the six starts per hour recommended above for pumps having a uniform pumping rate.

Reliability With its increased awareness of and concern for environmental matters, the public has little tolerance for the bypassing of sewage equipment because of power outages, equipment failure, insufficient pumping capacity, or any other reason. Reliability is of extreme importance, and the design of pumping facilities should be premised on providing continuous service. Where electric motors are used, two incoming power lines from separate sources with automatic switching from the preferred source to the standby source are the minimum required for reliability. Standby engine-driven pumps, engine-driven right-angle gear drives, or standby engine-driven generators should be provided where dual electric service cannot be obtained or where the degree of reliability provided by two feeds is not considered adequate. Raw sewage pumping installations are particularly critical. Plant pumping installations usually can be out of service for as long as 4 hours without adversely affecting the treatment process, provided the liquid will flow by gravity through the plant.

Speed The maximum speed at which a pump should operate is determined by the net positive suction head available at the pump, the quantity of liquid being pumped, and the total head. When specifying pumps, especially those that are to operate with a suction lift, the speed at which the pumps will operate should be checked against limiting suction requirements as set forth by the Hydraulic Institute.

In general, it is not good practice to operate sewage pumping units at speeds in excess of 1750 rpm. This speed is applicable only to smaller units. Larger pumps should operate at lower speeds.

Preparation of Head-Capacity Curves Pump selection generally involves preparation of a system head-capacity curve showing all conditions of head and capacity under which the pumps will be required to operate. Frictional losses can be expected to increase with time, materially affecting the capacity of the pumping units and their operation. For this reason, system curves should reflect the extreme maximum and minimum frictional losses to be expected during the lifetime of the pumping units as well as high and low wet-well levels.

Where two or more pumps discharge into a common header, it is usually advantageous to omit the head losses in individual suction and discharge lines from the system head-capacity curves. This is advisable because the pumping capacity of each unit will vary depending upon which units are in operation. In order to obtain a true picture of the output from a multiple-pump installation, it is better to deduct the individual suction and discharge losses from the pump characteristic curve. This provides a modified curve that represents pump performance at the point of connection to the discharge header. Multiple-pump performance can be determined by adding the capacity for points of equal head from the modified curve. Figure 9 shows a typical set of system curves, together with representative individual pump characteristic curves, modified pump curves, and combined modified curves for multiple-pump operation. Intersection of the modified individual and combined pump curves with the system curves shows total discharge capacity for each of the several possible pumping combinations. A typical set of system curves consists of two curves with a Williams-Hazen coefficient of $C = 100$ (one for maximum and one for minimum static head) and two curves with a Williams-Hazen coefficient of $C = 140$ (for maximum and minimum static head). These coefficients represent the extremes normally found in sewage applications.

Pumps should be selected so the total required capacity of the installation can be delivered with maximum water level in the wet well and maximum friction in the discharge line. Pump efficiency should be maximum at average operating conditions. In the case of Figure 9, assuming that the total capacity of the installation is to be obtained by operating pumps 1, 2, and 3 in parallel, the total head required at the discharge header would be approximately 51 ft (15.5 m). Projecting this point horizontally to the individual modified

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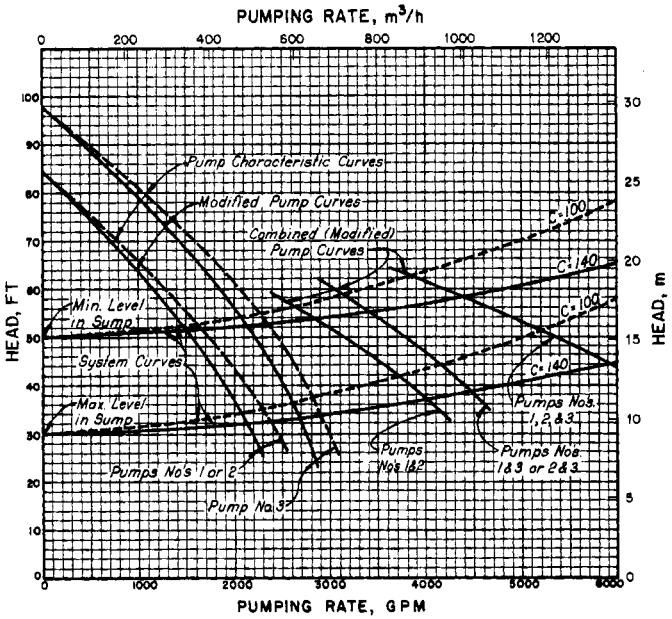


FIGURE 9 Typical head-capacity curves

pump curves and thence vertically to the pump characteristic curves, the required head for pumps 1 and 2 should be 54 ft (16.5 m) and for pump 3 approximately 57 ft (17.4 m). The difference between the head obtained from the pump characteristic curve and the modified curve is the head loss in the suction and discharge piping for the individual pumping units.

Figure 9 also shows the minimum head at which each pump has to operate, approximately 39 ft (11.9 m) for pumps 1 and 2, and about 42 ft (12.8 m) for pump 3. These minimum heads are important and should be made known to the pump manufacturer because they will usually determine the maximum brake power required to drive the pump and the maximum speed at which the pump may operate without cavitation.

PUMP DRIVERS

In the majority of cases, pumps are driven by electric motors. Sometimes, however, they are driven by gasoline, gas, or diesel units where firm power is not available or where pumping is required only at infrequent intervals. Variable-speed drivers are used extensively in sewage applications. These units generally consist of variable-speed motors or constant-speed motors with adjustable slip couplings of either the eddy-current or the fluid coupling type. Selection of the type of variable-speed driver to be used is usually based on space considerations, initial cost, operating cost over the expected life of the equipment, and customer preference. Emphasis is increasingly being placed on operating cost over the expected life due to government and environment requirements. See Section 9.2 for these and other types of speed-varying devices.

Variable-speed drivers are particularly appropriate for raw sewage installations that discharge to a treatment plant. Use of this equipment allows the treatment facilities to operate continuously instead of intermittently surging the plant at incremental pumping rates. Variable-speed drivers are used to pump settled sewage and biological sludge where

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intermittent surging would adversely affect the process. Also, sludge pumps used to feed dewatering equipment are often equipped with variable-speed drives because it is necessary to vary the rate of discharge with the dewatering characteristics of the sludge.

For dry-pit applications, the choice between horizontal- and vertical-drive motors depends considerably upon the station arrangement and available space. Horizontal motors are usually preferred, provided there is space and no potential flooding problem. Horizontal pumps are more easily maintained, and they are generally less expensive in first cost. Vertical drivers are generally used, however, for the pumping of raw sewage because of their smaller space requirements. Also, vertical units are advantageous in that the motor is located higher and is less susceptible to flooding.

In vertical dry-pit installations, intermediate shafting is normally preferred. This allows the drivers to be located above a potential flood level in the station. However, the use of intermediate shafting can become quite expensive in terms of initial installation costs and maintenance costs. Additionally, they may be prone to vibration problems if proper precautions are not taken in the design of the station. Vertical drive motor can also be mounted directly above a pump without the use of intermediate shafting. In doing so, it is connected to the pump by means of a suitable coupling. A separate support or bracing of the motor may also be required to provide for adequate installed stiffness.

PUMP CONTROLS

Some means of controlling pump operation is required at most pumping installations. This is usually done from either wet-well level or flow.

Level Control With level control, each pump is turned on and off at specific water levels in the suction wet well; in the case of variable-capacity pumps, the level control attempts to maintain a preset level, once started. Pumps turn on with a rising level and off as the level lowers. Level control is generally used in raw sewage pumping applications. In this manner, it is possible to match discharge with incoming flow.

Flow Control Flow control is used sometimes where there is no limitation on the availability of flow to the pump suction and where it is desirable to maintain a predetermined rate of discharge. Where flow control is used, a flowmeter is used as the primary instrument to measure flow and to serve as a basis for varying pump speed, which in turn controls capacity. The speed can be changed either manually or automatically through closed-loop instrumentation.

Additional Level Control Low-water pump cutoff and high-level alarm are provided on most pumping installations. The low-level cutoff is required to prevent the pumps from running dry, and the high-level alarm notifies the operator in the event the pump should fail to operate. When the pump stops because of low-water cutoff, there is usually some means of indicating this to the operator.

MISCELLANEOUS DESIGN CONSIDERATIONS

In addition to the matters discussed previously, there are certain other items that should be given consideration in the design of pumping installations.

Piping and Valves Suction and discharge piping should normally be sized so the maximum velocities do not exceed 5 and 8 ft/s (1.5 and 2.4 m/s), respectively. Higher velocities, however, may be justified by economic analysis for particular installations. Lines less than 4 in (102 mm) in diameter should not be used for raw sewage. Preferably, sludge lines should be at least 6 in (152 mm) in diameter; 4-in (102-mm) lines are sometimes used for dilute biological sludge.

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Valves should be installed as required on the suction and discharge sides of each pump to allow removal and maintenance of individual pumping units without disturbing the function of the remainder of the installation. It is customary to use either ball or plug valves on raw sewage and concentrated sludge applications. Either plug or butterfly valves can be used for settled sewage or for dilute sludge.

Piping should be designed with sufficient flexibility to avoid stress on the pump flanges. Flange-coupling adapters are sometimes used for this purpose on both the suction and discharge sides of the pump.

Surge Control Careful attention should be given to surge control wherever a pump discharges into a force main of appreciable length. Generally this is a problem only in the design of raw sewage pumping stations located within the collection system. Changes in fluid motion caused by starting or stopping of pumps or by power failure can create surge conditions.

Surges caused by normal starting and stopping of pumps driven by electric motors may be controlled by (1) selecting individual pump capacities such that the change in velocity in the system when a single pump starts or stops will not result in excessive surges, (2) using variable-speed drives to bring pumps gradually on- or off-line, or (3) using power-operated valves that are controlled so the pumps are started and stopped against a closed valve.

Surges caused by power failure can be controlled by devices designed to open on an increase in pressure, by devices that will exhaust sewage from the system upon sudden pressure drop in anticipation of surge, or by a surge tank.

Pump Seals Most sewage and sludge pumps can be obtained with either mechanical seals or packed stuffing boxes. Conventional mechanical seals have the disadvantage of requiring a pump to be disassembled so the seal can be repaired. Present mechanical seal technology offers a solution to this in the form of a split mechanical seal. Such a seal can be removed and repaired or replaced without the necessity of disassembling the pumping unit.

Often it is easier to replace the seal rather than repair it, and it is desirable to keep a spare on hand for this purpose. Packed stuffing boxes provided with water-sealed lubrication are still the most common choice for non-submersible sludge and sewage pumps. Grease seals are sometimes used for some of the smaller sewage pumps that do not run continuously.

Water serves multiple purposes as a sealing medium: it seals, lubricates, and flushes. Flushing is particularly important where abrasive material is involved in that it helps prevent this material from entering the seal. Grit and ash are very abrasive, and either will cut the shaft sleeves in a relatively short time. Where pumps are controlled automatically, a solenoid valve interlock with the pump starting circuit should be provided in the seal water connection to each pump. A manual shutoff valve and strainer should be provided on each side of each solenoid valve, and a bypass line should be provided around it.

Mechanical seals are normally lubricated with a clean external water source supplied at a pressure and flow rate as recommended by the seal manufacturer. Seals can be lubricated by the product being pumped provided it is filtered and can provide a reliable pressure and flow rate to the seal cavity. In such cases, a connection is normally provided between the pump discharge and the seal with a 0.10 to 0.20 in (2.5 to 5 mm) in-line filter to prevent foreign material from entering the seal cavity.

Occasionally, a pump station is so remote that sealing water is not readily or economically available. Mechanical seals can be provided constructed of special materials to withstand such adverse conditions. Also, several types of "formed-in-place" packing are available that do not require a smooth surface on which to seal as the packing develops its own sealing surface. However, these packing materials are not normally recommended for temperatures over 130°F (54°C) or pressures over 60 lb/in^2 (4 bar).

Pump Bearings Pump bearings must be adequate for the service and should be designed on the basis of not less than a minimum life of 5 years in accordance with the American Bearing Manufacturers Association life and thrust values. The larger sewage pumps are usually equipped with both case and impeller rings of bronze or chrome steel.

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Cleanout Ports Pumps should be provided, where possible, with cleanout ports on both the suction and discharge sides of the impeller. These are desirable for inspection and maintenance purposes.

Wet-Well Design Raw sewage wet wells should not be so large that sewage is retained long enough to go septic. It is usually desirable to limit storage to a maximum of 30 min. Shorter retention time is desirable. With the variable-speed controls now available, many stations can be designed so that the pumping rate matches the inflow rate and the inherent difficulties of frequent pump cycling or long retention times in wet wells can be avoided.

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SECTION 12.3
DRAINAGE AND IRRIGATION

JOHN S. ROBERTSON
JIM BIRD

"Water is a good servant, but it also is a cruel master." (John Bullein, 1562)

INTRODUCTION

The truth of John Bullein's insight is nowhere more evident than in the topics discussed here. Water can be the greatest of blessings or the tool of destruction, based simply on its quantity and location. Water has the power to turn deserts into gardens but it also can lay waste to years of building and development in a matter of hours. Harnessing the power of water is key to creating a durable infrastructure for humans within their environment, and the key invention for achieving this is the pump. It is virtually certain that the first efforts at moving large quantities of water were directed at solving the problems of too much or too little water at the desired location. The history of the earliest civilizations document devices that are the predecessors of today's pumping equipment. The shadoof, noria, and inclined screw were all developed to move water from rivers into fields for farming and into rudimentary public water supply systems. Floods presented another challenge and similar devices were used to drain low lying areas. The same challenges of moving water exist today, drainage systems exist to provide protection from the rages of water, and irrigation systems open up new areas to agriculture to spread the benefits of water over larger areas.

DRAINAGE PUMPS

Drainage pumps are used to control the level of water trapped in a protected area. Entrapment occurs when high lake levels, stream stages, and tides preclude the normal discharge of streams, storm runoff, and seepage from the protected area. These high-water conditions are created by floods and hurricanes or impoundment.

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Floods and hurricanes occur infrequently and are relatively short-lived. However, as normal drainage is not possible at such times, all interior drainage that cannot be ponded must be pumped over the protective works if the protective works is a levee or through it if it is a concrete flood wall. Pumps for this purpose are strategically located at the edge of pending areas and streams, in sewer systems, or in the protective works. These pumps are operated continuously or are cycled on and off as necessary to maintain the water level in the protected area below the elevation at which damage would be experienced. As their use is required only during emergency situations and usually under adverse weather conditions, reliability of operation is essential.

The history of one such installation chronicles the history of drainage systems in the United States. Many early settlements were established along waterways in an effort to provide both a ready source of water and access to transportation. This was true of a small French settlement established in 1718 on the Mississippi River. The initial location was on high ground, 14 ft above river level. Growth of the settlement moved the population away from the river and into a natural basin as much as 6 ft below normal water levels. Heavy rains, high tides, and floods produced rivers in the streets. Trapped water resulted in threats to the health of the residents. In the 1890s the Sewerage and Water Board was established to implement a drainage plan to meet the current and future need for a safe and secure New Orleans. The first installations were driven by windmills and ineffective. In 1906, A. B. Wood, an engineer, employed by the Sewerage and Water Board developed a pump with a 6-ft-diameter (1.8 m) rotor capable of moving larger quantities of water than possible before. He developed larger pumps, 12- and 14-ft (3.7- and 4.3-m) diameters, using the same principle. A photograph of one installation is shown in Figure 1. Wood's pumps were tested many times over the years by floods, rain, and tropical storms and performed admirably in resolving the city's drainage problems. In 1974, the A.B. Wood Screw Pump at the No. 1 Pump Station in New Orleans was designated as a National Historic Mechanical

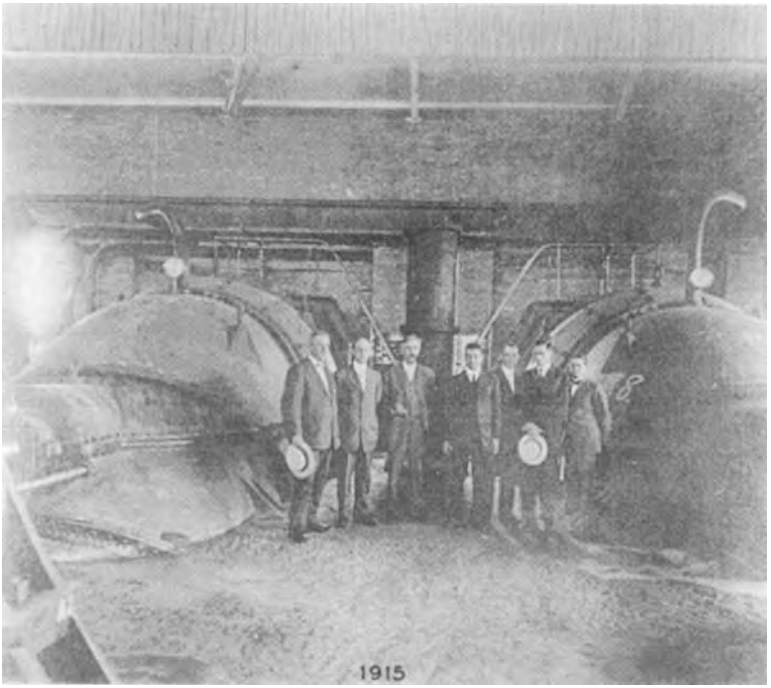


FIGURE 1 Wood's drainage pumps for New Orleans (ASME, 1974)[1]

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FIGURE 2 A recent New Orleans drainage pump (Flowserve Corporation)

Engineering Landmark.[1] Other pump stations were built over the years with installations and designs very similar to the Wood's pumps. One of these is shown in Figure 2.

In 2005, the New Orleans drainage system was put to its greatest test by Hurricane Katrina. The pumping stations feeding the drainage systems have been designed to lift water out of the basin and into the drainage canals moving large quantities of water at low pressures. The drainage canals, secured by levees, stretch into the inner city; and the failure of the canal system placed the pump stations under a load of fast, massive flooding that they were not designed to resist. New proposals for innovative designs capable of pumping high flows against heads necessary to utilize closed piping systems and avoiding the open canal system are being considered to replace the Wood's pumps and their offspring.

IRRIGATION PUMPS

From the annual floods on the Nile River that the Egyptians attributed to their gods the benefits of water on agriculture have been well known. The problem faced was how to maximize the benefits. Annual floods deposited rich silt for top soil but also took so long to dry that the growing season did not allow for multiple plantings or crops taking longer times form planting to harvest. Irrigation systems with pumps as the prime deliverer of water play an important part in making vast areas of arid and semiarid land agriculturally productive. As another benefit, proper irrigation management can make fertile areas capable of producing multiple crops throughout the year. Two types of irrigation systems exist differentiated by the source of the water that is to be brought to the crops.

In areas where surface water is plentiful and available from rivers, streams, lakes, and ponds, it can be pumped directly into the distribution system or into a conveyance system. Many systems use large pump stations to lift water from these sources into distribution canals that move water by gravity throughout the planting areas. At specific locations along the canals booster pumps remove water from the canals to delivery systems such as sprinkler systems. Most of the pumps used in these installations are mounted permanently and are arranged to operate as necessary without constant attendance during the growing season.

Ground water, water from subsurface sources, is lifted up to the surface by well pumps. The pumps normally provide energy to lift the water out of the ground and also to deliver it to the irrigation system. In such installations, a well is drilled in the vicinity of the area

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FIGURE 3 Center pivot irrigation pumping system

to be irrigated and is fitted with the proper size pump. Center pivot systems as shown in Figure 3 each have a pump at the center. The traveling sprinkler system is driven though the circular path by hydraulic motors. In most instances, more than one well is needed to provide the required capacity. Well pumps require care in sizing to be certain that as water is removed from the local aquifer the draw down of the water table does not fall below the pump intake. For this reason wells and well pump settings are becoming much deeper as the rate of withdrawal increases. Pumps with settings of 1000 ft (300 m) below the surface are not unusual.

PUMP TYPES

Centrifugal pumps are used almost exclusively in drainage and irrigation installations. Because of the large selection of propeller, volute, turbine, and portable pumps manufactured today, there is little difficulty in finding a pump that will meet the conditions encountered in these fields. Pumps in the sizes needed to meet the requirements of the majority of installations are available as standard items. Specialized pumps and extremely large pumps are designed and built to meet the needs of individual projects.

Propeller Pumps These units are used for low-head pumping. As most of the pumping in drainage and irrigation is low-head, the propeller pump is the most widely used type. In general, vertical, single-stage axial- and mixed-flow pumps as shown in Figure 4 are used; however, there are instances where two-stage axial-flow pumps should be considered for economic reasons. In systems that are designed for high flow rates the casing of the pump may be formed in concrete with a pull out pumping element. A typical design for one such station and a detail of the pumping element are shown in Figure 5.

Horizontal axial-flow pumps are used for pumping large volumes against low heads and usually employ siphonic action when not of the submersible type. When higher heads are involved, these pumps can be arranged to operate with siphonic action until the back pressure places the hydraulic gradient above the pump.

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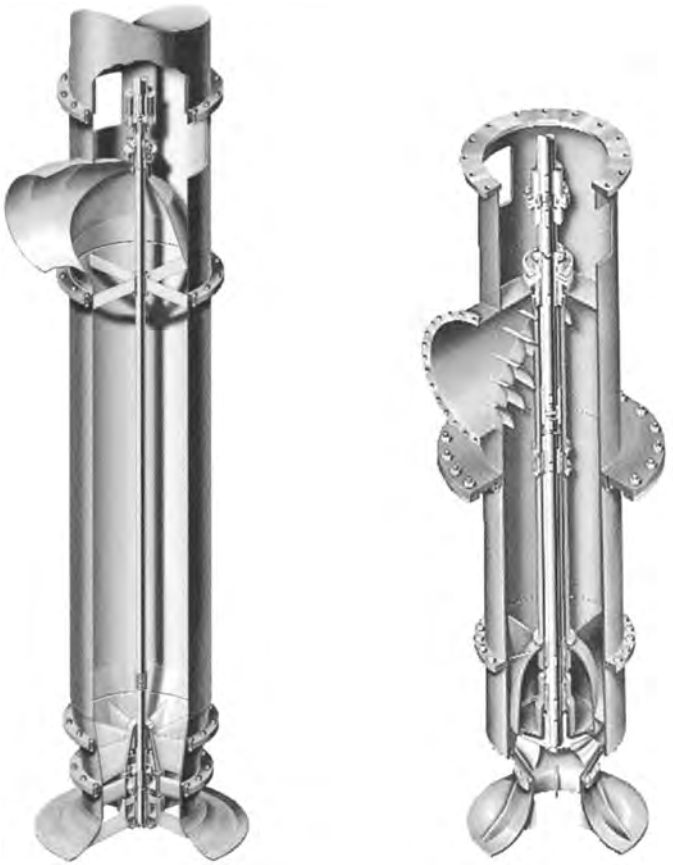


FIGURE 4 Vertical, single-stage pumps of the propeller type (left) and the mixed-flow type (right) (Flowserve Corporation)

Volute Pumps Volute pumps are used when pumping from surface sources and in general when the total head exceeds approximately 45 ft (14 m). Such pumps are available in many types, such as vertical and horizontal shaft, end suction, bottom suction, and double suction with semi-open or closed impellers, etc., as illustrated in Figure 6. These pumps are mounted in dry pits when located below grade and on slabs or floors when located above grade. The particular type used depends on the capacity and kind of service to be performed.

Deep-Well Turbine Pumps Deep-well turbine pumps (vertical-shaft, single-suction pumps having one or more stages, as shown in Figure 7) are used in irrigation primarily to pump water from a subsurface source into a distribution system. The head against which the pump will operate determines the number of stages that must be provided. For large flow rates, more than one pump will be needed.

Submersible Turbine Pumps In these deep-well units, the motor is close-coupled to the pump and submerged in the well as illustrated in Figure 8. This type of pump is used for high-head applications where long intermediate shafts are undesirable.

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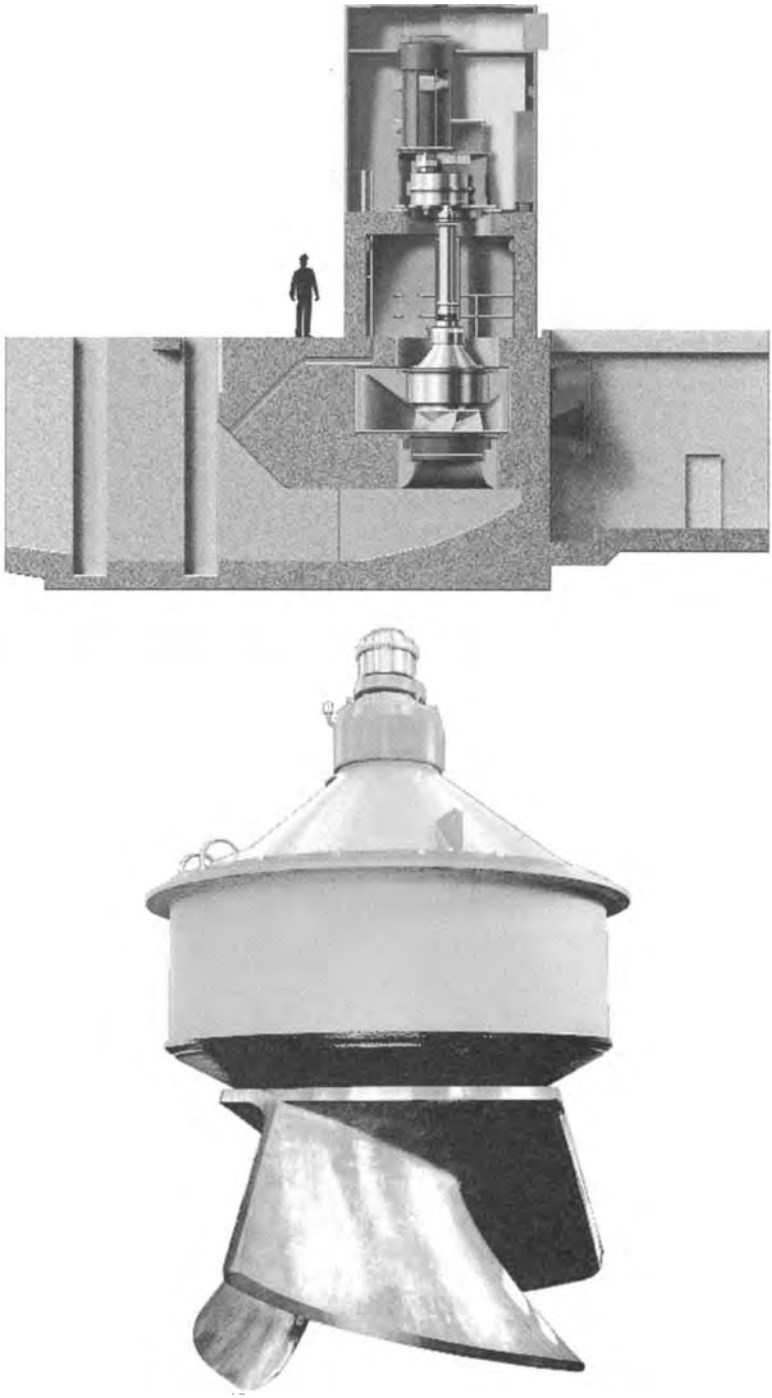


FIGURE 5 High-flow-rate pull-out pump and concrete casing (Flowserve Corporation)

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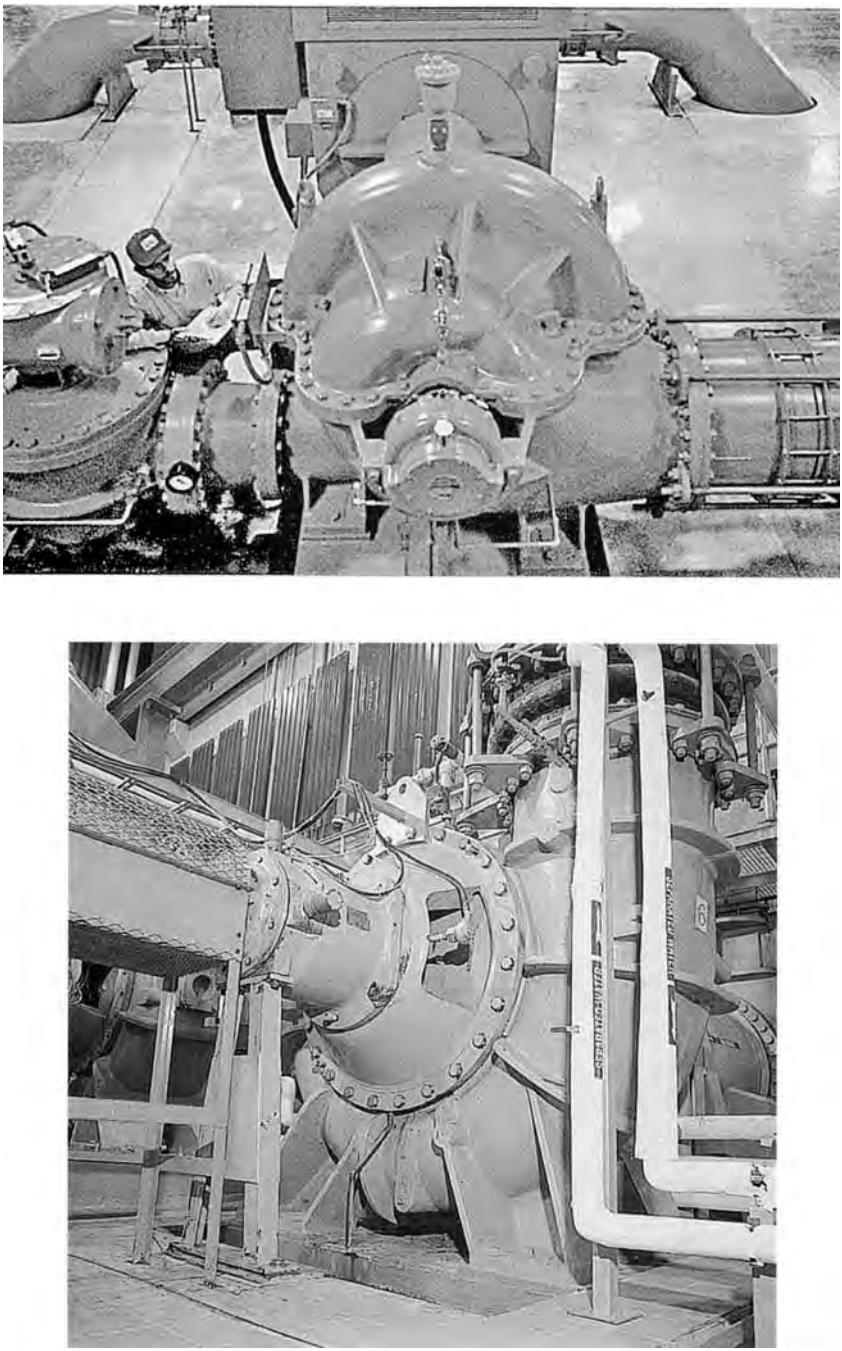


FIGURE 6 Horizontal-shaft volute pumps: double suction (above) and end suction (below) (Flowserve Corporation)

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FIGURE 7 Multistage, deep-well, vertical turbine pump (Flowserve Corporation)



FIGURE 8 Submersible vertical turbine pump (Flowserve Corporation)

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Portable Pumps In drainage work, portable pumps are used as emergency equipment to control ponding elevations where mobile equipment, such as tractors and trucks, having power takeoffs are available to drive them. In irrigation work, they are usually used to irrigate from a surface source. Such pumps are small-capacity, low-head, economical pumps that must be submerged in order to be used.

PUMP SELECTION

A pump selection study should always be made, and its importance cannot be emphasized enough. Such studies permit selection of the type of pump and discharge system best suited to the project and provide the information needed to proceed with the design of the installation.

Before initiating such a study, all previous studies made to determine the total pumping requirement or station capacity, pertinent water-surface elevations, terrain, utility locations, proposed station or well locations, points of discharge, and the proposed method of operation should be reviewed.

Number of Pumps First costs are generally of more concern than operating costs in drainage and irrigation work because the operating period for the majority of installations is relatively short and occurs only once a year. Costs can be minimized by using as few pumps as possible. However, one-pump installations are seldom used except in the case of wells. For reliability, a minimum of two pumps should be installed in drainage pumping stations, where the loss of even one pump during an emergency situation could result in considerable damage. Three or more pumps are preferred. Standby units are provided only in those installations where continuous operation precludes taking a pump out of service for maintenance.

The number of pumps ultimately used should be consistent with the demands of the project. For instance, when the installation is located in an agricultural area or is a part of an urban sewer system, a standby pump should be provided because these installations must always be capable of discharging project requirements during periods of blocked drainage. In this way, considerable damage may be avoided. When the installation is used to pump storm water from pondage or irrigation water from a lake, the loss of a pump is not critical and so a standby pump is not needed.

For installations requiring the use of large pumps, foundation conditions become important. To prevent the installation from being relocated to a less desirable site or the necessity of providing a more expensive pile foundation because the bearing pressures at the selected site exceed the allowable limit, the number of pumps should be increased, provided the loading can be reduced to an acceptable amount and the resulting installation continues to be the most economical.

Inadequate depth on the suction side of the pumps may necessitate the use of more pumps. Should the water not be deep enough to provide the submergence needed by the contemplated pump, more pumps of a smaller size may have to be used or the sump and approach channel may have to be excavated to the needed depth. The latter alternative could cause operational and maintenance problems and might be the more expensive solution.

Capacity The capacity of a pump (i.e., the flow rate which it is designed to handle) is a function of the total pumping requirement, the number of pumps, and, in the case of wells, the capacity of the well. Whenever possible, all pumps in a multiple-pump installation should be of the same capacity. This is advantageous from a cost standpoint as well as from a maintenance standpoint. In drainage installations, three pumps are generally provided in order to have the capability of pumping not less than two-thirds of the project requirement with one pump inoperative. Each pump would then have a capacity equal to one-third of the total capacity. When more than three are used, the capacity should be the total required capacity divided by the number of pumps being used. If two pumps are used, each pump should be sized to pump not less than two-thirds of the total capacity.

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In irrigation installations utilizing multiple pumps, the capacity of each pump should be the same. In single-pump installations, the pump should be sized to meet project requirements. When wells are involved, the capacity of the pump will be determined by the capacity of the well. Often several wells are needed to give the capacity to satisfy the established requirements.

When a standby pump is to be provided, its capacity should be equal to that of the largest pump being furnished.

Head Total head is the algebraic difference between the total discharge head and the total suction head. In drainage and irrigation work total suction head can usually be determined, but this is not always the case for the total discharge head. The losses in the discharge systems often must be determined by hydraulic model test rather than by calculation and therefore must be estimated for preliminary selection purposes. The head specified will have to be some head other than total head; generally pool-to-pool head is used.

Total discharge head is defined in the Hydraulic Institute Standards, and its value is to a great extent determined by the type of discharge system used. A number of the many possible discharge systems used are shown in Figure 9. The losses in systems *A, B, C, D,* and *E* can be calculated. Thus the performance for pumps discharging into these systems can be put on a total head basis.

System A is an "over the levee" siphonic discharge line, the use of which can seldom be justified. However, continuous operation over long periods could effect a savings that would be sufficient to justify the additional cost of the installation and the taking on of the operational hazards usually encountered with such lines. The total discharge head for this system is equal to the height of the discharge pool, stream, or lake above the impeller plus the exit loss and the discharge line losses from the pump discharge nozzle to the line terminus. The absolute pressure at the high point of the line should be not less than 9 ft (2.7 m). Lower values have been used successfully, but this should be the exception rather than the rule. For additional information on the determination of heads in a siphonic system, refer to "Siphon Head" in Section 11.1.

Check valves should be installed on the discharge end of all lines subjected to a cycling operation. The closing of these valves following pump shutdown prevents reverse flow into the protected area and permits development of the pressures needed to keep the lines full (primed) during those short periods when pumps are idle. These pressures will be less than atmospheric pressure, with the minimum pressure (absolute) being at the high point. Should there be significant leakage at the joints or in the valves, the pressures will rise and the water level at the high point will drop. If the water level drops below the invert of the line at the high point, the two legs of the line will be separated by an air space and priming will be necessary when the pump is started.

Air-release valves should be installed at the high point of all siphonic discharge lines to provide an escape for the air being compressed as the water rises in the line during priming, and vacuum-break valves should be used to prevent reverse flow into the protected area. For lines equipped with flap valves, the vacuum-break valve should be manually operated and used when the flap gate fails to seat properly or to provide a rapid means of draining the line when pumping is no longer required. Units for almost any size line can be obtained from manufacturers who specialize in such equipment or they can be assembled, using swing check and angle valves.

System B in Figure 9 is an "over the levee" nonsiphonic discharge line. To preclude siphonic action, this line should be vented at the high point with a vent having a diameter that is approximately one-fourth that of the line. The invert of this line at the high point should be placed at the same elevation as the top of the protective works so the pumped flow can discharge from the down leg of the line under gravity without backwater effects for all discharge pool elevations up to maximum. Thus the total discharge head will have a constant value because it will not be affected by changes in the level of the discharge pool. To ensure adequate prime mover capacity when using this system, it is the practice in all cases to use the top of the line at the high point in lieu of the hydraulic gradient when determining the total discharge head. Therefore, total discharge head is the height of the top of the line at the high point above the impeller plus the velocity head in the line at the high point based on a full pipe and the losses in the line from the pump discharge nozzle to the beginning of the down leg of the line.

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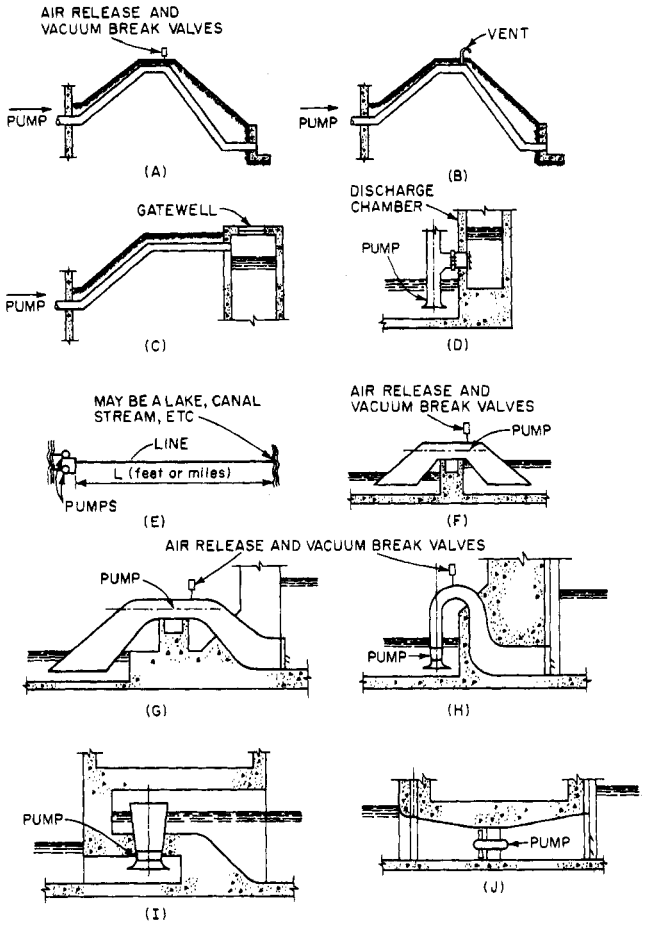


FIGURE 9A through J Discharge systems (adapted from Reference 2)

System C is used when there is a conduit carrying the normal gravity discharge under the levee adjacent to the station that must be valved off against reverse flow into the protected area during periods of high water. The closure gate is located in a gate well constructed on the stream or lake side of the levee to prevent subjecting the gravity conduit to high-water conditions. The pump discharge lines go over the levee and terminate in the gate well above the maximum water level. This shortens the lines and reduces the cost. The total discharge head for this system is equal to the height of the top of the line at the terminal end above the impeller plus the exit loss and the losses in the line between the pump discharge nozzle and the terminal point. The total discharge head for this system, as in system *B*, is independent of the discharge pool and therefore constant. Neither flap valves nor vents are required on these lines.

System D is used when the pumping station is constructed as an integral part of the levee or flood wall. The invert of the pump discharge line is placed at an elevation that is above the stream or lake level that will prevail approximately 70% of the time or as is dictated by the physical dimensions of the pump. Owing to the extreme turbulence in the discharge chamber, gates with multiple shutters, which are less likely to be damaged, should be used instead of flap gates on discharge lines that are larger than 36 in (914 mm) in diameter.

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When the water level in the discharge chamber is below the top of the discharge line, the total discharge head is determined in the same manner as for system *C*. For higher discharge water levels, the total discharge head is equal to the height of the water level in the discharge chamber above the impeller plus the exit loss and the losses between the pump discharge nozzle and the chamber side of the flap valve.

System E is perhaps the most common discharge system in use today. It is used to connect one pump or several manifolded pumps with a lake, canal, stream, ditch, reservoir, or sprinkler system. For short lines and low static heads, valve and fitting losses, frictional losses, and exit losses are very important, whereas in long lines or very high static head installations only frictional losses are given consideration. In manifolded installations using propeller pumps, a check valve and gate valve are installed immediately downstream of the pump. The gate valve should always be opened before the pump is started because the motors provided are not usually sized to operate against shutoff head. Positive shutoff valves are placed immediately downstream of volute or turbine pumps because these pumps are usually started and stopped against a closed valve. They also prevent reverse flow into the sump when one of the pumps is inoperative.

Pool-to-pool head is the difference in elevation between the sump and discharged-water surfaces and is used instead of total head in drainage work because the losses in the discharge system are not easily determined. Installations of this type are exemplified by systems *F*, *G*, *H*, *I*, and *J* in Figure 9. For such installations, it is best to specify the pumps on a pool-to-pool basis, to have the pump manufacturer design the pump and the discharge system, and to verify the predicted performance by model test. It should be noted that in such installations the discharge systems are usually constructed within the confines of the pumping station structure.

System F is operated as a siphon with the pump supplying energy equivalent to the pool-to-pool head plus the system losses. The invert of the pump discharge pipe at the highest point is located above the maximum river stage, and vacuum pumps are generally used to aid in priming the pump. This system is used for pool-to-pool heads of up to approximately 6 ft (1.8 m) and where the physical dimensions of a vertical pump would make it necessary to operate against higher heads. In estimating the losses, entrance losses, which are small [approximately 0.14 ft (4 cm)], should be neglected. The centerline of the suction piping should be assumed to make an angle of 45° with the horizontal, and the diameter of the discharge piping as measured at the discharge flange of the discharge elbow should be such that the velocity at maximum discharge is approximately 12 ft/s (0.037 m/s) or less.

System G is used for pool-to-pool heads of up to approximately 15 ft (4.6 m). The water passages change in cross-section from round at the pump bowl to rectangular at each end. The width at the suction end is the same as or less than that of the suction bay, and the height is such that the entrance to the suction passage is always submerged when the pump is in operation. The discharge velocity should be kept to approximately 6 ft/s (1.8 m/s). Multiple-shutter gates (Figure 2) arranged to be raised when the pump is in operation are provided to prevent prime mover overload when the pump is started and to prevent reverse flow when the pump is stopped or inoperative. These pumps are usually large and slow and require substantial prime movers. Vacuum priming equipment is used so additional power will not be required for priming.

System H is essentially the same as system *G*, except that the former is used for pool-to-pool heads up to approximately 26 ft (7.9 m). Also, a splitter may be required in the discharge water passage for structural purposes as well as for keeping the multiple-shutter gates to a reasonable size. The pumps used with this system are vertical and in general smaller than those used with system *C*.

System I can be constructed with the lip of the pump column either above or below the design flood elevation or the maximum surge, if it is being provided for hurricane protection. If the lip of the pump column is above and the chances of reverse flow through the pumps are extremely remote, gating of either the pump column or the discharge water passage is unnecessary. If the lip of the pump column is below, a decision of whether to gate both the pump column opening and the discharge water passage or just the water passage must be made. In general, if the pumps will be in operation continuously during high-water conditions and if reverse flow through an inoperative pump will not have substantial detrimental effects, only the multiple-shutter gate at the end of the discharge water passage

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need be provided. Reverse flow could occur should the flap for some reason fail to close. Like system *C*, splitter walls may be required if pumps are large. Also, the clearance between the lip of the pump column and the ceiling of the discharge water passage is critical and should be determined by model test.

The shape of the pump column lip will also affect pump efficiency and should be determined by test.

System J could have many configurations but, regardless of arrangement, would have to be gated on the suction side with a positive shutoff gate, such as a pressure-seating slide gate, and on the discharge side with a positive shutoff gate and a multiple-shutter gate or a flap (check) gate if the installation is small and a pipe is used in place of formed water passages. When formed water passages are used, transition sections between the pump and the gates sections will be needed and should be designed by the pump manufacturer.

Total Suction Head The practice for propeller pump installations is to dimension the station sump or sump bays in accordance with Hydraulic Institute standards and to make the distance between the sump floor and the lip of the suction bowl conform to the standards or the recommendations of the pump manufacturer. The approach and entrance velocities resulting therefrom are small enough to be disregarded when calculating total suction head. Total suction head in vertical propeller pump installations is the height from the centerline or eye of the propeller to the water surface in the sump. For drainage installations using vertical propeller pumps, a total suction head of zero is not uncommon. In submerged horizontal propeller pump installations, total suction head is the height from the centerline of the propeller shaft to the water surface in the sump, and the minimum value should be not less than $1.2D$, where D represents the diameter of the propeller.

Volute pumps equipped with a formed suction or suction piping may operate with either a suction lift or a suction head, depending on the suction water level. In either case, all the losses between the entrance and the eye of the impeller should be included in any calculation. Approach velocity is not a consideration in these installations, but entrance losses are.

TOTAL SUCTION LIFT Approach and entrance velocities can be disregarded in suction lift installations by proper dimensioning of the station sump or sump bays and by proper setting of the suction bell. The sump or sump bay dimensions used, as in suction head installations, should conform to the Hydraulic Institute standards or to the recommendations of the pump manufacturer. The suction bell for both horizontal propeller pump and vertical volute pump installations should be set with the lip located approximately $0.5D$ above the sump floor. For horizontal propeller pump installations, the minimum submergence of the suction bell should be approximately $0.25D$, where D is the diameter of the suction pipe; for volute pump installations it should be $1.5D$, where D is the diameter of the suction bell. In determining the total suction lift for these installations, it is assumed that the suction piping is a part of the pump and that the approach and entrance velocities can be disregarded. Total suction lift therefore is the height from the water surface in the sump to the centerline of the propeller shaft or to the eye of the impeller.

Setting The setting of the pump or the locating of the centerline or eye of the propeller or impeller with respect to the water surface should be given careful consideration when selecting the pump to be used. Some installations offer few if any problems in this regard, whereas in others setting may have a considerable effect on the size and number of pumps selected.

TURBINE PUMPS Drawdown is a consideration in any well installation. In setting the pump to prevent cavitation, sufficient NPSH at the eye of the first-stage impeller and/or sufficient depth over the suction bell lip or tailpipe to prevent vortexing should be maintained when maximum draw-down is being experienced.

VOLUTE PUMPS Volute pumps may operate with either a suction head or a suction lift. If with a suction lift and of the horizontal type, the pump should be set above the maximum anticipated elevation of the suction water source in order to avoid inundation. As priming

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will be necessary when starting, a suction lift would be considered a satisfactory arrangement when long periods of continuous operation are anticipated.

Suction head installation is the preferred practice (because priming is unnecessary) and should be used whenever conditions permit. In drainage work, vertical volute pumps are used for pumping small amounts of rainfall runoff and seepage flows. These pumps are usually located in a dry sump adjacent to the storm water pump sump, with motors and valve operators located on the operating floor above. The submergence of these pumps should be such that when discharging at maximum capacity, the total suction head is zero or above. In many instances, water surface fluctuations of just a few feet occur. When small volumes are involved, a cycling operation occurs. For this type of operation, the pumps are usually started at the maximum water level and stopped at the minimum level.

PROPELLER PUMPS Sufficient water depth does not always exist or cannot always be provided to give the submergence needed to permit the smallest and perhaps the most efficient pump to be used. Excavation is one answer. However, depending on the soil type, the location of the installation, the kind of construction used, the silt-carrying characteristics of the stream, and the frequency of operation, excavation may be an operational and maintenance headache and is impractical when sumps are to be made self-draining. Another alternative—and the one most frequently used in drainage work—is to set the centerline or eye of the propeller at or slightly below the minimum sump elevation and select a pump that will operate at this setting with little or no cavitation damage. This means that a larger pump operating at a lower speed should be used.

PUMPING STATION

Sump The sump is perhaps the most important element in the structure of the pumping station. Unless it is properly located, designed, and sized, the flow conditions within could have an adverse effect on the operation of the pump. There are many variations in sump arrangements that are acceptable; however, best results are obtained when the sumps or sump bays are oriented parallel to the line of flow. Flows approaching from an angle create dead spots and high local velocities, which result in the formation of vortices, nonuniform entrance velocities, and an increase in entrance losses. The flow to any pump should not be required to pass another pump before reaching the pump it is meant for. When sumps or sump bays are normal to the direction of flow, such as in sewer systems, the distance between the sump or sump bay entrance and the pump must be sufficient for the flow to straighten itself out before reaching the pump. For additional information relative to sump design and sizing, refer to Chap. 13. If the installation is large enough to warrant it, modeling of the sump to permit the best design to be determined is advocated.

Sumps in drainage installations pumping storm water should be either located above normal water levels, in which case they would be self-draining, or isolated from normal flows by gates. In small installations, motorized pressure-seating gates should be used; in large installations, roller gates raised and lowered with a crane or by some other suitable system should be used. These gates should be sized so the velocity through them will not exceed 5 ft/s (1.5 m/s) for any condition of flow. One gate should be located directly opposite each pump when all pumps are installed in a common sump or should be located at the entrance to each sump bay when pumps are separated.

Frequent cycling of pumps is encountered in installations that pump from sewer systems and that pump seepage. In such installations, the sump should be sized so the volume stored in it and in the ponding area or the sewer lines, as the case may be, within the limits of the operating range, will be sufficient to prevent the starting of the pumps more often than once every four minutes.

Superstructure A superstructure is provided on practically all drainage pumping stations but not on all irrigation pumping stations. The type of superstructure provided should be consistent with the surrounding area and should have a minimum number of windows and openings. In rural areas, corrugated sheet metal structures, which are

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inexpensive, have been used extensively. These provide adequate protection from the elements and from vandalism. To eliminate the need for an indoor crane, hatches in the roof over each pump can be provided to permit a truck crane to remove motors and pumps as a unit. For engine-driven pumps and larger pump installations, indoor cranes of the appropriate size and type should be provided for installation, removal, and maintenance.

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SECTION 12.4
FIRE PUMPS

MARIO DI MASI
RUDOLPH KAWOHL

Compelling reasons dictate the installation of fire protection systems driven by stationary fire pumps. Foremost among these reasons is protection—protection of lives, of equipment, possessions, and inventories, and of major assets, such as hospitals, hotels, office and residential buildings, and warehouses. There are two additional, less obvious but equally compelling, reasons: the reduction of costs and the protection of income-generating operations.

Costs are reduced in a simple manner. Over the projected life of a facility, the total of construction costs plus fire protection equipment costs plus fire insurance costs is lower than the total of construction costs plus fire insurance costs without fire protection equipment.

FIRE PROTECTION SYSTEMS

Protection may be provided by a combination of several complementary means, acting at various levels:

- First by a simple fire detection system activating an alarm
- On the second level, the detection and warning system is combined with a first degree of fire fighting; for example, with a *sprinkler installation*, the role of which is to extinguish the fire at the beginning or, at least to limit its extension so as to permit intervention with additional equipment
- The third level is a combination of detection, warning, sprinkler, and extinguishing systems (water, inert gases, foam)

The degree of protection is obviously a function of risk. Whereas a fire in a single-family-dwelling may be warned by phone and then efficiently contained by means of fire hydrants or fire fighting trucks, the risk of fire in a chemical plant requires a completely

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different approach. The rules for adequate protection vary from one country to another. There are, for instance, well known pamphlets published by the U.S. National Fire Protection Association (NFPA). In Europe, various national rules exist, based on the recommendations of the European Committee of Insurances, such as those of APSAD in France (Assemblée Plénière des Sociétés d'Assurances Dommages), V d S in Germany (Verband der Sachversicherer e;v), and LPC in the United Kingdom (Loss Prevention Council).

FIRE PUMP CRITERIA

The most common liquid available for fire protection and fire fighting continues to be water. The efficiency of a fire fighting installation depends to a great extent on a dependable water supply, which will supply the required flow at the required pressure and be continuously available during the time necessary to extinguish the particular type of fire in question.

The needs for fire fighting liquid vary considerably with the degree of danger involved. When these needs cannot be satisfied from local water sources, such as public water networks, high level reservoirs, or pressurized water tanks, pumping stations need to be designed and situated (located) to address the types of fire possible and the national codes and standards that exist in a particular country or geographical area.

A typical fire fighting station may be composed of the following:

- A jockey pump (Figure 1) that maintains the desired system pressure within the fire fighting system. Typical performance for a jockey pump is 10 gpm (38 l/min) at heads up to 300 ft (92 m).
- A first intervention fire fighting pump (Figure 2) capable of feeding a limited number (up to 5) of sprinklers. This pump is typically driven by an electrical motor and is fed from a reservoir capable of supplying at least one hour of liquid flow without interruption at required flow and pressure. Records indicate that as many as 95% of all fires are extinguished by this first intervention pump. Typical performance for a first intervention pump is 250 gpm (950 l/min) at a head of 200 ft (60 m).
- An emergency fire fighting pump, capable of feeding all available sprinklers, and other supplementary fire fighting equipment, such as deluge installations and fire-hose nozzles.

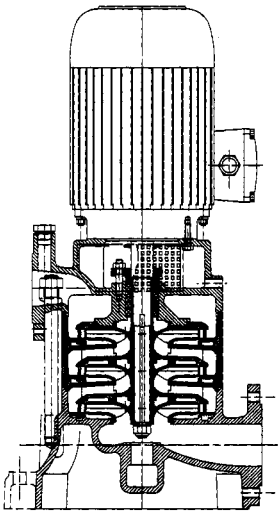


FIGURE 1 Small, vertical multistage jockey pump using an extended motor shaft (Flowserve Corporation)

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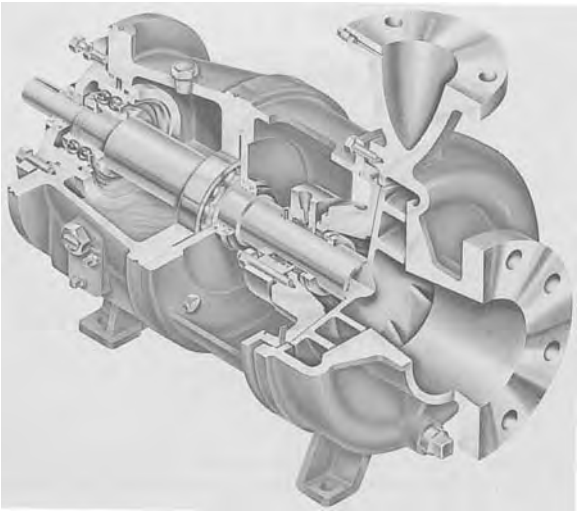


FIGURE 2 End suction, top discharge pump (Flowserve Corporation)

This pump is typically driven by a diesel engine and supplied with fire fighting water from a (practically) inexhaustible source. The emergency fire fighting pump standard in North America is a split case pump (Figure 3), whereas the European standard is an end suction pump (Figure 2). In each case, typical pump performance is from 500 to 5000 gpm (100 to 1200 m^3/h) at a head of 260 ft (80 m).

- High-rise buildings, protected by sprinklers or standpipes and hose systems, require an overpressure pump. This pump is generally a close coupled end suction pump (Figure 4) driven by an electrical motor, with typical performance of 150 gpm (34 m^3/h) and a head (according to the building height) of at least 200 ft (60 m), sourced from a public water network.

Although the various national authorities have different approaches to the details of fire fighting equipment and arrangement, they nevertheless have a similar basic philosophy. The pumps utilized in fire protection systems are usually classical designs. Specifications require skillful packaging of the complete fire protection system, including the pump(s), driver(s), starting and control devices, and other accessories.

Most European codes give considerable freedom of choice relative to pump type and design, whereas NFPA 20 (United States) is more restrictive. According to NFPA, the pumps must be of horizontal end suction or horizontal split-case design. Single stage close coupled vertical in-line pumps may be used for limited capacities, but these pump types

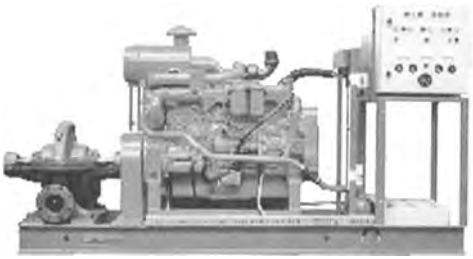


FIGURE 3 Split case fire pump (Flowserve Corporation)

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FIGURE 4 Close-coupled end suction pump (Flowserve Corporation)

may not be used where a static suction lift is required. If the pump must lift water from a well, panel, river, and so on, NFPA 20 requires the use of vertical pumps installed so the static water level is never below the bottom impeller.

Very high importance is given in all regulations to permanent availability of fire fighting water. Pumps must always be filled with water and must be ready for starting at any moment. This requirement is obviously satisfied when NFPA 20 compliance is mandatory. Although the same approach is preferred in Europe, some European codes nevertheless allow the use of horizontal pumps with a static suction lift. In such cases, the suction pipe must be fitted with a foot valve, and the pump set must be equipped with a controlled filling device.

Some rules common to both North America and Europe are

- The shape of the pump performance curves must conform to precise specifications.
- The design pressure of the pump casings is regulated.
- Hydrostatic tests are required.
- Certified performance test curves are required for most fire pumps.
- Fire fighting pumps must be approved, 'de jure' (according to written law or regulations) in North America and in most European countries; 'de facto' (according to well-established practices) in the others.

Stationary fire pumps can be purchased as packaged systems, which saves installation time and money. Single or multiple units of horizontal or vertical pumps can be packaged, and each packaged unit generally includes the pump(s), driver(s), controller(s), headers, accessories, and piping mounted on a common base. To the extent possible, all wiring and piping connections are made and the unit is factory-tested. Installation consists simply of positioning and leveling the package and making external piping and electrical connections.

FIRE PUMP DRIVERS

The principal objective of a fire pump driver is to provide the pump with motive power under any circumstances. Usual drivers are electric motors, diesel engines or, to a lesser extent, steam turbines. Although the reliability of the drivers themselves does not really pose problems, careful consideration must be given to the dependability of the power supply for electric motors and turbines, and to the fuel supply for diesel engines. Very complete regulations exist in every country which must be considered.

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Electric motors are the most economical driver type when a reliable power source is available. If this is not the case, power must be supplied by two or more independent sources, one of which may be an emergency generator set. Another choice might be to use one motor-driven pump and one diesel engine-driven pump. General guidelines for electric drives are

- Motor power and speed must be selected in accordance with the pump characteristics.
- Depending upon codes and available current, wound rotor, star delta, wye delta primary resistance, or part-winding start motors may be used.

Diesel engines are frequently used to drive stationary fire pumps. Equipped with battery packs and automatic controls, they rival electric motors for reliability and eliminate concern over the dependability of the source of electric power. In order to guarantee the fuel supply, the tank must be located above ground at an appropriate height. The prevailing philosophy of a diesel engine driven pump is that 'the pump must run' in actual fire conditions. All engine failures (oil pressure, cooling water temperature, speed) should be indicated only, and should not stop the engine. The only exception to this is the NFPA 20 requirement that a shutdown occur at an over speed of 120%.

Here are some general guidelines for diesel engines:

- Engine power and speed must be selected in accordance with the pump characteristics (diesel engines are very sensitive to altitude and ambient temperature).
- All energy sources for engine starting must be doubled: two batteries, or two air containers, or electrical and manual recharge of the hydraulic starting system, and so on.
- An instrument panel secured to the engine at an appropriate place must include a tachometer, an hour meter, an oil pressure gage, and a water temperature gage.

Occasionally, a fire pump is driven by a steam turbine. When it is desirable to use steam as the power source, details of the steam supply and exhaust need to be carefully planned to ensure that the required level of reliability exists.

FIRE PUMP CONTROLLERS

Regardless of the type of driver, most fire pumps are started automatically by a pressure signal from the pump discharge line. Each fire pump must have its own controller, including the jockey pump.

Depending upon application requirements, fire pump electric motor controllers must conform to the appropriate national regulations. Each controller must be arranged to match the starting characteristics of its motor and must include

- Manual disconnect switch
- Circuit breaker
- Starter without heaters or contactors
- Pressure switch (when NFPA is mandatory)
- Minimum-run timer to prevent motor cycling

The diesel engine controller is arranged to permit either automatic or manual start. The manual start is required to permit periodic run tests. The power source for the controller and for starting is a dual set of batteries or air containers. The controller is arranged to show the following engine conditions:

- Low lubrication oil pressure
- High water jacket temperature
- Failure to start automatically

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- Shutdown due to overspeed (when NFPA 20 is mandatory)
- Battery and battery charger failure or, when equipped with one, an air system failure

The engine controller must also provide a means of relaying the following information to remote indicators:

- Engine running
- Engine switch off or manual position
- Trouble signal (activated by one or any combination of the engine or controller signals described above)

Approval of the control system for all pumps, except the jockey pump, is required in nearly all national codes.

GENERAL ENGINEERING PROCEDURE FOR FIRE PROTECTION SYSTEMS

As described earlier, fire protection systems can vary greatly depending on the type and significance of the fire risk, the site conditions and, finally, the various national regulations in force in the given country. Outlined below are engineering guidelines for stationary installations using water as protection agent. Figures 5 and 6 show typical installations for horizontal and vertical fire pumps.

Here are some project basics to keep in mind:

1. Refer to an insurance expert, or to a specialist of a certification board, to confirm that water may be used for the risks of concerned and to determine which kind of spray system is recommended: sprinklers, water jets, deluge installations, and so on.
2. Determine the appropriate governing regulations: NFPA (North America), APSAD (France), VDS (Germany), and so on.
3. Ensure the availability of sufficient water capacity, according to the type of risk. Preference should be given to a water supply above ground, which is generally more reliable and less expensive than a below-ground supply.

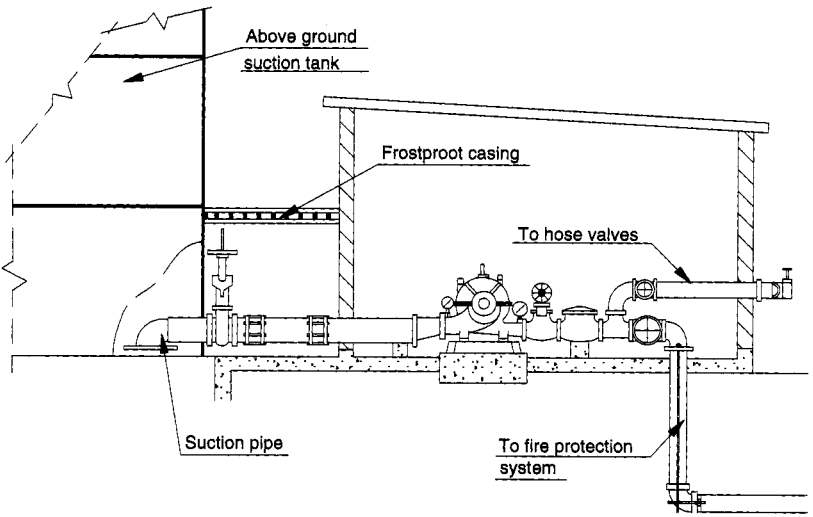


FIGURE 5 Typical installation for horizontal fire pumps

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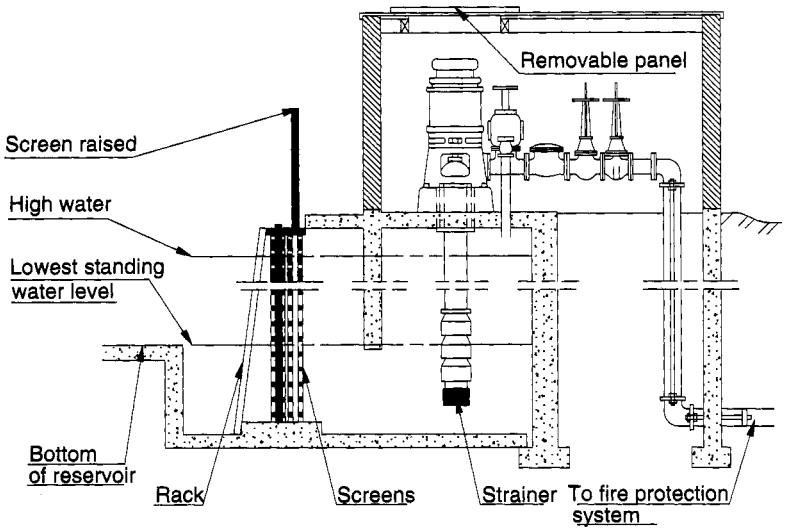


FIGURE 6 Typical installation for vertical fire pumps

4. Distinguish all the critical areas and functions to be protected on the site.
5. Determine the motive power of the pump drivers. The objective is to be independent from the site of risk. Diesel engine-driven installations many times provide a convenient, independent power source that addresses this concern. The costs associated with a diesel engine power source is usually equivalent to those of electrical installations; in some cases, they are even lower.
6. Provide a pump house, or a pump room, so the pump, driver and controller may be protected against possible damage or injury. In addition to the normal precautions and concerns of a pump installation, also consider the following factors:
   - Easy access for installation and maintenance
   - Sufficient ventilation for motors and engines
   - Risk of earthquakes
   - Risk of freezing
   - Noise protection, where required
7. Determine the pipe network and all required fittings and accessories in order to obtain the lowest project costs.
8. Smaller pipe sizes (less expensive) increase friction losses and may require more powerful (more expensive) pump sets.
9. Larger pipe sizes (more expensive) decrease friction losses and may allow the use of less powerful (less expensive) pump sets.

Keep in mind the following factors when selecting a pump:

1. Compute the total capacity and pressure required to feed the protection system.
2. Determine the NPSH available at the pump suction nozzle. This is an important selection criterion for establishing the speed and design of the pump. Reminder: U.S regulations forbid static suction lifts for fire pumps, whereas most European regulations will allow limited suction lifts.

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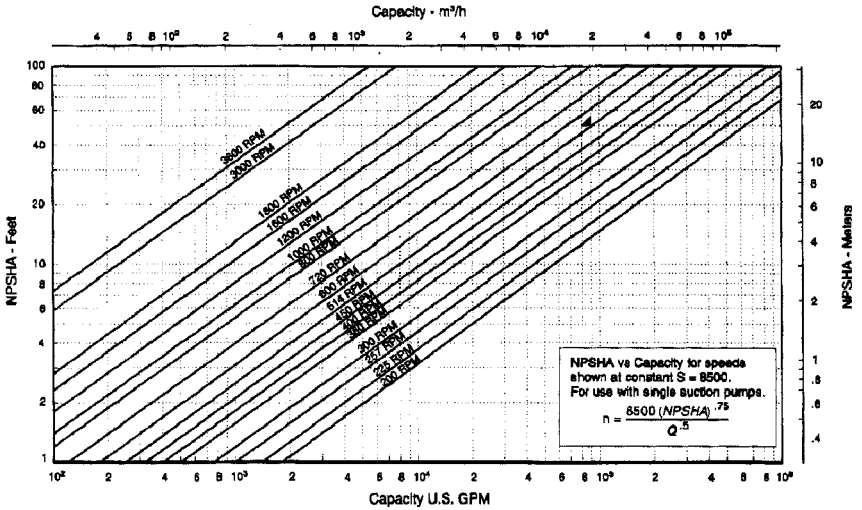


FIGURE 7 Recommended operating speeds for single suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Refernce 1)

3. Select the highest possible pump speed based on the required capacity and the available NPSH. The higher the allowable pump speed, the lower the cost of the pump set. Figures 7 and 8 give recommended operating speeds for single suction and double suction pumps.
4. Select pump size and design according to the governing (applicable) regulations.

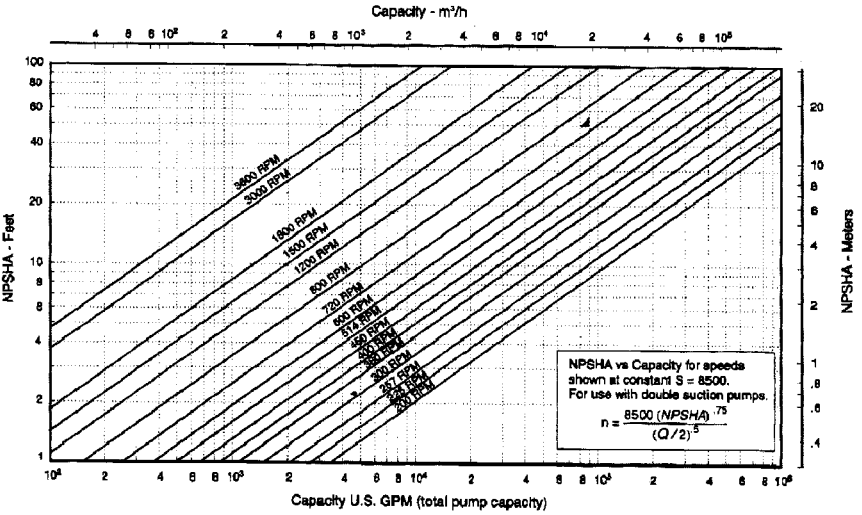


FIGURE 8 Recommended operating speeds for double suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Refernce 1)

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Drivers and Controllers As outlined earlier under 'Fire Pump Criteria,' it is recommended that a completely packaged fire pump set, including the pump(s), driver(s), controller(s), headers, accessories and piping, mounted on a common base, be furnished.

PUMP DESIGN AND MATERIALS

Constant operational readiness and a high degree of operational reliability are the most important aspects of fire pump designs. Although not mandatory, the features listed and described next are recommended for fire pumps.

- Shaft sealing should be by means of stuffing boxes (packing).
- Grease-lubricated rolling element bearings should be used on horizontal pumps. Water-lubricated (rubber) sleeve bearings (open line shaft) or oil-lubricated bronze sleeve bearings (enclosed line shaft) and, if so equipped, a grease-lubricated thrust bearing should be used on vertical pumps.
- Combinations of materials, running clearances, and rotor end plays should permit trouble-free operation, even if the pump has been idle (not run) for extended periods of time.
- Repairs should be achievable by means of standard tools.
- Corrosion-resistant materials must be utilized if the fire-fighting water is corrosive, such as brackish or salt water.
- Additional features or specific materials, as may be mandated by specific national standards and codes.

Because most fire pumps handle clear water, and because wear (from severe usage) is not usually a problem, fire fighting pumps should be of the simplest design possible and should be made of commonly used industrial materials. The most economical fire pump is either a single-stage, end suction, or vertical in-line pump. This class of pump is readily available to heads of approximately 260 ft (80 m). Unfortunately, capacities have been limited in the United States (by NFPA 20) to 750 gpm (200 m^3/h) for horizontal end suction pumps and to 500 gpm (115 m^3/h) for vertical in-line pumps. The 1996 edition of NFPA 20 eliminated these historical limitations. In Europe, however, end suction volute pumps are commonly employed up to 4000 gpm (900 m^3/h).

The jockey pump (Figure 1), used to maintain the pressure within the system, is generally a small, vertical multistage pump, utilizing an extended motor shaft (no shaft coupling). Recommended materials for jockey pumps are

Casings:	Cast iron
Impellers:	Bronze or thermoplastic resin
Diffusers:	Cast iron
Shaft:	Chromium steel
Bearings:	Grease lubricated rolling element bearings, with an additional product lubricated bronze line bearing in close coupled designs

A large end suction fire fighting pump, as shown in Figure 2, is typical of units used in Europe and allowed to be operated over the entire pump capacity range, from shutoff to the highest capacities. The pump exhibits the following features:

- Back pull out construction for ease of maintenance without disturbing the driver
- The absence of shaft sleeves, giving highest possible shaft stiffness
- A bronze bearing bushing located behind the impeller that acts as a product lubricated auxiliary line bearing
- A conical (tapered shaft) impeller fit to ensure no looseness between the mounted impeller and the shaft (critical when pumps are driven by a diesel engine)

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Recommended materials for end suction fire fighting pumps are

Casing:	Cast iron or ductile iron
Impeller:	Bronze
Shaft:	13% chromium steel
Bearings:	Grease-lubricated rolling element bearings

For larger capacities, in excess of 500—750 gpm (115–200 m^3/h) in the United States and above approximately 4000 gpm (900 m^3/h) in Europe, single stage double suction axially split case pumps (Figure 9) are required. These larger units are more costly than the simple single stage overhung designs, but they do offer some advantages that could make them desirable even at lower flow rates, such as

- The required NPSH is lower (double suction impeller), thus allowing higher operating speeds.
- The axially split case design provides easy access to the pump interior (neither the pump nor the driver need to be disconnected) for inspection and rotor removal.
- Axially balanced hydraulic axial forces, and bearings on each end of the rotor supports unbalanced radial loads.

Recommended materials for single stage double suction axially split case for fighting pumps are

Casing:	Cast Iron
Impeller:	Bronze
Shaft:	Carbon steel
Sleeves:	Bronze
Casing Rings:	Bronze
Bearings:	Grease-lubricated rolling element bearings

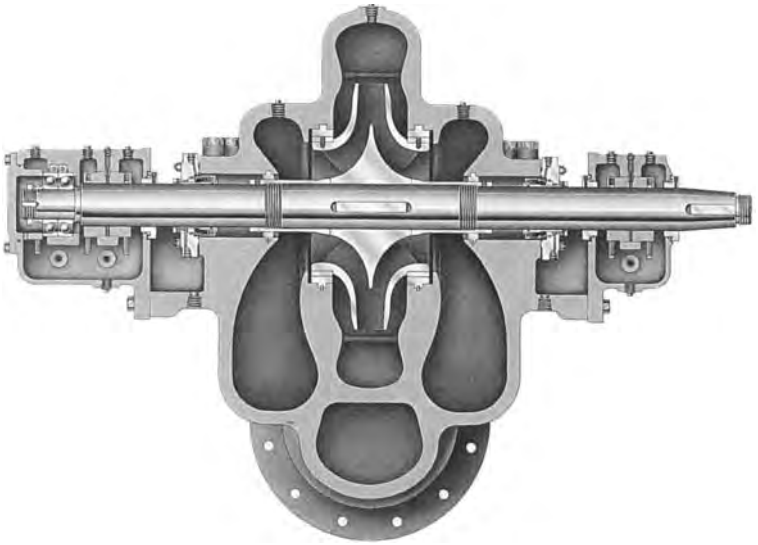


FIGURE 9 Single stage double suction axially split case pump used for larger flow rates (Flowserve Corporation)

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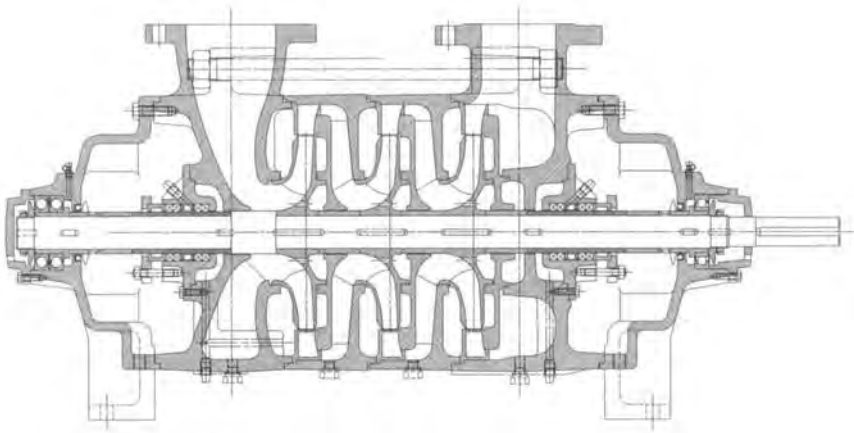


FIGURE 10 Multistage ring-section fire pump (Flowserve Corporation)

Normally, head requirements for fire pumps do not exceed 260 ft (80 m). For those systems that do require more head, end suction and axially split case pumps may be available up to 450 ft (140 m). Beyond this head range, multistage pumps are typically employed. Multistage pumps for fire services in Europe are usually of the ring-section design (Figure 10), whereas the axially split case design (Figure 11 shows a typical two-stage pump; Figure 12 shows a typical multistage pump) tends to be more popular in North America.

In North America, vertical pumps must be used when the water source is located below ground. In Europe, where the use of a limited static suction lift is generally permitted, such vertical pumps are required only when the NPSH available at the suction nozzle of a horizontal pump would exceed a certain limit, as may be established by national regulations in each country—for instance 16.4 ft (5 m) in France.

The submergence of vertical pumps require a particular care. As a minimum, the second stage (second impeller up from the bottom end of the pump) should always be submerged below the minimum water level.

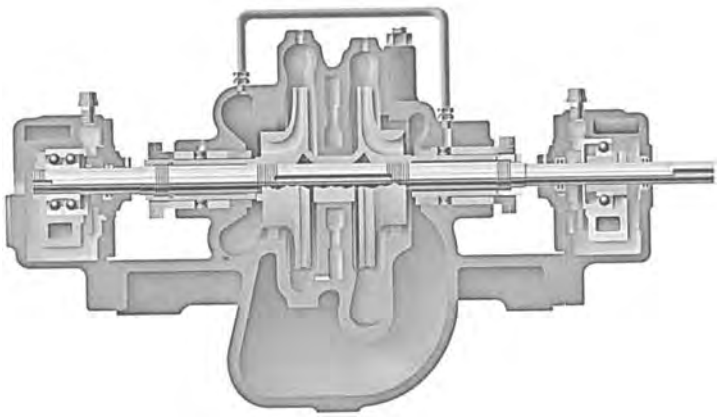


FIGURE 11 Two-stage axially split pump (Flowserve Corporation)

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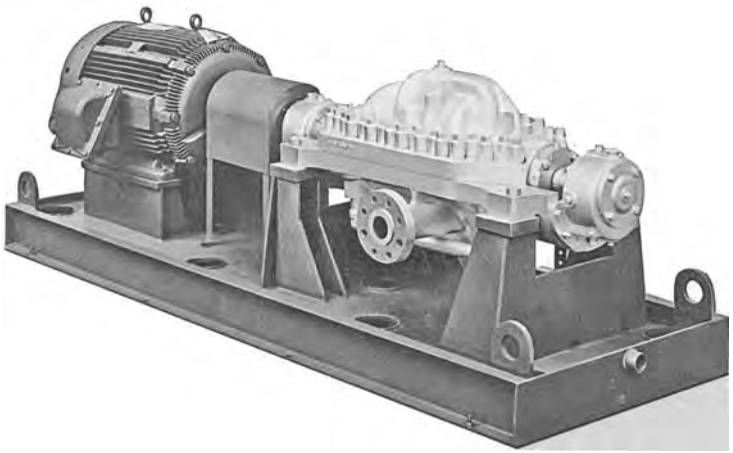


FIGURE 12 Typical axially split multistage pump (Flowserve Corporation)

The inner column (drive shafting) for vertical pumps is available in either of two configurations:

- Open line shaft (OLS) construction (Figure 13), within which a steel shafting rotates in water-lubricated rubber bearings, centered and stabilized by rigid bearing retainers, is used for static water levels 50 ft (15 m) or less below the pump discharge flange.
- Enclosed line shaft (ELS) construction (Figure 14), in which steel shafts rotate in oil-lubricated bronze sleeve bearings, is used for static water levels of more than 50 ft (15 m). In ELS construction, the outside of the sleeve bearings are threaded and tubes enclosing and supporting the bearings and shafts isolate the shafting from the water being pumped.

If an oil-lubricated line shaft design is utilized, the appropriate environmental protection or health department should be consulted with regard to special requirements for installation and protection.

Recommended materials for vertical pumps are

Discharge head and columns:	Cast iron or welded steel
Water-lubricated line shaft:	Carbon steel protected by nonrusting shaft sleeves through packing and water-lubricated bearings (Note: chromium steel shafts typically do not require shaft sleeve protection.)
Oil-lubricated line shaft:	Carbon steel possible, but chromium steel is preferred
Pump shaft:	Chromium steel
Bowls:	Cast iron
Impellers:	Bronze

OVERVIEW OF COMMONLY USED ACCESSORIES

See Figures 15 and 16 for typical fire pump systems.

1. Common to horizontal and vertical pumps:
   a. **Casing relief valve** To prevent a no-flow condition when system is running at shutoff

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FIGURE 13 Vertical pump with open line shaft construction (Flowserve Corporation)



FIGURE 14 Vertical pump with enclosed line shaft construction (Flowserve Corporation)

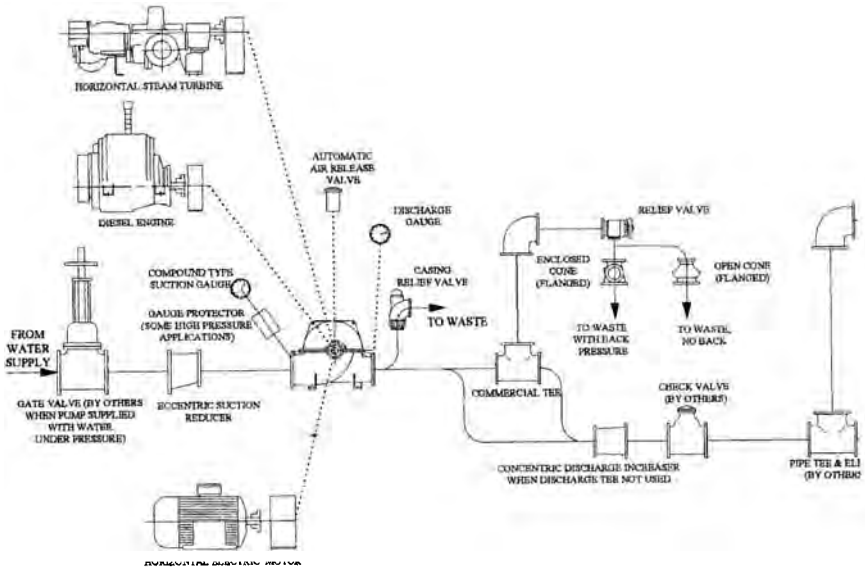


FIGURE 15 Typical horizontal fire pump system

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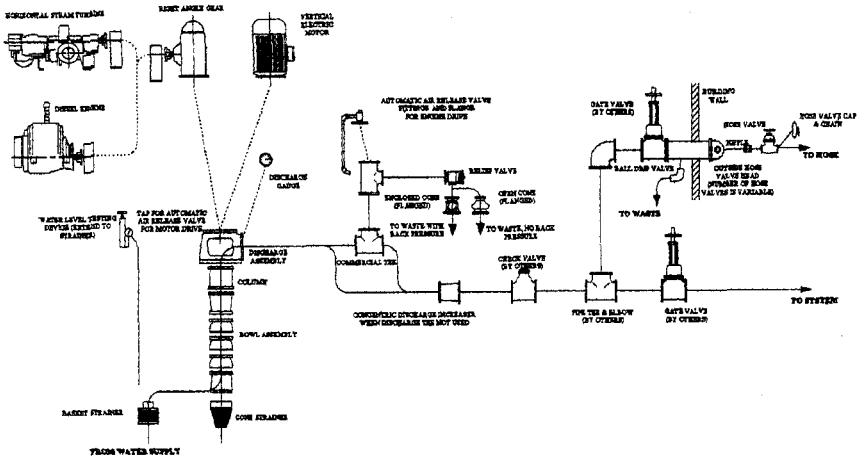


FIGURE 16 Typical vertical fire pump system

   b. **Automatic air release valve** To vent entrapped air in automatic systems
   c. **Umbrella cock** To vent air in manually operated systems
   d. **Hose valve head with hose valves, caps, and chains** To permit pump flow (capacity) tests (most modern systems use flowmeters in lieu of hose valve heads)
   e. **Ball drip valve** Installed upstream to prevent freeze damage to hose valve head installed outside
   f. **Overflow cone** To show whether the relief valve is open
   g. **Commercial discharge tee with (if required) 90° elbow** Used when main relief valve is required
2. Horizontal pumps only
   a. **Eccentric suction reducer** Required when the size of the suction pipe does not match the size of the pump suction connection
   b. **Concentric discharge increaser** Required when the size of the discharge pipe does not match the size of the pump discharge connection
   c. **Main relief valve** Required when pump shutoff pressure plus suction pressure exceeds system design pressure and when engine or other variable-speed driver is used
   d. **Splash partition** Used for motor-driven units where hose head are mounted indoors near pump
3. Vertical pumps only
   a. **Main relief valve** Required when engine or other variable-speed driver is used
   b. **Water level testing device** To determine distance to surface of water; required for well pump installations

REFERENCES AND FURTHER READING

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SECTION 12.5
STEAM POWER PLANTS

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STEAM POWER PLANT CYCLES

Power is produced in a basic steam power plant by pressurizing water, adding heat energy to produce steam under pressure, and transforming part of that steam's thermal energy into mechanical energy to do useful work. The pressurized water, called feedwater, acts as a conveyor of energy. Modern steam power plants use centrifugal pumps to deliver pressurized water to some heat source such as a boiler, steam turbines as heat engines to transform thermal energy into mechanical energy, and condensers to recover the steam in a continuous cycle. Most steam power plants produce electric power with generators driven by the steam turbines.

This basic cycle is improved by connecting a condenser to the steam turbine exhaust and by heating the feedwater with steam extracted from an intermediate stage of the main turbine. This results in an improvement of the cycle efficiency, provides deaeration of the feedwater, and eliminates the introduction of cold water into the boiler and the resulting temperature strains on the latter. The combination of the condensing and feedwater heating cycle (Figure 1) requires a minimum of three pumps: the condensate pump, which transfers the condensate from the condenser hot well into the direct-contact heater; the boiler-feed pump; and a circulating pump, which forces cold water through the condenser tubes to condense the exhaust steam. This cycle is very common and is used in most small steam power plants. A number of auxiliary services not illustrated in Figure 1 are normally used, such as service water pumps, cooling pumps, ash-sluicing pumps, oil-circulating pumps, and the like.

The required improvements in operating economy in the 1970s dictated further refinements in the steam cycle, and these created new demands for power plant centrifugal pumping equipment. This evolution involved a steady increase in operating pressures until 2400 lb/in^2 (165 bar*) steam turbines became quite common. Many plants are operating at supercritical steam pressures of 3500 lb/in^2 (240 bar). Several central station plants constructed in the late 1970s are operating between 4000 and 5000 lb/in^2 (275 and 345 bar).

Other refinements were directed toward a greater utilization of heat through increased feed-water heating, introducing a need for heater drain pumps-equipment with definite

*1 bar = 10^5 Pa.

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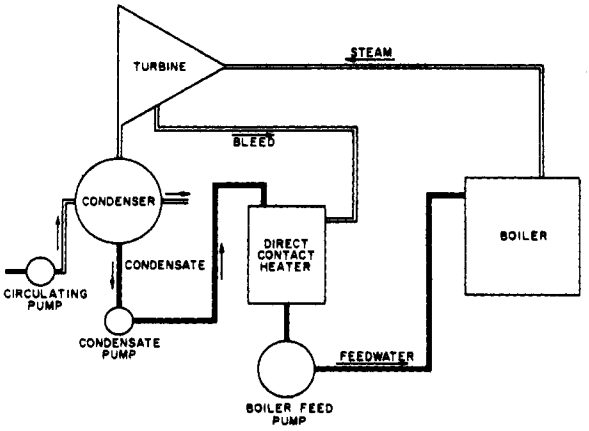


FIGURE 1 Simple steam power cycle

problems of its own. Finally, the introduction of forced or controlled circulation as opposed to natural circulation at 650°F (343°C) in boilers created a demand for pumping equipment of again an entirely special character.

Although direct-contact heaters would have thermodynamic advantages, a separate pump would be required after each such heater. The use of a group of closed heaters permits a single boiler feed pump to discharge through these heaters and into the boiler. The average power plant is based on a compromise system: one direct-contact heater is used for feedwater deaeration, whereas several additional heaters of the closed type are located upstream as well as downstream of the direct-contact heater and of the boiler-feed pump (Figure 5). Such a cycle is termed an *open cycle.* The major variation is the *closed cycle,* where the deaeration is accomplished in the condenser hot well and all heaters are of the closed type (Figure 6).

In the 1970s, steam power plant technology advanced with the implementation of nuclear fission reactors as heat sources in commercial electric power generation. Steam cycles with nuclear fission reactors utilize two concepts for producing steam: boiling water reactors (Figure 3) in which feedwater is delivered directly to the reactor and pressurized water reactors (Figure 4) in which reactor heat energy is transferred indirectly to the feedwater by steam generators. In both cases, the basic steam cycle is the same as in steam power plants with conventional fossil fuel-fired boilers. However, feedwater pressures for nuclear fission reactors are substantially less feedwater pressures for modern fossil fuel-fired boilers.

In the 1980s, steam power plant technology was reapplied to recover exhaust energy from gas turbines and thereby create an efficient form of electric power generation. These combined-cycle power plants have multiple electric generators driven by combinations of gas turbines and steam turbines (Figure 2). The heat source for the steam cycle is called a heat recovery steam generator (HRSG), which transfers heat energy from the hot exhaust gases of a gas turbine to the feedwater. The pumps in these steam cycles are similar to, but usually smaller than those typically found conventional fossil fuel-fired boilers.

In the 1990s, steam power plant technology advanced yet again with new boiler designs and improved emission controls. Once-through supercritical designs improve the heat rates and commercial viability of fossil fuel-fired boilers. Fluidized bed-type boilers and flue gas desulfurization subsystems, such as wet-limestone scrubbers, reduce emissions from coal-fired plants. The reemergence of these coal-fired plants has required larger, higher-energy feedwater pumps and new slurry-type pumps.

Presently, the world's electric power producers are utilizing and constructing all three types of power plants: combined-cycle, nuclear reactor, and fossil fuel-fired boiler. This

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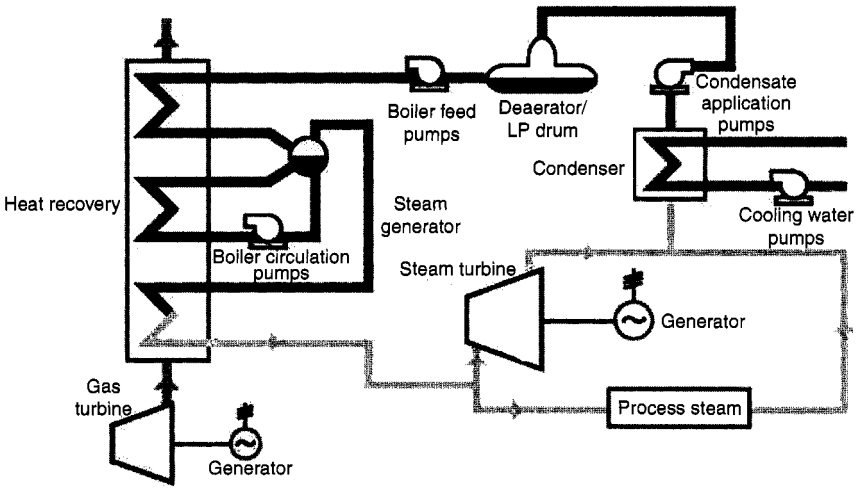


FIGURE 2 Natural gas and steam, combined cycle power

multiple-type approach is expected to continue as power producers worldwide strive to serve ever growing electric energy demands and as changes in technologies, local fuel resources, environmental needs, and commercial factors emerge. Going forward, pumps will continue to serve both traditional and specialized requirements in new steam power plants.

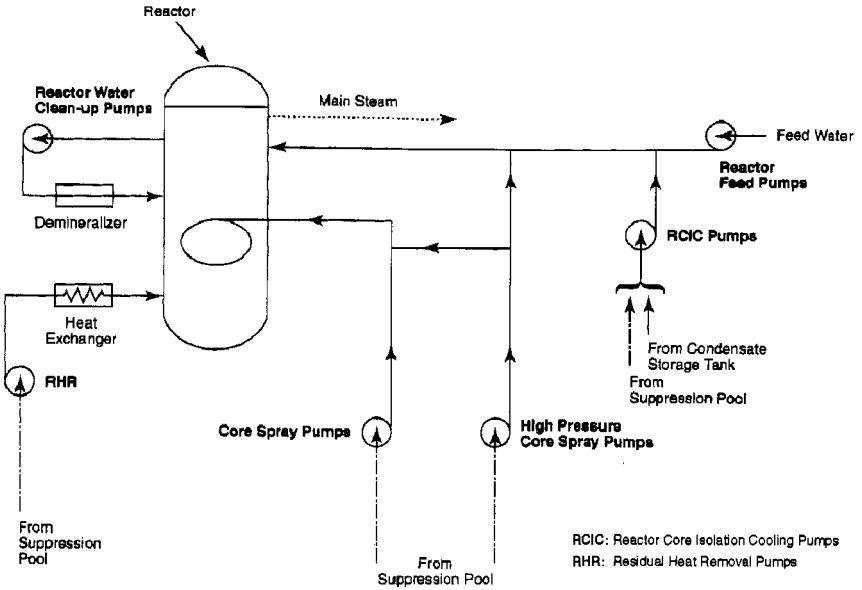
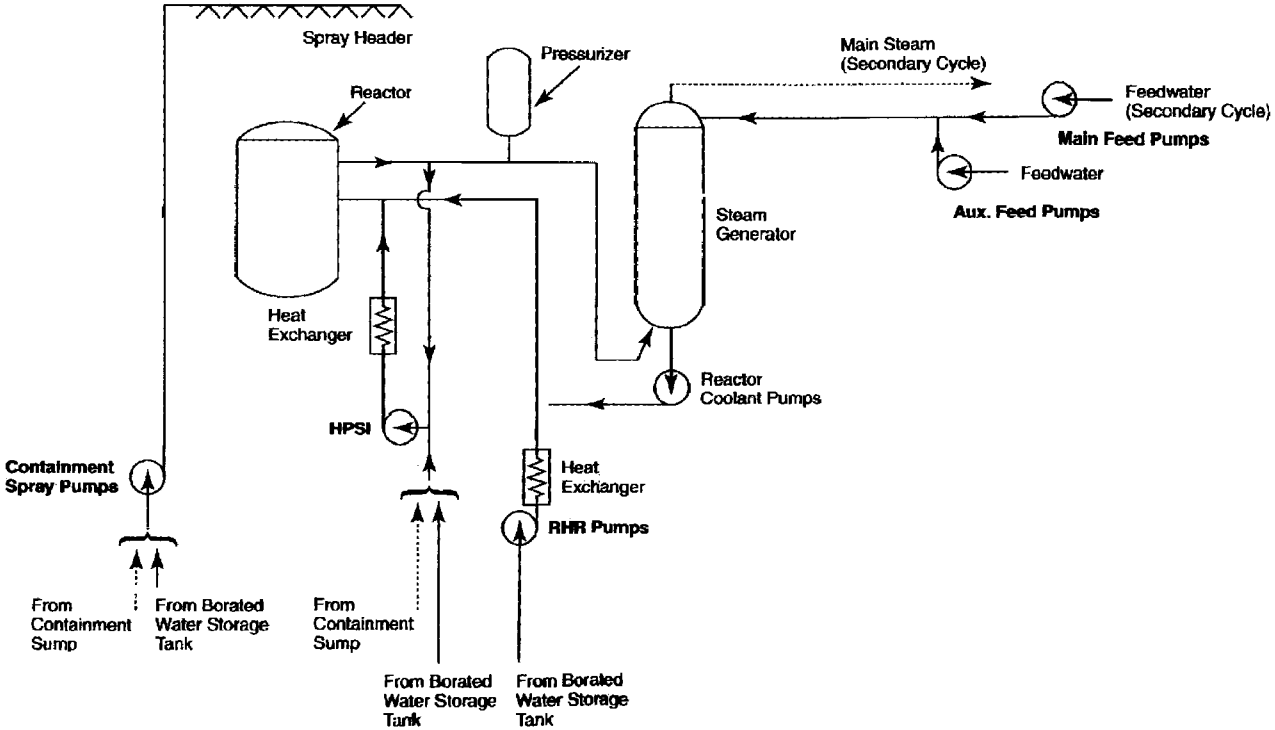


FIGURE 3 Nuclear power steam cycle, boiling water reactor

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RHR: Residual Heat Removal Pumps
HPSI: High Pressure Safety Injection Pumps

FIGURE 4 Nuclear power steam cycle, pressurized water reactor

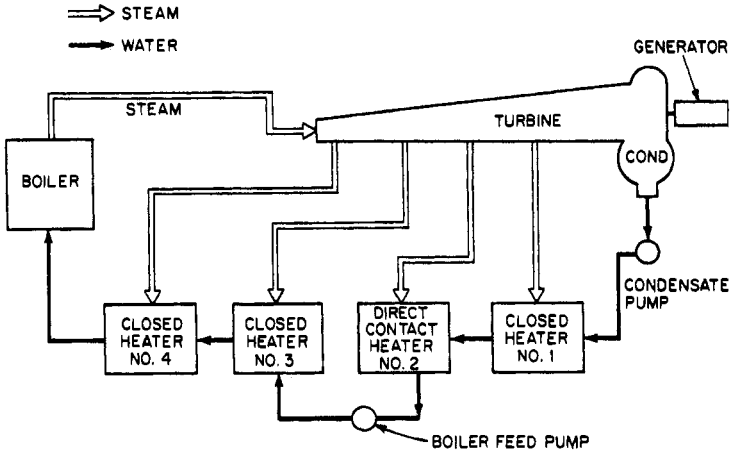


FIGURE 5 Open feedwater cycle with one deaerator and several closed heaters

STEAM POWER PLANT PUMPING SERVICES

Pumps are very important components of a steam electric power plant. The major applications are the condensate, boiler-feed, heater drain, and condenser circulating pumps. The all-inclusive category of "miscellaneous pumps" includes such a variety of services that it merits being broken down into its components and included in a representative listing. Table 1 provides such a listing for conventional (fossil fuel) steam power plants. The list is not necessarily complete but is reasonably representative.

BOILER-FEED PUMPS

Under the term *conditions of service* are included not only the pump capacity, discharge pressure, suction conditions, and feedwater temperature but also the chemical analysis of

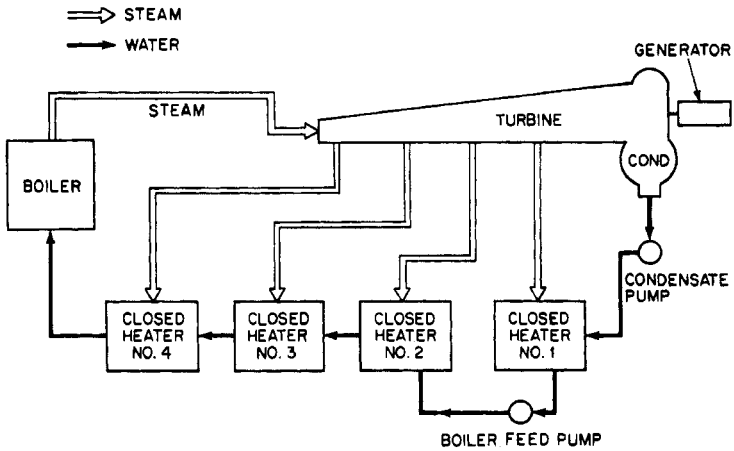


FIGURE 6 Closed feedwater cycle

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TABLE 1 Pump services in conventional steam power plants

- Turbogenerator and auxiliaries
  - Condenser circulating pumps
  - Screen wash-water pumps
  - Cooling tower make-up pumps
- Steam generator equipment
  - Condensate pumps
  - Condensate booster pumps
  - Boiler-feed pumps
  - Boiler-feed booster pumps
  - Deaerator make-up pumps
  - Heater drain pumps (low and high pressure)
- Chemical feed system
  - Amine pumps
  - Hydrazine pumps
  - Phosphate pumps
  - Caustic feed pumps
  - Acid feed pumps
  - Ammonia pumps
  - Regeneration waste pumps
  - Demineralizer pumps
  - Neutralizing metering pumps
  - Neutralizing tank sump pumps
  - Acid bulk-transfer pumps
  - Caustic bulk-transfer pumps
  - Inlet and effluent demineralizer waste tank pumps
- Fuel oil system
  - Fuel oil transfer pumps
  - Secondary fuel oil pumps
  - Secondary fuel oil heater drip pumps
  - Ignitor oil pumps
  - Auxiliary boiler fuel pumps
  - Warm-up oil pumps
  - High-temperature oil-circulating pumps
- Fuel oil system (continued)
  - Low-temperature oil-circulating pumps
  - Distillate oil unloading pumps
  - Fuel oil additive unloading pumps
  - Fuel oil additive transfer pumps
  - Fuel oil additive metering pumps
  - Fuel oil hose drain pumps
- Lubricating oil system
  - Lubricating transfer pumps
  - Starting oil pumps
  - Main oil pumps
  - Emergency oil pumps
  - Centrifuge feed pumps
- Fire protection system
  - Fire pumps
  - Jockey pumps
  - Foam proportioning pumps
- Heating, ventilating, and air conditioning system
  - Hot water circulating pumps
  - Chilled water pumps
- Service water system
  - Service water pumps
  - Air preheater wash pumps
  - Cooling water booster pumps
  - Primary air heating coil condensate return pumps
  - Heating drain tank return pumps
  - Sump pumps
  - Closed cooling water system pumps
- Miscellaneous
  - Ash sluice pumps
  - Slurry pumps
  - Acid cleaning pumps
  - Hydrostatic pressure test pumps

the feedwater, the pH at pumping temperature, and other pertinent data that may reflect upon the hydraulic and mechanical design of the boiler-feed pumps. Preferably, a complete layout of the feedwater system and of the heat balance diagram should be supplied to the boiler-feed pump manufacturer. The study of this layout will often permit the manufacturer to suggest an alternate arrangement of the equipment that would result in a more economical operation, in a lower installation cost, or even in longer equipment life to reduce the eventual maintenance expense.

Boiler-Feed Pump Capacity Total boiler-feed pump capacity is determined by adding margins for system surge and pump wear to the maximum required flow of the boiler. Typical flow margins range from 5 to 10% depending on the size of the boiler. The total required boiler flow may be served by a single pump or by some combination of identical pumps operating in parallel.

A common boiler-feed pump arrangement for large central power plants includes two auxiliary stream turbine-driven main boiler-feed pumps each with 50% boiler capacity and one electric motor-driven start-up boiler-feed pump with 20 to 30% boiler capacity. However, arrangements vary widely depending on preferences of different owners and engineers. Some owners prefer the use of auxiliary boilers in lieu of start-up pumps. Some have used a single 100% boiler feedwater pump for units as large as 1300 MW (Figure 7). Other owners require installed spare pumps in cases where pump element change-out cycle times are too long for a given plant availability requirement.

An HRSG in a combined cycle power plant typically has multiple conditions of service with differing pressures and flow requirements. These different conditions can be served

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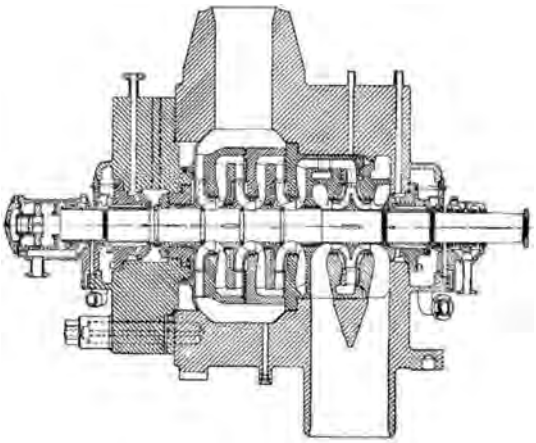


FIGURE 7 Cross-section, single 65,000 hp boiler feed pump, 1300 MW fossil power plant (Flowserve Corporation)

by unique high-pressure, medium-pressure, and/or low-pressure pumps. Or, they can be served by one high-pressure pump with medium- and/or low-pressure takeoff connections.

Suction Conditions The net positive suction head (NPSH) represents the net suction head at the pump suction, referred to the pump centerline, *over and above* the vapor pressure of the feedwater. If the pump takes its suction from a deaerating heater, as in Figure 5, the feedwater in the storage space is under a pressure equivalent to the vapor pressure corresponding to its temperature. Therefore the NPSH is equal to the static submergence between the water level in the storage space and the pump centerline less the frictional losses in the intervening piping. Theoretically, the required NPSH is independent of operating temperature. Practically, this temperature must be taken into account when establishing the recommended submergence from the deaerator to the boiler-feed pump. A margin of safety must be added to the theoretical required NPSH to protect the boiler-feed pumps against the transient conditions that follow a sudden reduction in load for the main turbogenerator.

Although the previous discussion applies primarily to the majority of installations, where the boiler-feed pump takes its suction from a deaerating heater, it holds as well in the closed feed cycle (Figure 6). The discharge pressure of the condensate pump or the booster pump must be carefully established so the suction pressure of the boiler-feed pump cannot fall below the sum of the vapor pressure at pumping temperature and the required NPSH.

Careful attention must be given to any strainer that might be installed in the pump suction piping. The pressure drop increase across the strainer is indicative of foreign material and it reduces the net positive suction head available (NPSHA) to the pump. Strainers in the pump suction pipe are most often removed following plant start-up qualification testing.

Transient Conditions Following Load Reduction Following a sudden load reduction, the turbine governor reduces the steam flow in order to maintain the proper relation between turbine and generator power and to hold the unit at synchronous speed. The consequence of this reduction is a proportionate pressure reduction at all successive turbine stages, including the bleed stage that supplies steam to the deaerator. The check valve in the extraction line closes and isolates the heater from the turbine. As hot feedwater continues to be withdrawn from the heater and cold condensate to be admitted to the heater, the pressure in the direct-contact heater starts to drop rapidly. The check valve reopens when the heater pressure has been reduced to the prevailing extraction pressure and stable conditions are reestablished.

It should be noted that, even though the feedwater system in a drum boiler may be provided with a three-element feedwater regulator, the feedwater flow will not instantaneously follow the steam flow as soon as the steam demand is reduced by a reduction in

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unit load. Because of the time lag between the reduction in steam demand and that of the fuel-burning rate and because of the heat retention in the steam generator, there is a momentary rise in the boiler pressure, with the resultant collapse of some of the steam and water bubbles in the boiler drum. This lowers the apparent boiler drum level, causing the level control to override to some degree the impulse from the change in steam flow. Therefore, there will generally be a definite lack of correlation between feedwater and steam flow following a sudden drop in load. The exact degree of the difference between these two flows will depend upon the particular type and setting of the feedwater controls. In some extreme cases, the feedwater flow after a sudden drop in load can actually exceed the feedwater flow at maximum design conditions. Thus, it is a safer practice to assume that the feedwater flow will not be reduced and to assume that the NPSH required will in turn correspond to at least its value under flow conditions preceding the drop in load.

In the interval, however, the pressure at the boiler-feed pump suction is reduced correspondingly. Unfortunately, until the suction piping has been completely voided of the feedwater it contained prior to the load reduction, its temperature and vapor pressure will not be reduced. As a consequence, the available NPSH will diminish and may become insufficient to provide adequate pump operation. In such a case, the pump will flash and serious damage may be incurred.

The factor that establishes the adequacy of an installation from the point of view of suction conditions after a load drop is the ratio between the direct-contact heater storage capacity and the suction piping volume. Based on a number of simplifying assumptions, a formula has been developed for the minimum value of this ratio:

$$\text{Minimum } \frac{Q_h}{Q_s} = \frac{h_{x0} - h_{c2}}{K_h H_x} \tag{1}$$

where Q_h = volume of feedwater in heater storage, gal (m^3)

Q_s = volume of feedwater in suction piping, gal (m^3)

h_{x0} = enthalpy of feedwater under initial conditions, Btu/lb (J/kg)

h_{c2} = enthalpy of condensate to heater under final conditions, Btu/lb (J/kg)

K_h = change in enthalpy with pressure at steam conditions prior to load reduction, Btu/lb · ft absolute pressure (J/kg · m) (Figure 8)

H_x = available excess *NPSH* = *NPSH* available − *NPSH* required, ft (m)

This relationship is somewhat conservative and does not take into account the residence time of the condensate in the piping and the closed heaters between the condenser hot well and the direct-contact heater. A slightly less conservative formula that takes some account of this residence time is

$$\text{Minimum } \frac{Q_h}{Q_s} = \frac{h_{x0} - [(h_{c0} + h_{c2})/2]}{K_h H_x} \tag{2}$$

where h_{c0} = enthalpy of condensate to heater under initial conditions, Btu/lb (J/kg)

For example, let

Initial heater pressure = 153 lb/in² (10.5 bar)

Initial feedwater temperature = 360°F (182°C)

Initial feedwater enthalpy = 331.4 Btu/lb (770.8 kJ/kg)

Final condensate enthalpy = 82.95 Btu/lb (192.9 kJ/kg)

K_h (from Figure 8) = 0.22 Btu/lb/ft (1679 J/kg/m)

H_x (available excess *NPSH*) = 15 ft (4.57 m)

Then

in USCS units $$\text{Minimum } \frac{Q_h}{Q_s} = \frac{331.4 - 82.95}{0.22 \times 15} = 75.3$$

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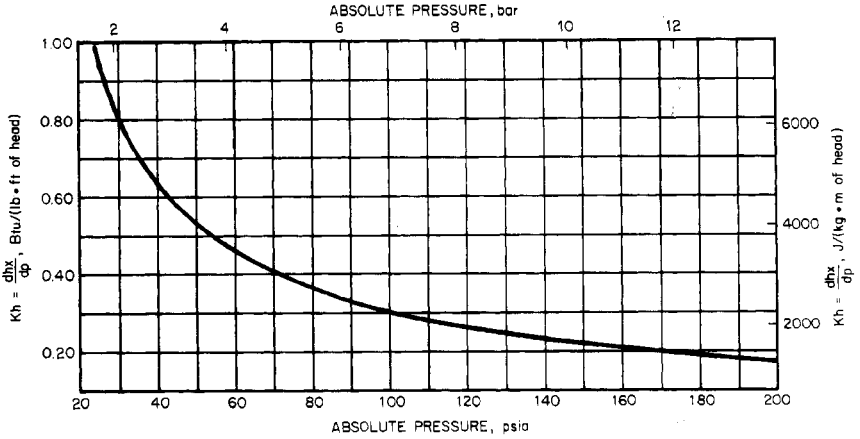


FIGURE 8 Enthalpy change with change of vapor pressure, for water

in SI units $$\text{Minimum } \frac{Q_h}{Q_s} = \frac{770{,}800 - 192{,}900}{1679 \times 4.57} = 75.3$$

This means that for safe operation after a sudden load drop in this particular ease, the heater storage volume must be at least 75.3 times the volume of the suction piping.

More complex and more rigorous calculations of the minimum ratio of heater storage volume to suction piping volume are provided in Reference 1.

Even where analysis indicates that the boiler-feed pumps are assured of their required NPSH during a reduction in turbine load, there is no guarantee that their operation will not be interrupted by flashing at some point in the suction piping. The criterion in determining the probability of flashing in the suction piping is to consider that the water that left the heater outlet at a saturated condition must pick up static pressure, by means of the vertical drop, at a rate at least equal to the pressure decay rate of the heater, or it will flash.

The most adverse conditions are those introduced by locating a horizontal run of piping too close to the heater outlet. A typical case is illustrated in Figure 9. (Because this example is used merely to illustrate the unfavorable effect of such a piping layout, the unit system used is immaterial and the example has been expressed in USCS units.) In the comparison of the two installations, we will stipulate that the total length and the sizes of the piping are the same for both arrangements and that the volumes of the suction piping between the heater outlet and points C and E of the two arrangements are the same. To simplify the comparison, the vertical distances between A and B, B and D, and A and E have been expressed in pounds per square inch instead of feet.

If a time interval x is selected such that water having left the heater outlet at the start of the transient conditions will have reached points C and E, respectively, at the end of the time interval, it becomes apparent that in the case illustrated on the left side of Figure 9, the pressure gain at point C is only 3 lb/in^2 by virtue of the vertical drop. Thus, x seconds after a pressure drop in the direct-contact heater from 52 to 48 lb/in^2 gage, the pressure at point C will be 51 lb/in^2, which is below the vapor pressure at the new temperature (296°F), and so flashing will occur. On the other hand, in the case of a straight vertical drop (right side of Figure 9), after the same time interval x the pressure will be 60 lb/in^2, which exceeds the vapor pressure, and so no flashing will occur. Formulas 1 and 2 can be used to determine the adequacy of the piping layout by selecting the critical point in the piping (in this case, point C) and substituting in the formulas so Q_s = volume in suction piping to point C and H_x = static head to point C less frictional losses to point C.

In the event that circumstances do not permit the provision of sufficient NPSH margin to provide adequate protection to the boiler-feed pumps during a sudden

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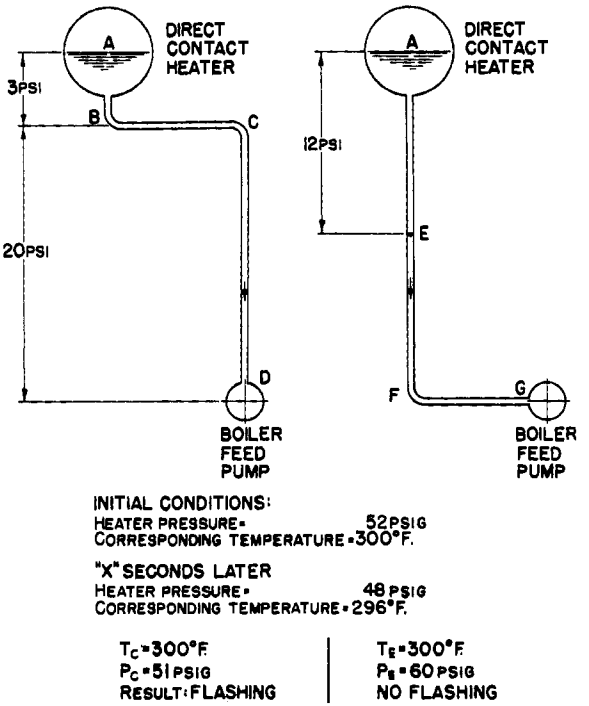


FIGURE 9 Comparison of suction piping arrangements

turbine load reduction, two alternate means are available to compensate for these circumstances:

1. A small amount of steam from the boiler can be admitted to the direct-contact heater through a pressure-reducing valve, to reduce the rate of pressure decay in the heater.
2. A small amount of cold condensate from the discharge of the condensate pumps can be made to bypass all or some of the closed heaters and be injected at the boiler-feed pump suction to subcool the feedwater, thus providing additional NPSH margin during load reduction.

Figure 10 illustrates the effect of subcooling (or temperature depression) on the available NPSH at various initial feedwater temperatures. Figure 11 shows, for varying ratios of injection flows, the temperature depression resulting from cold water injection plotted against the difference in temperature between the feedwater and the injection stream. For instance, if it were desired to provide 20 ft (6.1 m) additional NPSH to a boiler-feed pump that handles 325°F (163°C) water, the required temperature depression is 6°F (3.3°C). If the injection water temperature is 190°F (87.8°C), the difference between feedwater and injection water temperature is 135°F (75.2°C). From Figure 11, we can see that the injection flow must be 4.5% of the total feedwater flow.

An analysis of the relative merits of the two methods of protecting boiler-feed pumps against the unfavorable effects of transient conditions is presented in Reference 2. Either corrective action can be initiated automatically. This involves constant monitoring of the suction pressure and of the vapor pressure of the feedwater at the pump suction. The difference between the two is then constantly compared with a pre-established minimum NPSH. Any transient condition that causes the available NPSH to fall below this desired

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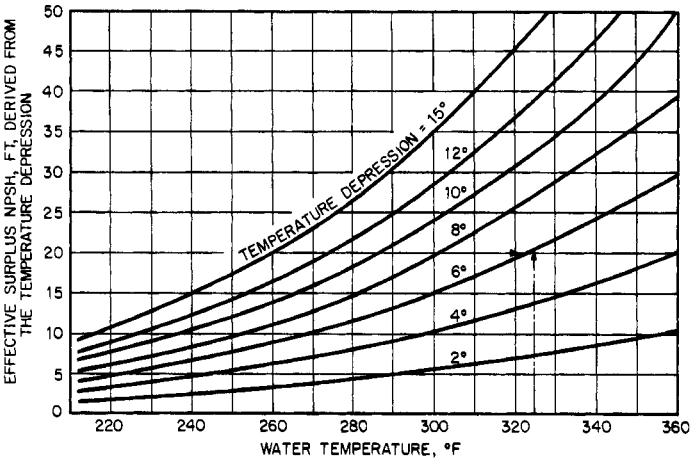


FIGURE 10 Effect of subcooling on available NPSH at various initial water temperatures [°C = (°F [min] 32)5/9; 1 ft = 0.3048 m]

minimum initiates corrective action, be it admission of cold condensate at the pump suction or admission of auxiliary steam to the direct-contact heater.

Another transient condition that will create a two-phase (steam-water mixture) flow at the pump suction—and in the pump—may occur during a "hot restart." When the plant experiences a trip, the pump is secured and the pressure in the deaerator drops. The temperature in the deaerator consequently drops within a relatively short period of time. The feedwater temperature has dropped from perhaps 350°F (175°C) to 250°F (120°C). The pump and suction piping near the pump remain at a higher temperature due to the mass of the metal (Figure 12). In a short time, the idle pump is switched on, and a two-phase flow condition occurs at the pump suction. Potential failure mode effects include suction cavitation, rotor upset and contact with stationary wear rings, and water hammer.

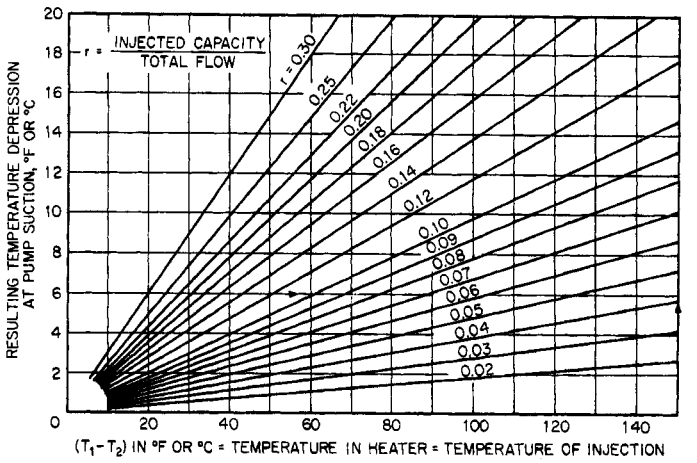


FIGURE 11 Required amount of cold water injection for a given temperature depression

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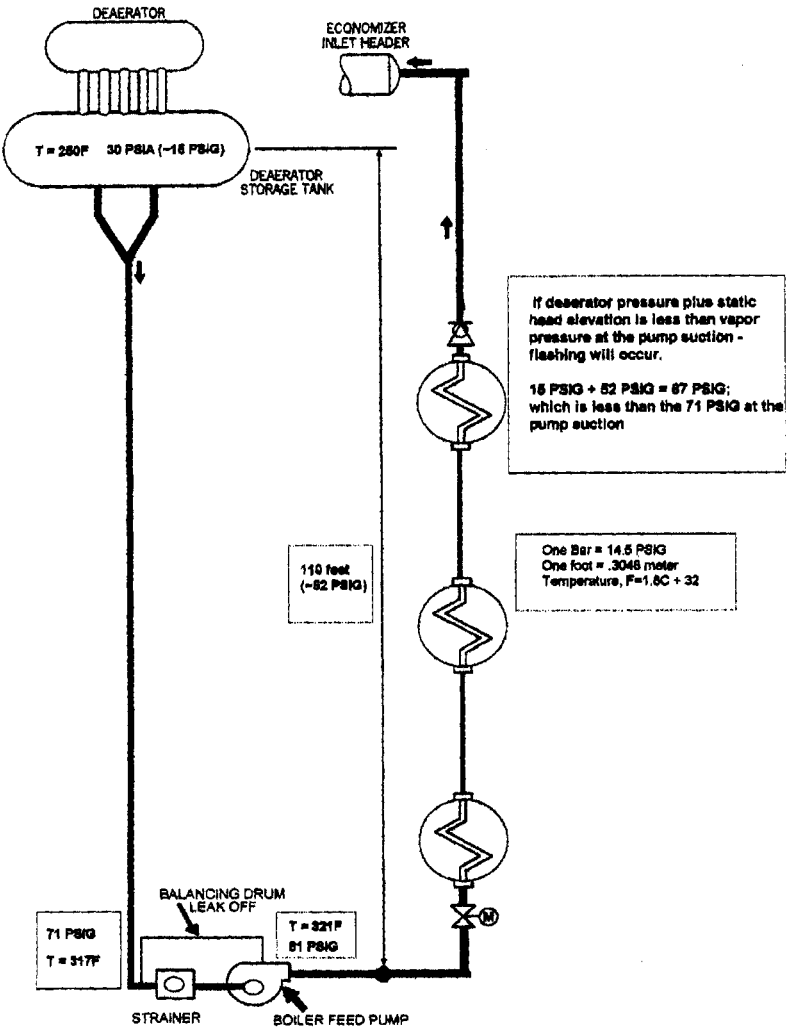


FIGURE 12 Feedwater system daigram, "hot restart" transient

BOOSTER PUMPS

Main boiler-feed pumps in central power plants are typically multistage pumps (three or more stages) operating at speeds between 5000 and 6000 rpm in order to delivery the required pressure to the boiler drum inlet. These large high-energy pumps need large values of NPSH required and benefit from large margins of NPSH available. In many cases, raising the elevation of direct-contact heaters high enough to provide sufficient NPSH available is costly and even impractical. Therefore, low-speed booster pumps are often utilized to maintain reasonable deaerator elevations and to provide sufficient NPSH available to the main boiler-feed pumps to avoid problems associated with cavitation.

Booster pumps for boiler-feed service are one- or two-stage pumps operating at lower speeds than the main pumps. They usually have double-suction first stage designs in order

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to minimize their NPSH required and thereby the height of the deaerator. Booster pumps can be driven through gearboxes from the same drivers as main boiler-feed pumps, or they can have separate electric motor drivers (Figure 42).

Discharge Pressure and Total Head The discharge pressure is the sum of the maximum boiler drum pressure and the frictional and control losses between the boiler-feed pump and the boiler drum inlet. The required discharge pressure will generally vary from 115 to 125% of the boiler drum pressure. The net pressure to be generated by the boiler-feed pump is the difference between the required discharge pressure and the available suction pressure. This must be converted to a total head, using the formula

in USCS units $$\text{Total head, ft} = \frac{\text{net pressure, lb/in}^2 \times 2.31}{\text{sp. gr.}}$$

in SI units $$\text{Total head, m} = \frac{\text{net pressure, bar} \times 10.2}{\text{sp. gr.}}$$

Slope of the Head-Capacity Curve The slope of a boiler-feed pump head-capacity curve is important, because the intersection of pump curve and the system curve affects system control and stability. Greater pump curve slope yields greater angles of intersection with the system curve, which are less sensitive to dynamic pressure variants and thereby inherently more stable. Greater curve slope and intersection angles also maintain better load-sharing between multiple pumps operating in parallel on the same system, as one pump wears ahead of another.

All boiler-feed pumps should have sloping curve shapes with continuously rising head with decreasing flow, within their expected flow range of operation. Rise-to-shutoff (RTSO) is a simple parameter used to evaluate the relative slope of various pump head-capacity curves. From a pump design standpoint, RTSO refers to the ratio of the pump head under closed-valve condition over the pump head at the best-efficiency-point (BEP). From a system standpoint, RTSO applies to the ratio of the pump head under closed-valve condition over the rated pump head at full unit load. Therefore, it's important that the rated flow point by close to the BEP of pump and that the pump have sufficient RTSO for stable operation.

RTSO values vary from pump design to pump design, but are somewhat dependent on pump specific speed (Ns). Designs with Ns ranging from 1000 to 1400 are often used in smaller multistage pumps with constant-speed motor drivers, because these pumps benefit from designs with high heads per stage at relatively low flow. Unfortunately, these designs can have low RTSO, which can cause system instability with boiler-feed pumps operating back to the left on their curve. Pumps with constant-speed drivers operate further back on their curves than pumps with variable-speed drivers. Therefore, attention to RTSO values is particularly important in these circumstances.

Designs with Ns ranging from 1600 to 2000 are often used for large boiler-feed pumps in large steam power plants with nuclear reactors, because these pumps have conditions of service with high flows and relatively low heads. Unfortunately, these designs can have discontinuities in their head-capacities curves in which a head depression occurs at partial flow conditions (Figures 13 and 14). Pumps with this characteristic can be unstable at lower flows and should not be operated near the head depression.

While generous RTSO benefits system stability, it is important to recognize that excessive RTSO increases the design pressures of pump casings and system components from the pump discharge and to the regulator valve. For a continuously rising head-capacity curve, the pump shutoff condition achieves the highest pump discharge pressure and therefore defines vessel and piping working pressures. A typical specified range for RTSO for boiler-feed pumps is not less than 120% and not more than 135%.

Control of Boiler-Feed Pumps Boiler-feed pumps operating at constant speed are controlled by feedwater regulator valves, which throttle the difference between pump head output and boiler pressure requirements. As unit loads decrease, the amount of throttling

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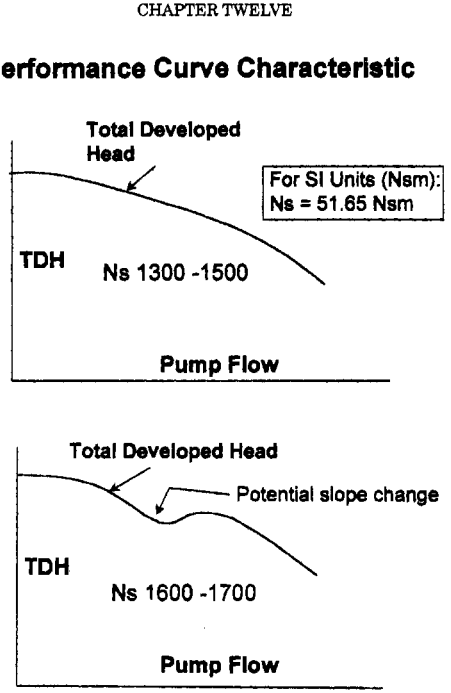


FIGURE 13 Pump performance curve characteristic—specific speed versus stability (Universal specific speed $\Omega_s = N_s/2733 \cdot N_q$ (in rpm, m³/s, m) = $N_s/51.65$)

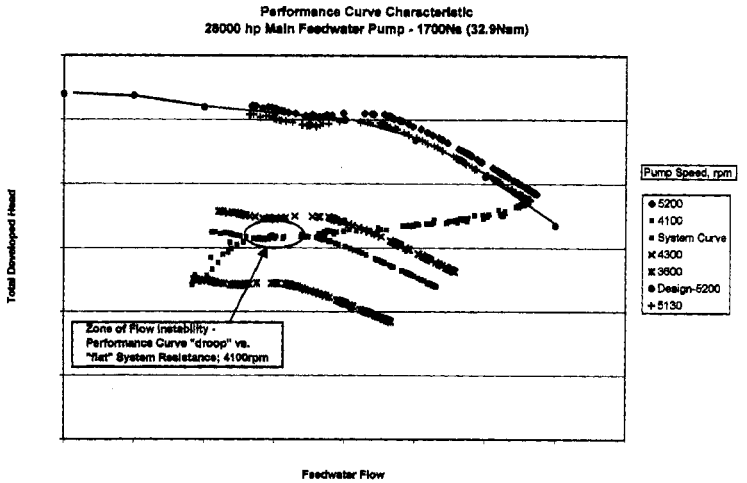


FIGURE 14 Performance characteristic—28,000 hp boilder feed pump (N_s – 1700) (Universal specific speed $\Omega_s = N_s/2733 \cdot N_q$ (in rpm, m³/s, m) = $N_s/51.65$)

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must increase because of the pump RTSO characteristic. Thus, throttle control of constant speed boiler-feed pumps wastes a portion of the energy of that pump. Also in large steam power plants, continuous throttling of large high-energy boiler-feed pumps can be damaging to the pumps and to the regulator valves. For these reasons, boiler-feed pumps with constant-speed driver are typically limited to 9000 BHP (6700 kW).

Large, high-energy pumps are best controlled by varying their speed. In some of these cases, feedwater regulator values are provided to throttle the pumps during low speed operation during plant starting. However, once the plant is at partial load, the feedwater delivery to the boiler is regulating by varying the pump driver speed. Typically, the boiler pressure curve for a large plant follows the affinity laws of the pump quite closely. Therefore, variable speed control keeps the pump running near its BEP, which maximizes machinery life and conserves energy.

Driver Power A single boiler-feed pump sized to deliver 100% capacity to a boiler or an HRGS generally will not operate at a capacity beyond the specified conditions of service. This is because conditions of service, which include safety margins, correspond to the largest operating capacity for the boiler or HRGS. In other words, a single 100% pump operates on a system head-capacity curve made up of the boiler drum pressure plus frictional losses in the piping and design margins. No further capacity is possible, unless the feedwater regulator valve goes full open, which is not a normal operating condition. Therefore, driver power ratings for single boiler-feed pumps may be selected with only 5% margin above the power at the rated conditions of service.

When two pumps are operated in parallel, feeding a single boiler, the situation is somewhat different. If one of the pumps is taken off the line at part load, the remaining pump could easily operate at capacities in excess of its design because its head-capacity curve would intersect the system-head curve at a head lower than the design head (Figure 15). In such a case, it is necessary to determine the pump capacity at the intersection point; the power corresponding to this capacity will be the maximum expected. It is not always necessary to select a driver that will not be overloaded at any point on the boiler-feed pump operating curve. Although electric motors used on boiler-feed service generally have an overload capacity of 15%, it is usually the practice to reserve this overload capacity as a safety margin and to select a motor that will not be overloaded at the design capacity. Exceptions occur in the case of very large motors. For instance, if the pump brake horsepower is 3100, it is logical to apply a 3000-hp motor, which will be overloaded by about 3% rather than a considerably more expensive 3500-hp motor. Because steam turbines are not built in definite standard sizes but can be designed for any intermediate rating, they are generally selected with about 5% excess power over the maximum expected pump power.

General Structural Features Multistage boiler-feed pumps are designed in three basic pressure casing types: Axial-split case, radial-split segmental ring, and radial-split barrel. Each casing type has its advantages and its limitations.

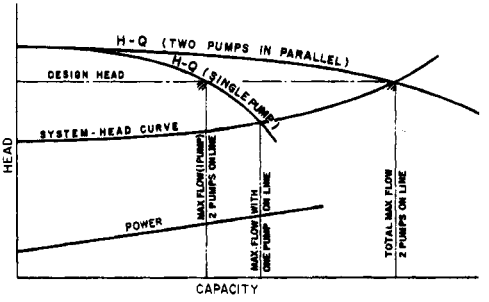


FIGURE 15 Method of determining maximum pump power for two boiler-feed pumps operating in parallel

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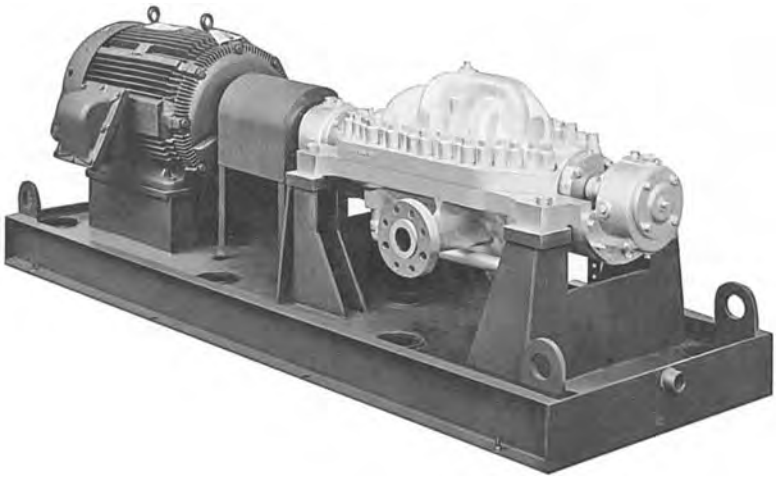


FIGURE 16 Axially split case multistage boiler feed pump, up to 3500 lb/in² (241 bar) (Flowserve Corporation)

Pumps with axial-split casings have two cast casing halves bolted together along a horizontal parting flange (Figure 16). Pumps with this casing type are easily inspected and serviced by removing the upper-half casing for access to the rotor without disturbing the piping. Pumps with axial-split cases require controlled prewarming and cooling for boiler-feed pump service, because of the complexity and wall thickness variations of the casings. Axial-split case designs are typically limited to design pressures of 2300 lb/in² (170 bar) in boiler-feed pumps.

Pumps with radial-split segmental ring casings have multiple stage casings clamped together between two head with tie-rods (Figure 17). Pumps with this casing type utilize minimal material with uniform thicknesses and are therefore capable of thermal transients hot or cold without warming provisions. They are troublesome to service, because any substantial work involve removal of the entire pump from the suction and discharge piping. Radial-split segmental ring casings nominally use O-rings at each joint for sealing. Therefore, boiler-feed pumps of this type are limited to approximately 350°F (177°C).

Pumps with radial-split barrel casings have double casings: an inner casing assembly made up of stage casings and an outer assembly made up of a barrel and a cover (Figure 18). The inner stage casing assembly contains the pumping elements (inner casings, diffusers, impellers, and so on). The outer barrel and cover assembly serves as the pump pressure vessel. Pumps with this casing type can be readily serviced without disturbing the piping, however once extracted, the inner assembly must be dismantled stage-by-stage for inspection and repair. Typically, the outer barrel and cover of a boiler-feed pump are steel forgings with uniform wall thicknesses, with confined gaskets at their radial joint. Therefore, radial-split barrel casings are capable of very high pressures and temperatures and are nominal used for high-energy boiler-feed pumps.

High-Speed, High-Pressure Boiler Feed Pumps Historically as fossil fuel-fired boiler pressures increased from 1800 lb/in² (120 bar) to 3000 lb/in² (200 bar) and eventually to 4500 lb/in² (310 bar), the total head required from main boiler-feed pumps increased from approximately 4000 ft (1220 m) to 7000 ft (2140 m) and 12,000 ft (3660 m). Initially during this transition, head increases were achieved by adding stages to pumps operating at two-pole motor speeds. Pumps became longer to accommodate the increased stages, which increased rotor deflections and lowered rotor critical speeds. These mechanical compromises interrupted the long life cycles of boiler-feed pumps, to which owners had became accustomed. The logical solution was to reduce the numbers of stages by increasing the head produced by each stage.

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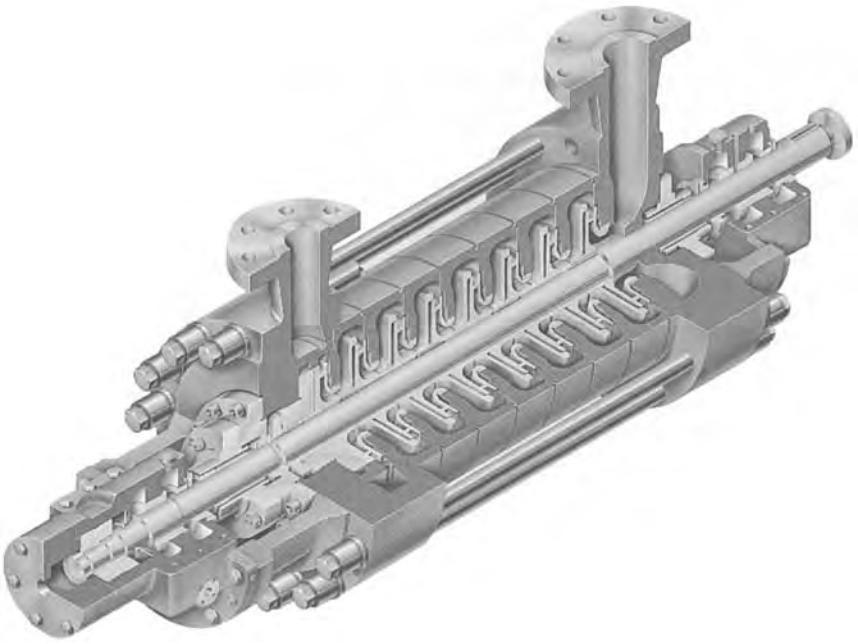


FIGURE 17 Radially split, segmental ring boiler feed pump (Flowserve Corporation)

In the 1970s, typical maximum head per stage values of main boiler-feed pumps increased from 800 ft (240 m) to 3000 ft (900 m). These increases were achieved by increasing rotating speeds from 3000 or 3600 rpm to 5000 to 6000 rpm, and even as high as 9000 rpm. As a result, high-speed high-energy pumps for boiler-feed service were established.

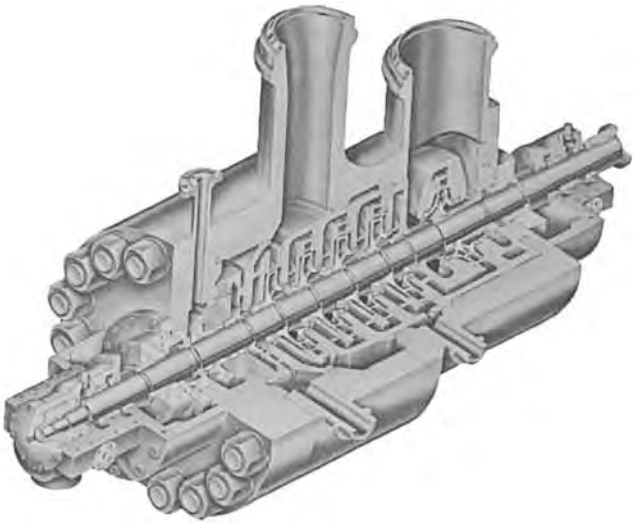


FIGURE 18 Radially split, double-case, barrel boiler feed pump (Flowserve Corporation)

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Subsequently during the 1980s, pump suppliers learned about the limitations of high-speed pumping particularly as it relates to high suction energy levels associate with high-speed pumps and the potential for cavitation erosion at first-stage impellers. Significant technological advances were made in new impeller vane designs and new impeller materials, which improve impeller life under these circumstances. Nevertheless, main boiler-feed pump installed in today's large fossil fuel-fired stream power plants typically have speeds between 5000 and 6000 rpm and heads produced per stage between 1800 ft (550 m) and 2400 ft (730 m). These design ranges with the latest pump technologies appear to provide the best balance of pump value, efficiency, and reliability.

Boiler-Feed Pump Drives The majority of boiler-feed pumps in small-size steam power plants, including most pumps in the steam cycles of combined-cycle power plants, are driven by electric motors at constant speeds. Furthermore, when the total head required in these plants is less than approximately than 5000 ft (1525 m), the pumps can reasonably operate at the motor speeds, either 3000 or 3600 rpm. Pumps required to produce greater total heads are driven at higher speeds using speed-increasing gearboxes with the electric motors. In some small plants where continuous process steam is critical (e.g., oil refinery-based steam power plants), multiple boiler-feed pumps are driven by both motors and steam turbines as protection against interruptions in electric power.

Boiler-feed pumps in large-size steam power plants are variable-speed driven with steam turbines, kinetic-type fluid couplings and electric motors, or with electric motors equipped with variable frequency drive controls. Variable-speed drives are typically used for boiler-feed pumps with power levels greater than 8000 hp (6000 kW) in order to match pump output with boiler demand and thereby achieve desired plant efficiencies and equipment life. Auxiliary steam turbines are by far the most common type of boiler-feed pump drives in these applications because:

1. Steam turbines provide variable-speed operation for matching pump output to boiler requirements as plant load varies.
2. Steam turbines provide ideal rotating speeds for multistage pumps with reasonable numbers of stages.
3. Steam turbines improve plant efficiency by utilizing exhaust steam from feedwater heaters.
4. Steam turbines reduce plant auxiliary electric power by eliminate large electric motors.
5. Steam turbines produce the high power levels needed for large high-energy boiler-feed pumps (Figure 19).

Operation of Boiler-Feed Pumps at Reduced Flows Operation of centrifugal pumps at shutoff or even at certain reduced flows can lead to very undesirable results. This subject is covered in detail in Sections 2.1.3 to 2.1.5, Sections 8.1 and 10, and Chap. 15, where methods for calculating minimum permissible flows and means for providing the necessary protection against operation below these flows are discussed.

Recent experiences have clearly defined the need to understand hydraulic instability, cavitation, and separation as they relate to off-design flow operation.

As deregulation and economic constraints dictate plant load cycling to match electricity demands and operating costs, the large central station boiler feed pumps experience significantly low operating flow. The low flow operation, high impeller suction specific speed, and high inlet tip speeds result in mismatched flow angles, backflow recirculation, and severe suction impeller inlet cavitation damage. This low flow hydraulic instability will also result in damage to pump volute cutwaters and diffuser vanes (Figure 20). Suction impellers where the suction specific speed exceeds 10,000 and eyebore inlet tip speeds exceed 200 ft/sec. are highly susceptible to this low flow instability and component damage. The series of photos in Figure 21 show a condition typical for many high-energy pumps operating at flows below design levels. They demonstrate how serious low-flow instability can be when it is coupled with two-phase flow activity.

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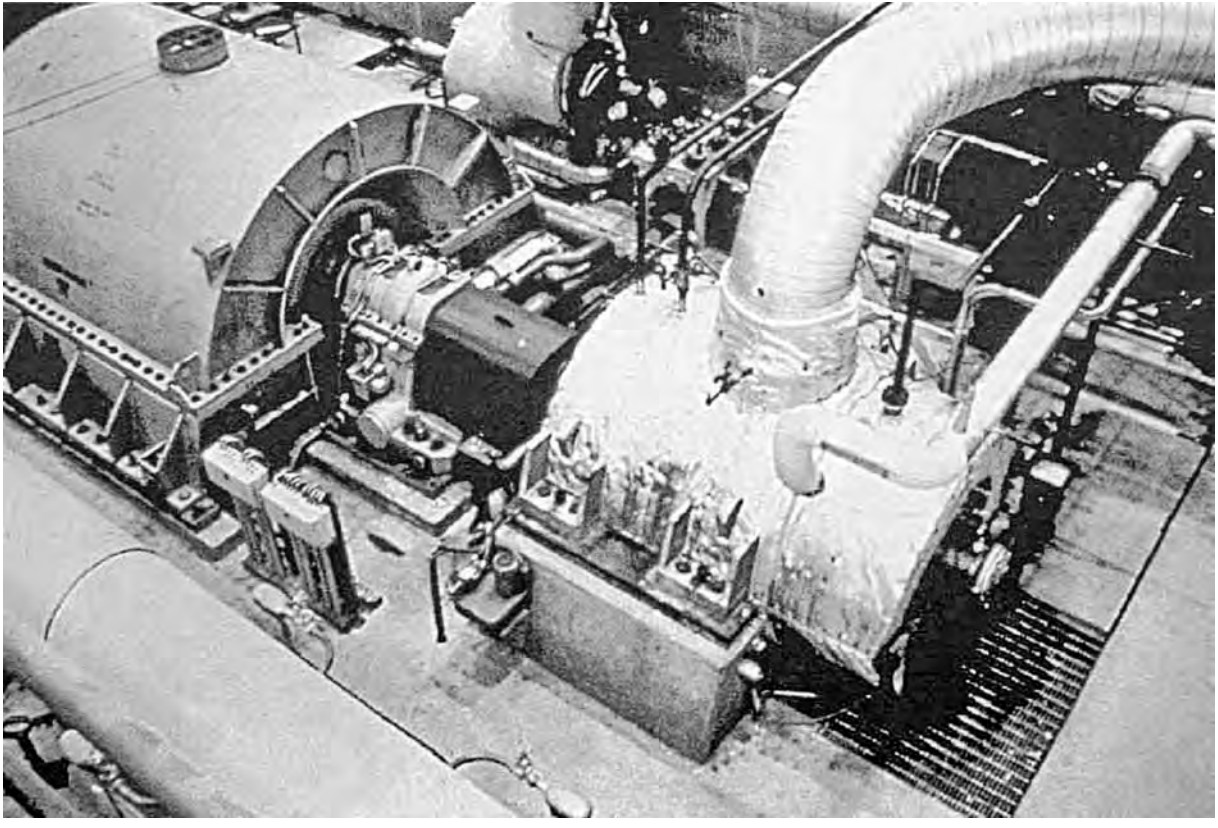


FIGURE 19 Installation of a single 65,000 horsepower (48,500 kW) boiler pump feed (Flowserve Corporation)

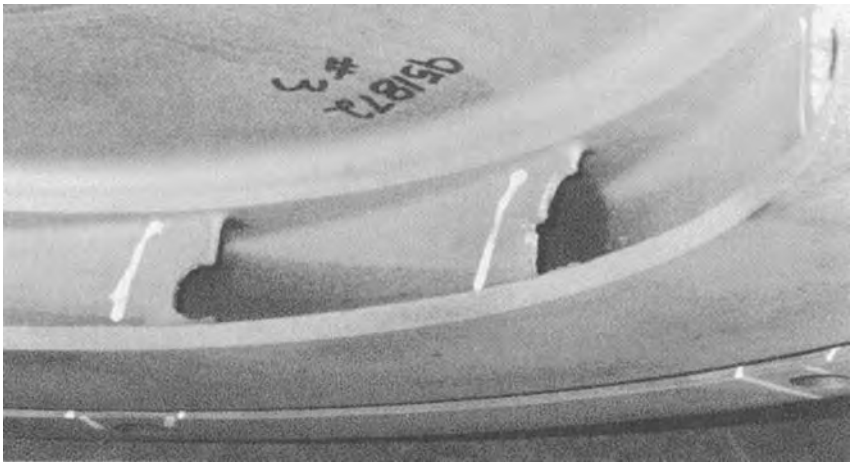


FIGURE 20 Diffuser inlet vane erosion damage (Flowserve Corporation)

The severity of cavitation erosion is highly dependent on the inlet tip speed of the suction stage impeller, the NPSHA, and the thermodynamic properties of the fluid being pumped. The erosion seen in Figure 22 was caused by the collapse of discrete cavitation vapor bubbles. Cavitation forms around an impeller blade because of local static pressure falling below the vapor pressure of the liquid being pumped. The vapor cavity shown in Figure 23 is an example of this phenomenon.

With decreasing flow rates (due to operating at off-design conditions), the fluid approaches the impeller blade with larger and larger angles of incidence altering the velocity and pressure fields inside the impeller.

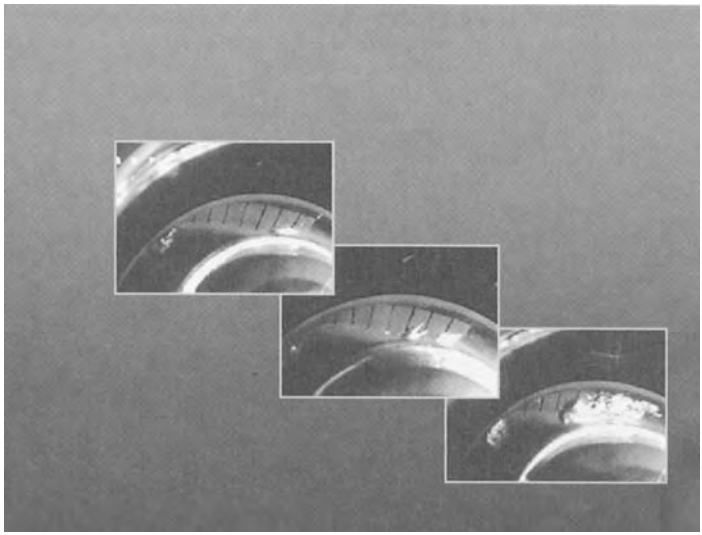


FIGURE 21 Unsteady vapor cavity behavior, feed pump at low flow (Flowserve Corporation)

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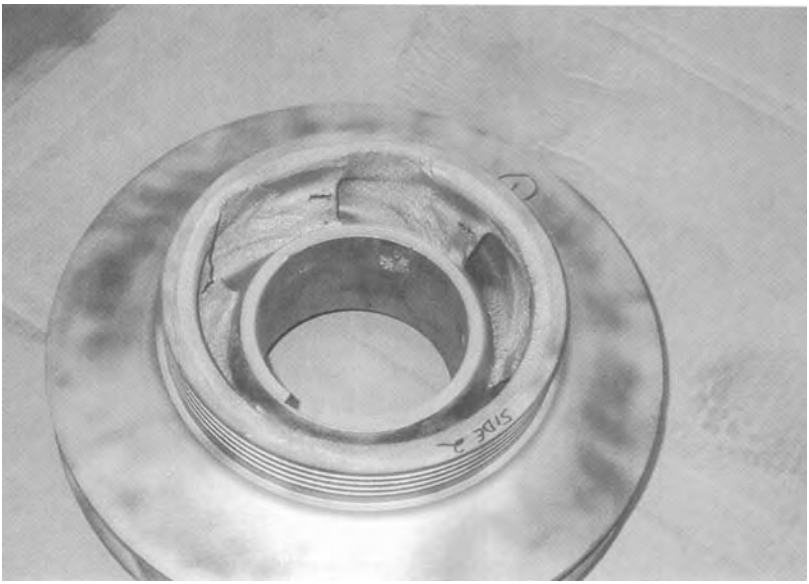


FIGURE 22 Impeller inlet vane cavitation erosion damage (Flowserve Corporation)

Pumps that are cycled between minimum-flow and flows in excess of the best efficiency point (BEP) create conditions at the impeller in excess of what "fixed geometry" machines can effectively tolerate. Impeller geometry has been shown to influence the degree and severity of cavitation problems experienced with high-energy pumps.

Through the 1980s, attempts were made to pursue "non-traditional" designs of impeller blading. These efforts took the form of profiling the inlet blade in a way that rapidly increased and then decreased the blade thickness.

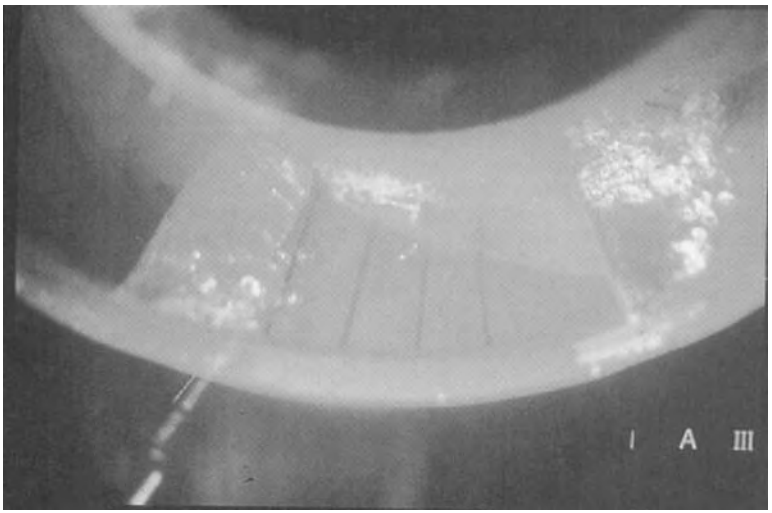


FIGURE 23 Cavitation vapor bubbles on suction surface of the impeller inlet vane (Flowserve Corporation)

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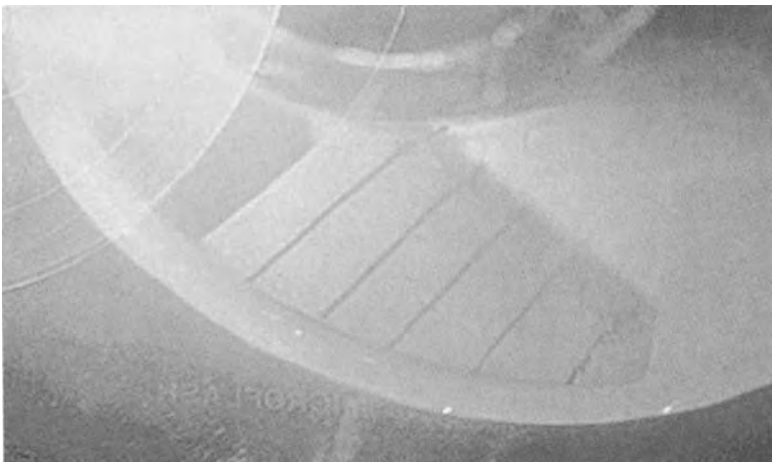


FIGURE 24 Operation at same flow as Figure 23 improved inlet vane shape—dramatic vapor bubble reduction (Flowserve Corporation)

A new impeller blade design approach, referred to as a "biased-wedge" design, has been found to provide a manufacturable configuration that enables cavitation bubble-free operation over a wide fluid flow range. This design approach is a result of extensive flow visualization test work and computational fluid flow analysis of many impeller geometries. It successfully advances the performance of high-energy pump suction stages to levels not achievable with conventional designs. Dramatic reduction in cavitation activity on the impeller was recorded as seen in the photo (Figure 24) of the suction surface of the final impeller taken at identical positions in the suction inlet and at the same operating conditions (baseload and minimum flow) as Figure 23. The inlet vane air foil shape has proven successful in facilitating feed pump flow rangeability.

Fundamentals for Successful Operating Life—Efficiency/Reliability Best practices for extended successful operating life of pumps are outlined in Chap. 15. Essential fundamentals to emphasize for boiler feed pumps are proper pump warm-up, standby warming, and shaft (fixed bushing) seal drain temperature control. These characteristics have become more critical as central station plants are cycled and large feed pumps are operated with varying loads and in standby modes. Current designs of multistage pumps (Figure 17) installed in combined cycle plants are less sensitive to thermal transients and wide swings in load (pump flow).

Pre-warming of the pump and maintaining warm-up flow to an idle pump to assure dimensional thermal uniformity is essential to maintenance of internal clearances, pump efficiency, and long life. This process is critical for multistage pumps to minimize thermal stratification within the pump. The distortion, including shaft bowing (Figure 25), will cause the following potential failure modes:

1. Flashing
2. Internal rubbing
3. Increased wear ring clearances
4. Pump seizure
5. Worn seal bushing clearance and excessive leakage
6. Loss of pump performance and efficiency
7. High pump vibration
8. Worn bearings/bearing clearances

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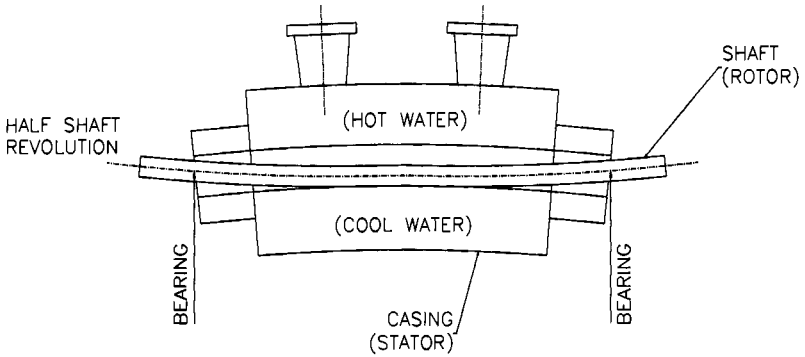


FIGURE 25 Thermal distortion of feed pump casing and shaft, due to improper warm-up and thermal stratification

Installation features and operating practices that extend pump life, efficiency, and reliability are

1. Proper pump insulation (Figure 26) at the casing and discharge head
2. Warm-up orifice, piped around the discharge check valve. Preference is to inject warm-up flow to the bottom of the pump casing to minimize short-circuiting of the hot feedwater and potential thermal stratification within the casing.
3. Maintaining shaft seal leakage drain temperature (Figure 27 between 150 and 170°F (65 and 77°C); utilize an electro-pneumatic temperature control system.

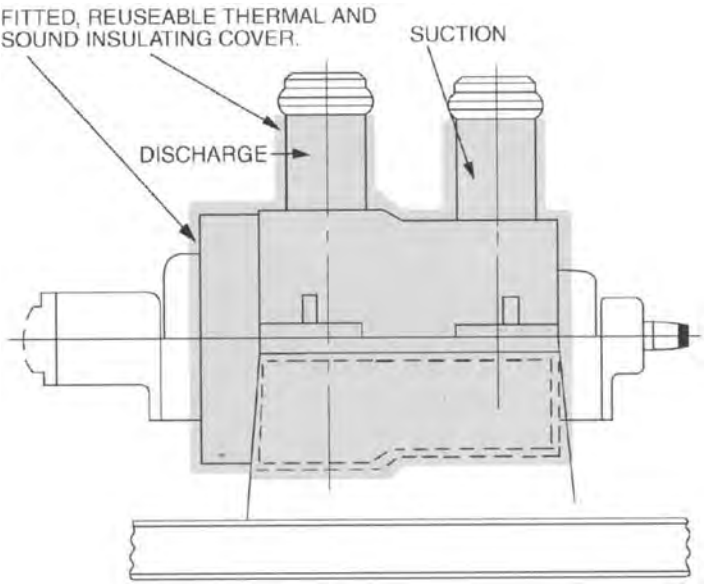


FIGURE 26 Recommended thermal insulation of a boiler feed pump (Flowserve Corporation)

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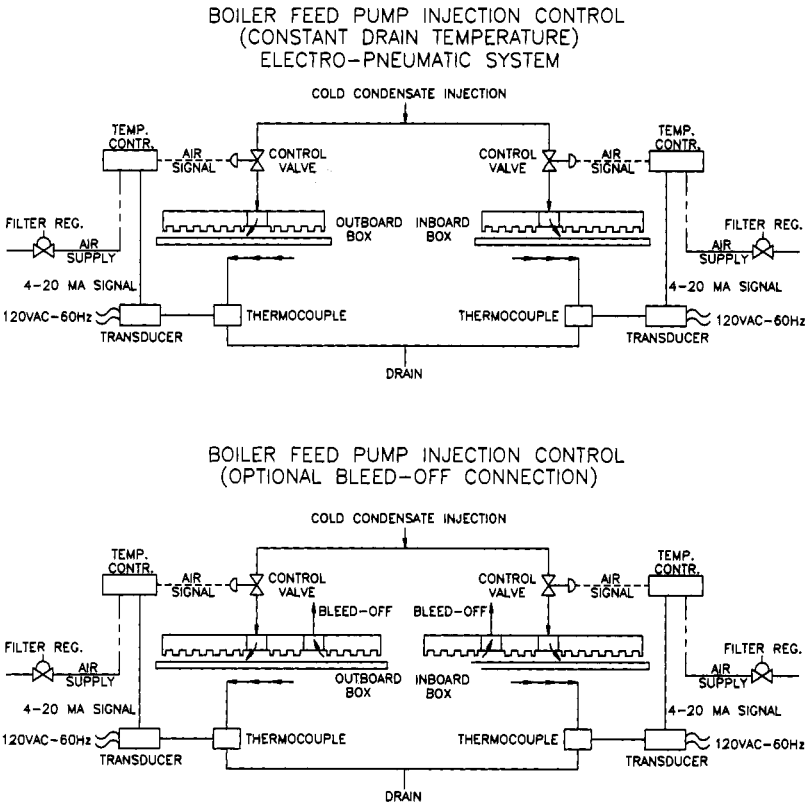


FIGURE 27 Boiler feed pump shaft seal injection/leakage control system; electro-pneumatic control of constant drain temperature

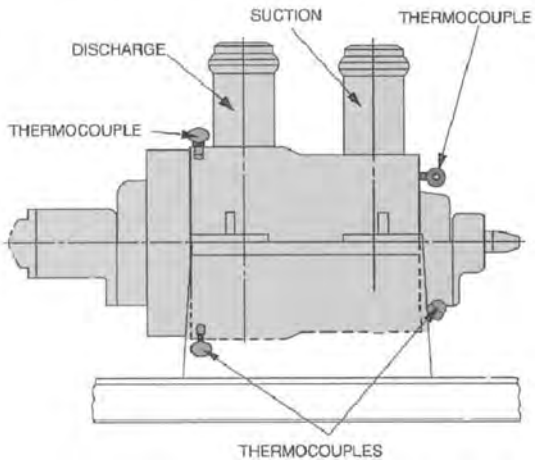


FIGURE 28 Feed pump casing and discharge head thermocouple installation, to monitor and control uniform temperature distribution (Flowserve Corporation)

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4. Installation of thermocouples or other temperature-detecting instruments (Figure 28) in the pump casing and discharge head to confirm temperature differences within 50°F (28°C) across the pump and relative to the feedwater temperature.
5. Assurance of proper functioning of the pump casing "pin" and "key" block to allow uniform thermal growth. Confirm that the hold-down bolts for the outboard casing feet are not over-torqued, preventing uniform axial thermal growth as the pump is heated.
6. Assurance of proper location and functioning of critical pipe hangers to minimize pipe strain on the pump suction and discharge nozzles.

Nuclear Power Plants In oversimplified form, the nuclear energy steam power plant differs from the conventional power plant only in that it uses a different fuel. Thus what is called the secondary cycle (consisting of turbogenerator, condenser and auxiliaries, and boiler-feed pumps) is not very different from its counterpart in the conventional steam power plant. The main differences are a desire for even greater equipment reliability and a preference for an absence or minimum of leakage to avoid any possibility of contamination with radioactive material. One other difference distinguishes most nuclear power plants today from their fossil fuel counterparts: their operating steam pressures and temperatures are much lower. Consequently, in most cases, reactor feed pumps are single-stage pumps; a typical section is shown in Figure 29. The lower operating conditions result in higher heat rates, and the flows—both of the feedwater and of the condenser circulation—are about one-third higher than for fossil fuel power plants of equal megawatt rating.

More detailed information on other nuclear power plant pumping services is given in Section 12.6.1.

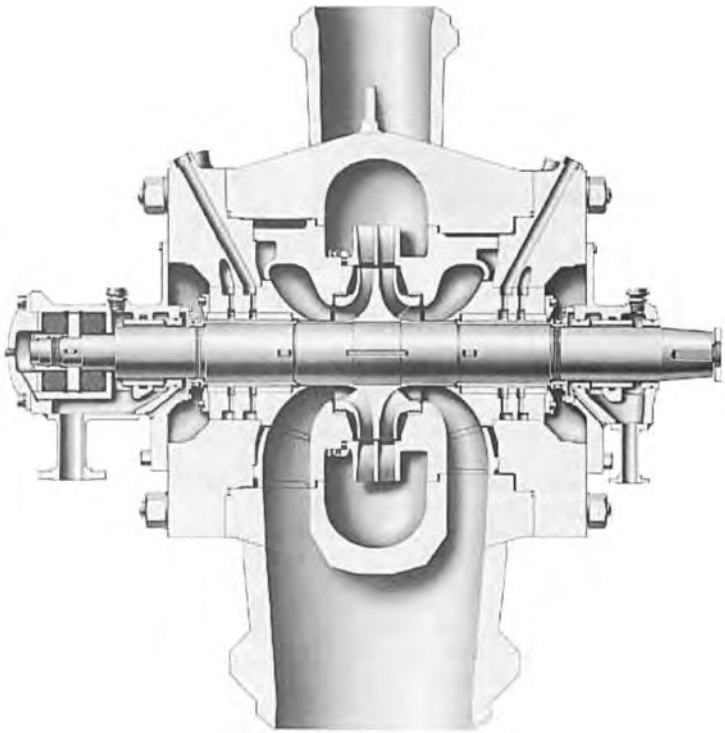


FIGURE 29 Single stage double suction reactor feed pump, 12,000 horsepower (8950 KW) (Flowserve Corporation)

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CONDENSATE PUMPS

Condensate pumps take their suction from the condenser hot well and discharge either to the deaerating heater in open feedwater systems (refer to Figure 5) or to the suction of the boiler-feed pumps in closed systems (refer to Figure 6). These pumps, therefore, operate with a very low pressure at their suction. The available NPSH is obtained by the submergence between the water level in the condenser hot well and the centerline of the condensate pump first-stage impeller. Because it is desirable to locate the condenser hot well as low as possible and avoid the use of a condensate pump pit, the available NPSH is generally extremely low, on the order of 2 to 4 ft (0.6 to 1.2 m). The exception to this occurs when vertical-can condensate pumps are used because these can be installed below ground and higher values of submergence can be obtained. Frictional losses on the suction side must be kept to an absolute minimum. The piping connection from the hot well to the pump should therefore be as direct as possible and of ample size and should have a minimum of fittings.

Because of the low available NPSH, condensate pumps operate at relatively low speeds, ranging from 1750 rpm in the low range of capacities to 880 rpm.

It is customary to provide a liberal excess capacity margin above the full-load steam condensing flow to take care of the heater drains that may be dumped into the condenser hot well if the heater drain pumps are taken out of service for any reason.

Types of Condensate Pumps Historically, horizontal multistage pumps with complex split-case casing configurations and double-suction-type impellers were used for condensate pump service (Figure 32). These were large low-speed pumps with limited head capability, because of the limited NSPH available of condensate extraction applications.

Modern steam power plants use vertical can-type multistage pumps (Figure 30). The length of these pumps provides adequate NPSH available at the first-stage impeller and their multistage configuration produce the required heads and flows required of any condensate pump service. Vertical can-type condensate pumps are available with either single-suction or double-suction impellers at the first stage position (Figure 31). For a given flow and speed, double-suction impellers need less NPSH required than suction-suction impellers. Therefore, double-section impellers reduce the length of the pump assembly and suction can, because the NPSH available at the first-stage position is strictly a function of that static elevation. However, high-flow condensate pumps benefit more from the advantages of double-suction impellers than low-flow pumps, because their NPSH required and costs of installation are greater.

Condensate pumps are located very close to the condenser hot well, and the suction piping is generally so short that the frictional losses in this piping are not significant. However, strainers are occasionally installed in this piping, and a great deal of attention must be paid to the frictional losses through them and to their frequent cleaning. Cases have been reported on occasion where the pressure drop across these strainers was sufficient to cause flashing at the suction nozzle of the condensate pumps.

Vertical can-type condensate pumps are driven by electric motors. Operating speeds are typically fixed at nominally 1500 or 1800 rpm. Larger pumps with flows greater than approximately 5000 gpm have motors with nominal speeds of 1000 or 1200 rpm.

To prevent air leakage at the stuffing boxes, condensate pumps equipped with packing are always provided with seal cages. The water used for gland sealing must be taken from the condensate pump discharge manifold beyond all the check valves. Pumps fitted with mechanical seals also use injection of condensate at a pressure higher than atmospheric pressure to prevent air from being drawn into the pump and feedwater system across the seal faces.

Condensate pumps are typically supplied with cast iron casings or bowls. Impellers are typically stainless steel or chrome steel materials. Carbon or other nonmetallic bearings are now used in place of bronze materials, because of potential steam turbine blade sensitivity to any copper in the flow stream.

Condensate Pump Regulation When condensate pumps operate in a closed cycle ahead of boiler-feed pumps, the two sets of pumps in series combine insofar of the system curve is concerned. With this arrangement, variation in flow through both sets of pumps

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FIGURE 30 Vertical condensate or heater drain pump (Flowserve Corporation)



FIGURE 31 Vertical turbine condensate pump (Flowserve Corporation)

is regulated by throttling the feedwater regulating valve or by varying the speed of the boiler-feed pump.

In an open cycle with a deaerator heater, the condensate pumps must be regulated separately from the boiler-feed pumps. This can be accomplished several methods:

1. Limit the NPSH available to the condensate pump such that it breaks at higher flows (Figure 32).
2. Throttle the discharge of the condensate pumps using deaerator level control.
3. Bypass excess flow from the condensate back to the hot well.
4. Varying the speed of the condensate pumps.

Method 1 requires the condensate pumps operate with significant amounts cavitation at all flow rates. Historically, specialized pumps worked without severe damage under

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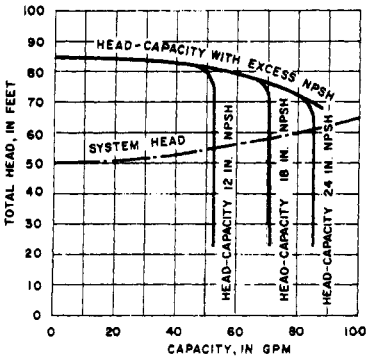


FIGURE 32 Characteristic of a condensate pump operating on a submergence-controlled system

these circumstances, because their suction energy and the damage potential of saturated water were not substantial. However, today this method is rarely used, because it is not appropriate for vertical can-type pumps.

The condensate pump discharge can be throttled by a float control arranged to position a valve that increases the system-head curve as the level in the hot well is drawn down. This eliminates the cavitation in the condensate pump, but at the cost of a slight power increase. Furthermore, the float necessarily operates over a narrow range, and the mechanism tends to be somewhat sluggish in following rapid load changes, often resulting in capacity and pressure surges. This transient condition is often the root cause of failed thrust bearings and axial rotor shifting. Another critical piping arrangement feature is the discharge piping check valve. The check valve in every condensate pump must be below the condenser hotwell level to ensure prevention of air entrapment and start-up waterhammer. [Refer to Section 11.3.]

When condensate delivery is controlled through bypassing, the hot well float controls a valve in a bypass line connecting the pump discharge back to the hot well. At maximum condensate flow, the float is at its upper limit with the bypass closed and all the condensate is delivered to the system. As the condensate flow to the hot well decreases, the hot well level falls, carrying the float down and opening the bypass. Sluggish float action can create the same problems of system instability in bypass control as in throttling control, however, and the power consumption is excessive because the pump always operates at full capacity.

A combination of throttling and bypassing control eliminates the shortcoming of excessive power consumption. The minimum flow at which bypassing begins is selected to provide sufficient flow through the ejector condenser.

A modification of the bypassing control for minimum flow is illustrated in Figure 33, which shows a thermostatic control for condensate recirculation. With practically constant steam flow through the ejector, the rise in condensate temperature between the inlet and outlet of the ejector condenser is a close indication of condensate flow rate through the ejector condenser tubes. Therefore an automatic device to regulate the condensate flow rate can be controlled by this temperature differential. A small pipe is connected from the condensate outlet on the ejector condenser back into the main condenser shell. An automatic valve is installed in this line and is actuated and controlled by the temperature rise of the condensate. Whenever the temperature rises to a certain predetermined figure, indicating a low flow of condensate, the automatic valve begins to open, allowing some of the condensate to return to the condenser and then to the condensate pump, which supplies it to the ejector at the increased rate. When the temperature rise through the ejector condenser is less than the limiting amount, indicating that ample condensate is flowing through the ejector condenser, the automatic valve remains closed.

As condensate flow design demand increased, the vertical multistage pumps were installed as two half-capacity pumps. Flow variation is accomplished by operating one or two pumps and by utilizing the regulating discharge valve.

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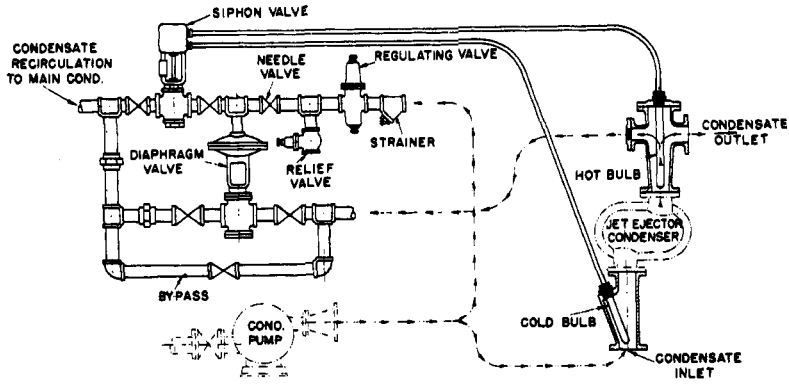


FIGURE 33 Thermostatic control for condensate recirculation

Today, most modern steam power plants use some combination of flow throttling and flow bypassing (methods 2 and 3 above) to control condensate pumps. When two or more pumps in parallel service are controlled in this manner with common control elements, emphasis must be placed on preventing zero-flow and low-flow operation of any pumps sharing the controls. Under these circumstances, a weaker pump can be "dead-headed" by a stronger pump and severe pump damage can be expected. In these cases, individual pumps should be monitored with instrumentation to confirm their operation in the allowable operating range.

Just as with boiler-feed pumps—or any high-energy pump for that matter—large high-energy condensate pumps should not be operated at reduced flows for prolonged periods of time. Such operation wastes significant pumping energy and risks pump damage. In these cases, variable speed control is a viable alternative. Application of variable speed control also reduces the magnitude of flow bypassed back to the hot well during reduced unit load operations.

HEATER DRAIN PUMPS

Service Conditions Condensate drains from closed heaters can be flashed to the steam space of a lower-pressure heater or pumped into the feedwater cycle at some higher-pressure point. Piping each heater drain to the heater having the next lower pressure is the simpler mechanical arrangement and requires no power-driven equipment. This "cascading" is accomplished by an appropriate trap in each heater drain. A series of heaters can thus be drained by cascading from heater to heater in the order of descending pressure, the lowest being drained directly to the condenser.

This arrangement, however, introduces a loss of heat because the heat content of the drains from the lowest-pressure heater is dissipated in the condenser by transfer to the circulating water. It is generally the practice, therefore, to cascade only down to the lowest-pressure heater and pump the drains from that heater back into the feedwater cycle, as shown in Figure 34. Because the pressure in that heater hot well is low (frequently below atmospheric even at full load), heater drain pumps on that service are commonly described as on "low-pressure heater drain service."

In an open cycle, drains from heaters located beyond the deaerator are cascaded to the deaerator. Although the deaerator is generally located above the closed heaters, the difference in pressure is sufficient to overcome both the static and the frictional losses. This difference in pressure decreases with a reduction in load, however, and at some partial main turbine load it becomes insufficient to evacuate the heater drains. They must be switched to a lower-pressure heater or even to the condenser, with a subsequent loss of

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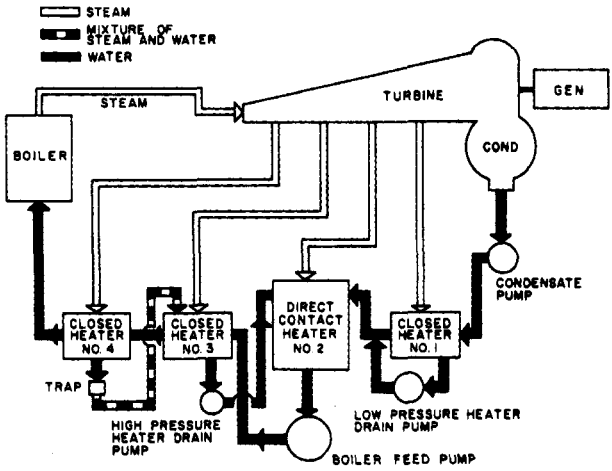


FIGURE 34 Typical arrangement for heater drain pumps

heat. To avoid these complications, a "high-pressure heater drain pump" is generally used to transfer these drains to the deaerator. Actually, this pump has a "reverse" system head to work against; at full load, the required total head may be negative, whereas at light loads, the required head is at its maximum.

Heater drain pumps are subjected to more severe conditions than condensate pumps:

1. Suction pressures are higher and can fluctuate from vacuum to positive.
2. Suction temperatures are higher.
3. Suction transients are more frequent, because of smaller capacitance in the pump suction.
4. Discharge transients are more frequent, because of regulating valve controls to prevent excessive pump runout.

Types of Heater Drain Pumps Heater drain pumps should be capable of continuous operation during sudden load fluctuations. They should be designed to run-dry momentary and once more pump forward as suction pressure is reestablished. They should be designed for wide ranges of flow from minimum to runout.

In the past, heater drain pumps were horizontal pumps with either single- or multistage construction depending on head requirements of a particular system. Horizontal pumps offered advantages such as accessibility and ease of maintenance. However, horizontal pumps with multistage construction were often unreliable, because of rotor contact problems during suction transient conditions.

For low-head services, single-stage horizontal pumps are appropriate. These pumps should be robust designs such as API process pumps (Figure 39). For high-head services, specialized vertical can-type pumps are most appropriate (Figure 30). As previous described for condensate pumps, vertical can-type pumps offer the advantage of increased NPSH available to the first-stage impeller below the floor elevation, which reduces the impact of transients. Also, the rotors of vertical pumps are not subject to static deflections which cause automatic contact at the impeller wear rings. Nevertheless, vertical can-type heater drain pumps should be fitted with carbon bearings and hard-surfaced journals to improve their dry-run capability. In addition, the bearing in the suction bell is often pressurized with water from the first-stage casing to ensure continuous lubrication during suction transients.

Like condensate pumps, all heater drain pumps should be continuously vented back to the steam space in the feedwater heater. Also like condensate pumps, shaft seals should

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be fitted to prevent air ingress during vacuum conditions. However, materials of construction for heater drain pumps should be like those of boiler-feed pumps: steel casings with stainless or chrome steel impellers.

CONDENSER CIRCULATING PUMPS

Service Conditions Condenser circulating pumps pump cooling water through the condenser (refer to Figure 1). The source of the cooling water can be a natural body of water, such as the sea, a river, or lake, or a cooling tower. The arrangement with natural source is called an open cooling water system. The arrangement with the cooling tower is called a closed cooling water system, because the same water is cycled through the system. Closed cooling water systems can use forced draft or natural draft cooling towers with makeup water from some natural source. Alternately, some closed cooling water systems with direct contact-type condensers and an air fin-type cooling towers use cool condensate as the cooling water.

Condenser circulating pump services are mostly the same for open and closed cooling water systems. Generally, closed cooling systems require more head from circulating pumps because of static height of cooling tower. Pump materials of construction must take into account increased concentrations of corrosive agents as water evaporates in closed cooling water systems. In open cooling water systems, concentrations of suspended solids from the water source must be considered.

Types of Pumps Condenser circulating pumps may be of either horizontal or vertical construction. For many years, the low-speed, horizontal, double-suction volute centifugal pump (Figure 35) was the preferred type. This pump has a simple but rugged design that allows ready access to the interior for examination and rapid dismantling if repairs are required.

Today, condenser circulating pumps in most new steam power plants are wet-pit vertical pumps with flows ranging from 60,000 to 300,000 gpm. Some large plants use concrete volute-type pumps with flows up to 500,000 gpm.

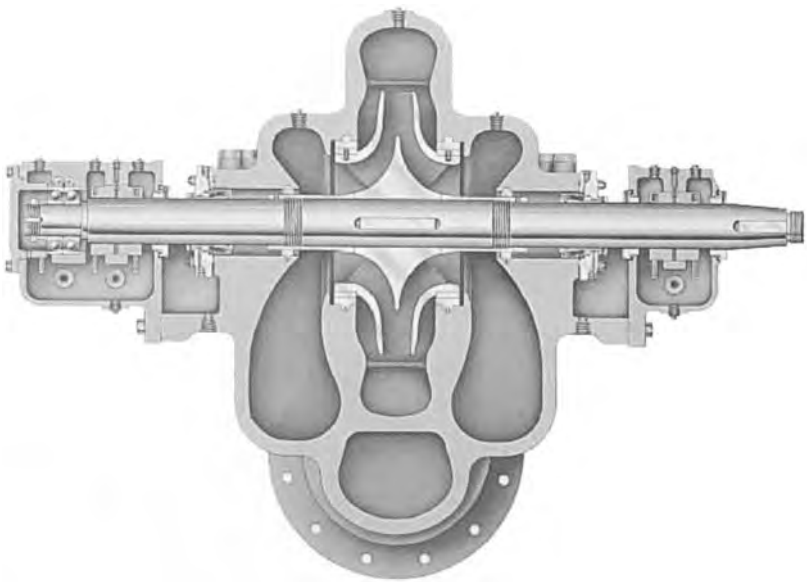


FIGURE 35 Horizontal double suction single stage pump, IDP model LN (Flowserve Corporation)

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Mechanical Considerations The dry-pit installation was a single-suction, medium-specific-speed, mixed-flow pump (Section 2.2.1, Figure 79). This design combined the high efficiency and low maintenance of the horizontal double-suction radial-flow centrifugal pump with lower cost and slightly higher rotative speeds.

Because of their suction and discharge nozzle arrangements, these pumps are ideally suited for vertical mounting in a dry pit, preferably at the lowest water level, so they are self-priming on starting. They are directly connected to solid-shaft induction or synchronous motors, either close-coupled or with intermediate shafting between the pump and the motor, which is then mounted well above the pump pit floor.

Like the horizontal double-suction pump, the vertical dry-pit mixed-flow pump is a compact and sturdy piece of equipment. Its rotor is supported by external oil-lubricated bearings of optimum design. This construction requires the least attention, for the oil level can be easily inspected by means of an oil sight glass mounted at the side of the bearing or oil reservoir.

Because the rotor is readily removed through the top of the casing, facilitating maintenance and replacement, the pump does not have to be removed from its mounting and the suction and discharge connections do not have to be broken to make periodic inspections or repairs.

In recent years, power plant designers have shown a preference for the wet-pit column-type condensate circulating pump. The term *wet-pit* normally implies a casing diffuser-type pump, employing a single open vane impeller. The wet-pit pump (Figure 36) employs a long column pipe that supports the submerged pumping element. It is available with open main shaft bearings lubricated by the water handled, or with enclosed shafting and bearings, lubricated by clean, fresh, filtered water from an external source. There is some danger of contamination of the lubricating water from seepage into the shaft enclosure tube during shut-downs.

Pulling up the column in a long pump requires special facilities and, in addition, the discharge flange must be disconnected when withdrawing the pump and column from the pit. This design has been designated a "non-pull-out" design (Figure 36). To avoid the necessity of lifting the entire pump when the internal parts require maintenance, some units are built so the impeller, impeller shroud, casing, and shaft assembly can be removed from the top without disturbing the column pipe assembly. (The driving motor must be removed.) These designs are commonly designated "pull-out" designs (Figure 37).

Condenser cooling water is often corrosive. Power plants are often located near salt or brackish bodies of water. Plants near rivers often encounter water contaminated with high silt levels. With such waters, selection of materials can be critical to long service life. Material selection for sea water applications must also consider the potential for electrolytic (galvanic) corrosion.

Performance Characteristics Condenser circulating pumps are normally required to work against low or moderate heads. Extreme care should be exercised in calculating the system frictional losses, which include losses from friction in the condenser. If more total head is specified than is required, the resulting driver size may be unnecessarily increased. For instance, an excess of 1 or 2 ft (0.3 or 0.6 m) in an installation requiring only 20 ft (6 m) of head represents an increase of 5 to 10% in excess power costs.

The range of suction lift for dry-pit pumps must be determined very accurately and checked with the manufacturer to ensure that cavitation will be avoided in the installation. Priming facilities must be provided, or the pump must be installed in a dry pit at such an elevation that the water in the suction channel leading to the pump will be maintained at the level recommended by the manufacturer. This presents no problem in a wet-pit installation because the pump column can be made long enough to provide adequate submergence, even with minimum water levels in the suction well or pit. The dry-pit pump will generally have 3 to 4% higher efficiency than the wet-pit type and therefore 3 to 4% lower power consumption. The two types are available for the same specific speed range. When pumping total head is 25 ft (7.6 m) or less, an axial-flow propeller (approximately 10,000 specific speed in USCS units) can be used in either type of pump.

The low-specific-speed, double-suction pump has a very moderate rise in head with reducing capacities and a nonoverloading power curve with a reduction in head. The mixed-flow impeller with a higher specific speed has a steeper head-capacity curve and a

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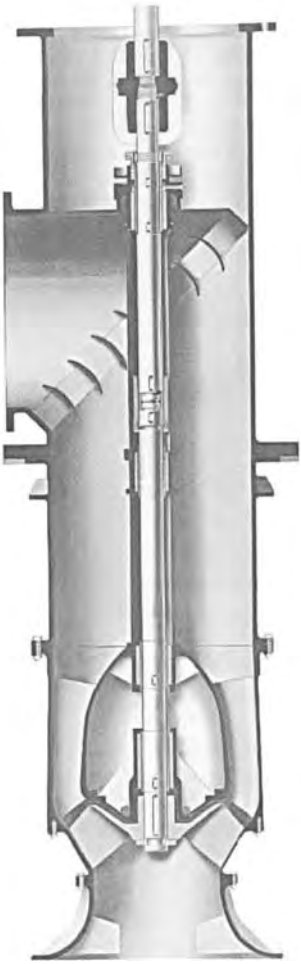


FIGURE 36 Vertical wet pit circulating water pump (non-pull-out) (Flowserve Corporation)



FIGURE 37 Vertical wet pit circulating water pump (pull-out) (Flowserve Corporation)

reasonably flat power curve that is also nonoverloading. As the specific speed increases, the steepness of the head-capacity curve increases and the curvature of the power curve reverses itself, hitting a maximum at the lowest flow. Finally, the curve of a high-specific-speed propeller pump has the highest rise in both head-capacity and power-capacity curves toward zero flow. The head range developed by the mixed-flow pump is ideal for condenser service; this pump is usually furnished with an enclosed impeller, which produces a relatively flat head-capacity curve and a flat power characteristic.

Higher head circulating water pumps were developed in the 1970s as cooling towers were introduced to improve plant efficiency and environmental contamination. The cooling tower arrangement effectively increased the total system resistance head requirements.

System Hydraulics The dry-pit pump is not too sensitive to the suction well design because the inlet piping and the formed design of the suction passages into the pump

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normally ensure a uniform flow into the eye of the impeller. On the other hand, the higher-speed wet-pit pumps are more sensitive to departures from ideal inlet conditions than the low-speed centrifugal volute pump or the medium-speed mixed-flow pump. A discussion of the arrangements recommended for wet- and dry-pit pump installations is presented in Section 13.1.

Condenser circulating pumps are typically provided in sets of two, three, or four identical pumps operating in parallel on the same cooling water system. Generally, discharge throttle valves are not provided. Flow is simply controlled by reducing the number of pump in service. Therefore, multiple condenser circulating pumps operate with flows from rated to runout conditions. Careful consideration must be given to pump runout capability, including NPSH required at maximum flow conditions.

Large concrete volute pumps are often configured as a single 100% pump in a condenser circulating cooling water system. If system flow reductions are needed, variable speed control of the pumps is best for single pump configurations.

Drivers Dry-pit horizontal pumps are typically coupled with flexible-element couplings to horizontal electric motors. These pumps operate at fixed speeds.

Wet-pit vertical pumps for condenser circulating service are usually coupled with rigid couplings to vertical electric motors. In this case, the motor thrust bearing is designed for the pump hydraulic thrust loads plus the rotor static weights. Alternative designs are available with thrust bearings mounted in the pumps and flexible element coupling. However, these designs are more difficult to service and increase the height of the pump-motor assembly.

Concrete volute-type pumps are typically driven by electric motors with speed-decreasing gearboxes, because these large pumps have low rotating speeds. The pumps have oil or grease lubrication bearings in the pump assembly and therefore flexible-element couplings are used.

BOILER CIRCULATING PUMPS

The forced circulation, or controlled circulation, boiler requires the use of circulating pumps that take their suction from a header connected to several downcomers, which originate from the bottom of the boiler drum and discharge through the various tube circuits operating in parallel (Figure 38). The circulating pumps therefore must develop a pressure

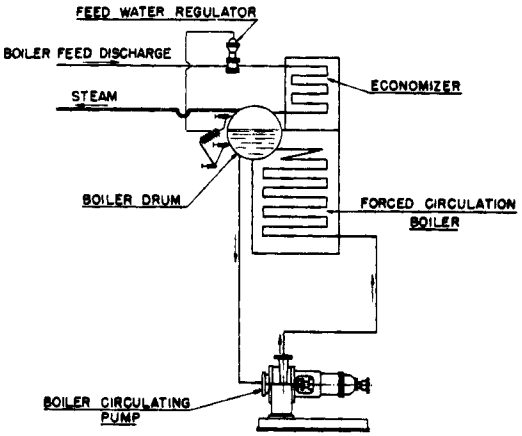


FIGURE 38 Forced, or controlled, circulation system

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equivalent to the frictional losses through these tube circuits. Thus, in the case of different boilers operating in pressure ranges from 1800 to 3000 lb/in² (124 to 207 bar), the boiler circulating pump must handle feedwater from 620 to 690°F (326 to 365°C) under a suction pressure of 1800 to 2900 lb/in² (124 to 200 bar). Such a combination of high suction pressure and high water temperature at saturation imposes very severe conditions on the circulating pump stuffing boxes, making it necessary to develop special designs for this part of the pump.

The net pressure to be developed by these pumps is relatively low, ranging from 50 to 150 lb/in² (3.4 to 10.3 bar). Hence these are single-stage pumps with single-suction impellers and a single stuffing box. The high boiler pressure imposes an extremely severe axial thrust on the pump, placing a load of several tons on the thrust bearing. In many cases, a thrust-relieving device must be incorporated to permit pump start-up. Two general types of construction are used for this service: (1) the conventional centrifugal pump with various stuffing box modifications (Figure 39) and (2) the submersible motor pump of either the wet- or dry-stator type (Figure 40). In the lower boiler pressure range-up to 500 or 600 lb/in² gauge (34 or 41 bar), the construction shown in Figure 41 may be used. The pump is of the same general type as is used on high-pressure heater drain service. The packed stuffing box or mechanical seal is preceded by a pressure-reducing bushing. Feedwater from the boiler-feed pump discharge, at a temperature lower than in the boiler drum and at a pressure somewhat higher than pump internal pressure, is injected into the middle of this bushing. Part of this injected feedwater proceeds toward the pump interior, making a barrier against the outflow of high-temperature water. The rest proceeds outward to a bleed portion of the bushing, from where it is bled to a lower pressure, often the deaerator. The packing or mechanical seal needs to withstand only the lower boiler-feed pump temperature and a much lower pressure than boiler pressure.

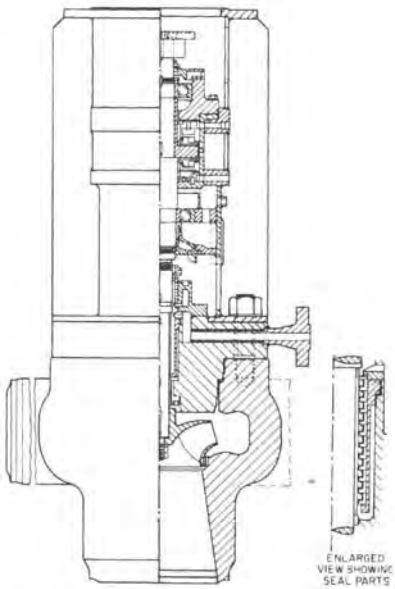


FIGURE 39 Vertical injection-type boiler circulating pump for high pressures (Flowserve Corporation)



FIGURE 40 Boiler circulating pumps driven by a wet stator motor (Hayward Tyler)

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FIGURE 41 End-suction boiler circulating pump for low-pressure range (Flowserve Corporation)

More sophisticated designs are required for pressures from 1800 to 3000 lb/in^2 gauge (124 to 207 bar) (Figure 39). The shaft is sealed by two floating ring pressure breakdowns and a water-jacketed stuffing box. Boiler feedwater is injected at a point between the lower and upper stacks of floating ring seals at a pressure about 50 lb/in^2 (3.5 bar) above the pump internal pressure. Here again, part of this injection leaks into the pump interior and the rest leaks past the upper stack of seals to a region of low pressure in the feed cycle. Leakage to atmosphere is controlled by the conventional stuffing box located above the upper stack. The seal injection and leakoff control system is very sensitive to boiler and feedwater pump transients. Loss of injection results in flashing in the sealing chamber and failure of the sealing rings. Current technology utilizes a two-stage, high-pressure mechanical seal. This eliminates the need for a separate seal injection system and seal-injection booster pump.

The available NPSH may not be sufficient at start-up, when the water in the boiler is cold and the pressure is low. Therefore certain installations include two-speed motors so a lower NPSH is required at start-up. There is an added advantage to this arrangement: under normal operating conditions, the feedwater will heat boiler saturation temperature and therefore will have a specific gravity of as low as 0.60; at start-up, however, the specific gravity will be 1.0. The power consumption on cold water would therefore be some 65% higher than in normal operation if the pump operates at the same speed, and a much larger motor would be required. If a two-speed motor is used, however, the pump is operated at lower speed when the water is cold, and the motor need be only large enough to supply the maximum power required under normal operating conditions.

ASH-HANDLING PUMPS

Boilers that use solid fossil fuels produce refuse that is generally classified as ash and includes bottom ash or slag, fly ash, and mill rejects such as pyrites and so on. This refuse must be removed and disposed of, either by hydraulic or pneumatic conveying. The latter, of course, does not involve the use of pumps and need not be discussed here.

Fly ash and bottom ash is removed from the boiler system through a series of water ejectors. Centrifugal pumps provide the water, taken from either the circulating water pump header or ash-settling pond. These pumps (Figure 42) are subjected to severe duty as pump flows vary significantly and the pumpage is often contaminated with suspended silt or fly ash.

Hydraulic conveying is generally restricted to bottom ash and mill rejects. There are a great number of different systems in use today, and the reader should consult boiler design and operation literature to become acquainted with this subject. What is common to most hydraulic conveying systems is that they use centrifugal pumps to handle concentrated

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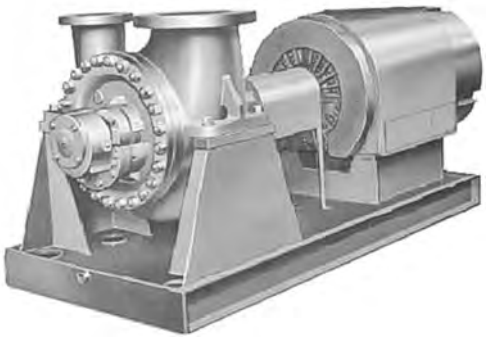


FIGURE 42 Horizontal single stage double suction, high head pump (Flowserve Corporation)

ash slurries that may contain considerable amounts of coarse, heavy pieces of stone, slate, and iron pyrites. Pumps suitable for this service are described in detail in Section 4.2.

SLURRY PUMPS

Modern coal-fired steam power plants use wet limestone scrubber systems to reduce in high rates the SO_2 emissions from flue gases. These systems utilize a variety of conventional and slurry pumps to prepare and feed limestone slurries and to dewater and transport by-product slurries including:

- Mill slurry pumps
- Limestone slurry feed pumps
- Filter feed pumps
- Filtrate return pumps
- Gypsum slurry pumps

The pumps in these services are single-stage, end-suction centrifugal designs. The slurry pumps have hard-metal or rubber-lined construction. For additional information about slurry pumps, refer to Section 4.2.

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FURTHER READING

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SECTION 12.6
NUCLEAR

12.6.1
NUCLEAR ELECTRIC GENERATION

W. M. WEPFER
W. W. PARRY, JR.

APPLICATION

The primary difference between the pumps for nuclear service and conventional steam generation is the stringent requirements for reliability and safety. The continuous functioning of nuclear pumps is critical to the safe operation of the plant. In addition, many of these units maintain the responsibility of ensuring safe shutdown of the facility during emergency conditions. Because the release of radioactive fluids could result in potential personnel and environmental hazards along with associated costly cleanup, nuclear pumps must be self-contained. The rules and regulatory standards for nuclear pumps are dedicated to the promotion of public safety through reliable operation. Pump manufacturers must be fully knowledgeable of these standards and certified by the American Society of Mechanical Engineers (ASME) (or an equivalent authority outside the United States) before they can offer pumps for the nuclear market.

All nuclear power plants built for public utility service involve the generation of steam. The method of steam generation differentiates the two major types of plants. The pressurized water reactor (PWR) employs two distinct systems, a primary and a secondary loop. The primary loop circulates pressurized radioactive fluid from the reactor to the steam generator where it heats the secondary fluid. The secondary loop directs steam from the steam generator to the turbines and condensed fluid back to the generators. A boiling water reactor (BWR), on the other hand, only employs one loop. The steam is generated in the reactor and then circulated through the turbines, condensers, and heaters and back to the reactor.

It is usual for the plant to contain one major system, or loop, and a number of supporting systems. Because virtually every system requires some form of pumping, it is conventional to designate each pump by the name of the system with which it is associated. For each of the systems previously discussed, the main and auxiliary systems are reviewed, including brief descriptions of the pumping requirements.

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PWR Plants PWR plants employ two separate main systems to generate steam. In the primary system, the water is circulated by *reactor coolant pumps* through the nuclear reactor and large steam generators. Overpressure is provided to prevent vapor formation. The secondary side of the steam generator provides nonradioactive steam to the turbogenerator. Typical primary water conditions are 2250 lb/in^2 gauge (15.51 MPa) and 550°F (288°C). Table 1 lists the parameters of more important pumps used in PWR plants.

Figure 1 is a simplified flow diagram of a PWR reactor coolant system. Only one primary loop is shown, but in practice, plants use two, three, or four loops in parallel, each loop consisting of its own reactor coolant pump, steam generator, and optional stop valves. A single reactor and pressurizer supply all loops. The pressurizer provides the overpressure referred to by maintaining a body of water at an elevated temperature such that its vapor pressure satisfies the primary loop pressure requirements.

In the PWR plant, high-pressure water circulates through the reactor core to remove the heat generated by the nuclear chain reaction. Figure 2 is an illustration of a typical PWR reactor vessel. The heated water exits from the reactor vessel and passes via the coolant loop piping to the primary side of the steam generators. Here it gives up its heat to the feedwater to produce steam for the turbogenerator. The primary-side cycle is completed when the reactor water is pumped back to the reactor by the reactor coolant pumps. The secondary-side cycle, isolated from the primary loops, is completed when the *feedwater pumps* return water to the secondary side of the steam generator.

Because a reactor, after it has been critical, continues to generate heat even when shut down, *residual heat removal* (RHR) *pumps* are provided to circulate reactor water through coolers any time the reactor is inoperable and at low pressure, even during refueling. These pumps serve other functions also, as described next.

The chemical and volume control system (Figure 3) performs a number of functions. Through *charging pumps,* which may be centrifugal or positive displacement or may include some of each type, the primary system can be filled and pressurized when cold. When the system is hot, the pumps are used to maintain the water level in the pressurizer and to replenish any fluid drawn from the primary loops by other systems. Additionally, the pumps supply clean water to the reactor coolant pump seals and are used to adjust the boric acid concentration in the reactor coolant water, which provides an auxiliary means of reactor power regulation. If positive displacement pumps are included in the chemical and volume control system, they are also used to hydrostatically test the reactor primary coolant system. Where all the charging pumps are centrifugal, it is customary to provide a small positive displacement pump exclusively for this hydrostatic testing.

For cooling essential components and for supplying a variety of heat exchangers, a component cooling water system is provided. *Component cooling water pumps* circulate clean water at low system pressures for the purpose of cooling: (1) primary water, which is continuously bled for purification, (2) main and auxiliary pump bearings and seals, (3) primary pump thermal barriers, (4) large motors, (5) the containment vessel, and (6) the spent fuel pit water.

When the primary pressure boundary is breached, elements of the emergency core cooling system (ECCS) are immediately activated. The primary function of the ECCS following a loss-of-coolant accident is to remove the stored and fission product decay heat from the reactor core. The safety injection system (Figure 4) does most of this. Upon actuation of the safety injection signal, the charging pumps inject boric acid solution, which is stored in special tanks and continuously circulated by the *boron injection recirculation pumps,* into the reactor coolant system. At the same time, the *residual heat removal pumps* are started. These pumps take suction from a large refueling water storage tank and inject cold water into the reactor coolant circuit. To provide additional capacity, the *safety injection pumps* are started, taking suction from the cold water in the refueling water storage tank and pumping this water into the reactor coolant system. If the large storage tank should run dry, these pumps will take suction from the containment sump.

Operated by a pressure signal, *containment spray pumps* condense any steam in the containment in order to lower the temperature and pressure in that environment. By taking suction from the containment sump, these pumps continue to circulate water through spray nozzles located near the top of the containment until the pressure has been reduced to an acceptable level.

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TABLE 1 Typical nuclear pump parameters in PWR plants

Pump	Number per plant	Flow, gpm (m^3/h)	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Reactor coolant	4	100,000 (22,700)	290 (88)	2500 (17.2)	650 (343)	7000 (5220)	Vert.	305 (7750)	1200	
Component cooling water	3	4800 (1090)	250 (76)	200 (1.38)	200 (93)	450 (336)	Horiz.	110 (2790)	1800	
Residual heat removal	2	3800 (863)	350 (107)	600 (4.14)	400 (204)	500 (373)	Vert.	97 (2460)	1800	
Containment spray	2	2600 (590)	450 (137)	300 (2.07)	300 (148)	400 (298)	Horiz.	112 (2840)	1800	
Spent fuel pit cooling	3	4500 (1022)	150 (46)	150 (1.03)	200 (93)	250 (186)	Horiz.	87 (2210)	1800	
Charging (centrifugal)	2	120 (27)	5800 (1768)	2800 (19.3)	300 (148)	600 (448)	Horiz.	234 (5940)	4850	Gear drive
Charging (reciprocating)	1	98 (22)	5800 (1768)	2800 (19.3)	250 (121)	200 (149)	Horiz.	208 (5280)	205	Reciprocating
Safety injection	2	440 (100)	2680 (817)	1750 (12.07)	300 (148)	450 (336)	Horiz.	190 (4830)	3600	
Chilled water	2	400 (91)	150 (46)	150 (1.03)	200 (93)	40 (30)	Horiz.	52 (1320)	3600	
Spent resin sluicing	1	150 (34)	250 (76)	240 (1.66)	250 (121)	30 (22)	Horiz.	46 (1170)	3600	
Reactor coolant drain tank	2	150 (34)	250 (76)	240 (1.66)	250 (121)	30 (22)	Horiz.	46 (1170)	3600	

(*Continued*)

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TABLE 1 Typical nuclear pump parameters in PWR plants (*Continued*)

Pump	Number per plant	Flow, gpm (m^3/h)	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Boric acid transfer	2	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Boron recycle evaporator feed	2	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Boron injection recirculation	2	20 (4.5)	100 (30.5)	240 (1.66)	200 (93)	3 (2.2)	Horiz.	18 (460)	3600	Canned
Spent fuel pit skimmer	1	100 (22.7)	50 (15.2)	150 (1.03)	200 (93)	3 (2.2)	Horiz.	42 (1070)	1800	
Refueling water purification	1	200 (45)	200 (61)	150 (1.03)	200 (93)	30 (22)	Horiz.	52 (1320)	1800	
Waste processing system	5	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Gas decay tank drain	1	10 (2.27)	90 (27)	150 (1.03)	180 (82)	3 (2.2)	Horiz.	15 (380)	3600	
S. G. blowdown—spent resin sluice	1	110 (25)	165 (50)	150 (1.03)	100 (37)	15 (11)	Horiz.	39 (990)	3600	

Source: Westinghouse Electric.

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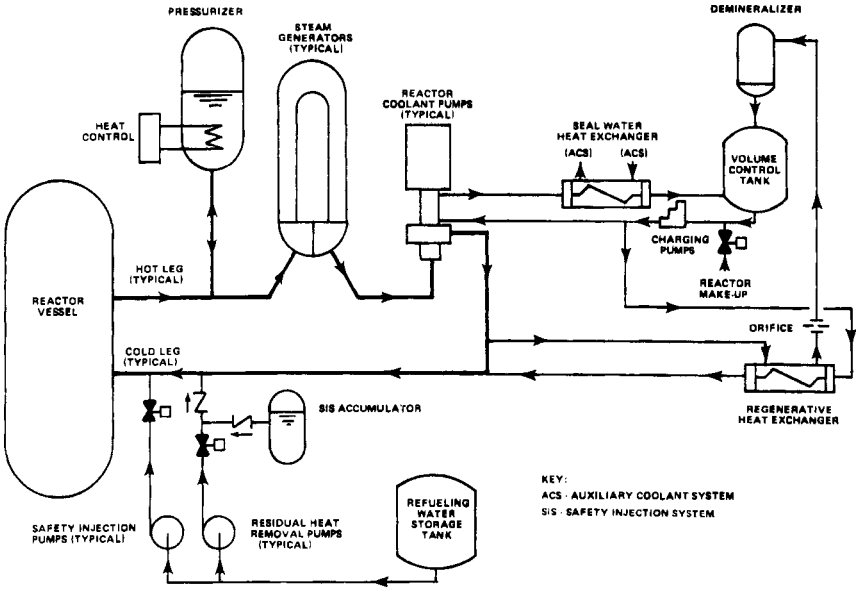


FIGURE 1 Flow diagram for pressurized water reactor coolant system (Westinghouse Electric)

All of the pumps and vital equipment associated with pipe rupture (that is, the emergency core cooling system) is supplied with diesel generator electrical power backup. If needed, the diesel generators will accept the various pump loads sequentially at intervals of a few seconds until all needed equipment is on line.

Other pumps serve other systems. *Spent fuel pit pumps* provide the necessary cooling of the fuel elements that have been removed from the reactor. Resin beds, which are a part of the water purification system, are flushed to a storage tank by *spent resin sluicing pumps.* An evaporator package, partly for removing boron from the primary water, is supplied by *recycle evaporator feed pumps. Chilled water pumps* supply the boron thermal regeneration system. Similarly, other pumps, some not listed in Table 1, support auxiliary systems.

BWR Plants In BWR plants, active boiling takes place in the nuclear core and steam is piped to the turbogenerator (Figure 5). Typical reactor water conditions are 1000 lb/in^2 gauge (6.895 MPa) and 550°F (288°C). In the United States, the plant is usually arranged with a low-leakage containment vessel completely surrounding a dry well and a pressure-suppression pool (Figure 6). The containment vessel is a cylindrical steel or concrete structure with an ellipsoidal dome and a flat bottom supported by a reinforced concrete mat. The containment forms a security barrier and prevents the escape of radioactive products to the atmosphere if an accident should occur.

Table 2 shows the principal pumps used in BWR plants together with significant characteristic data.

To assure a high reactor flow rate and to avoid local areas of core overheating, internal *jet pumps* have been used in all but the earliest U.S. BWR plants. These jet pumps are driven by large-volume, medium-head *recirculation pumps.* Variable flow rate is achieved either by flow-control valves or by variable-speed motors driven by motor generator sets. The latest designs employ a flow-control valve. The use of jet pumps decreases the size of the external loop piping and pumps and provides a core reflood capability in the event of pipe rupture. The main recirculation pumps are not required for emergency cooling. In normal operation, the recirculation pumps operate in conjunction with the jet pumps,

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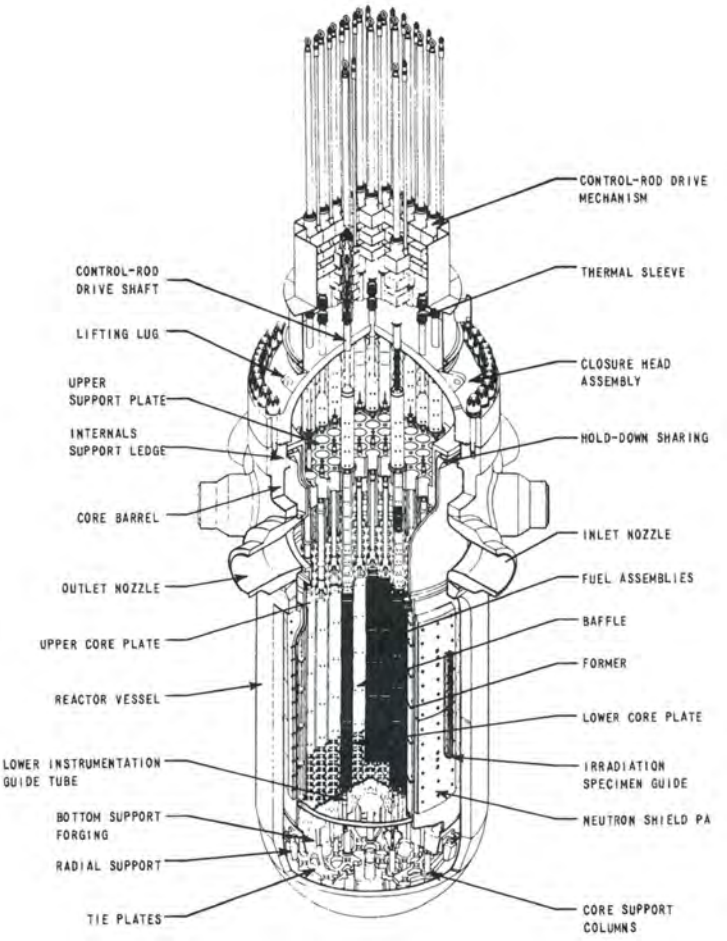


FIGURE 2 PWR reactor vessel (Westinghouse Electric)

which are wholly contained in the reactor vessel. The purpose of the jet pumps is to increase the flow from the recirculation pumps at reduced head for reactor cooling. The jet pumps have no moving parts. In addition to their normal service, they also play a role in the natural circulation of the reactor water during emergency cooling.

Several subsystems operate in support of the recirculation system, and each contains one or more pumps. *Reactor water cleanup pumps* are used in a filter-demineralizer system to remove particulate and dissolved impurities from the reactor coolant. This system also removes excess water from the reactor. The control rods are operated hydraulically with water pressure provided by the *control rod drive pumps.* The pumps are located in an auxiliary building, and the fluid is piped to the control rod drive units, which are positioned directly under the reactor vessel. For those components that require constant cooling, such as the recirculation pump motors and other equipment located in the containment, auxiliary, fuel, or radwaste buildings, *closed cooling water pumps* furnish the necessary flow. These pumps are located in an auxiliary building to permit ready access for servicing if needed. The system is closed so it can be isolated from an ultimate, usually raw water, heat sink, such as a river, lake, or ocean.

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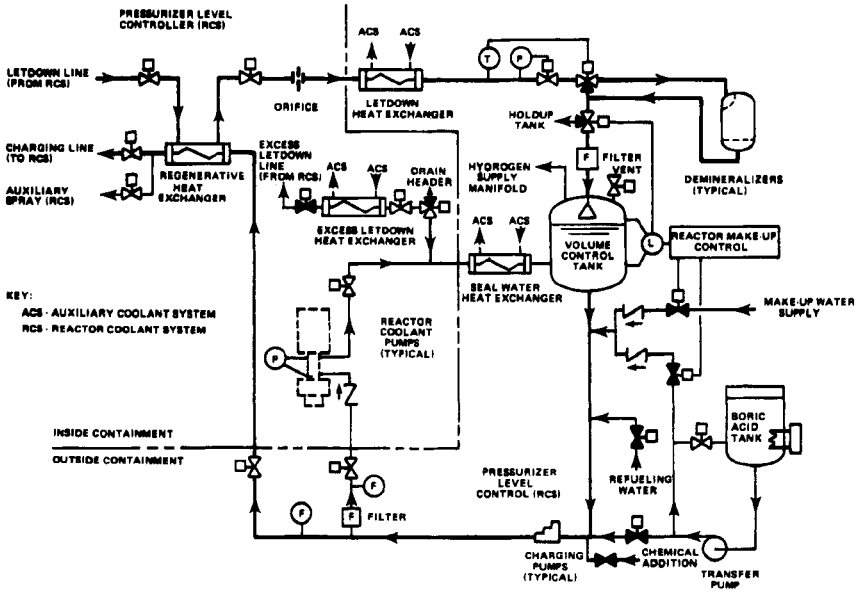


FIGURE 3 Flow diagram for chemical and volume control system (Westinghouse Electric)

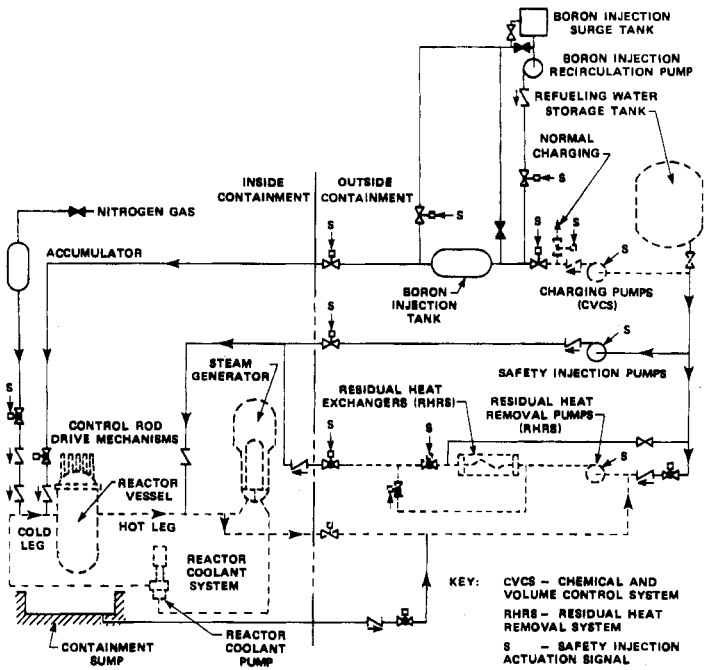


FIGURE 4 Flow diagram for safety injection system (Westinghouse Electric)

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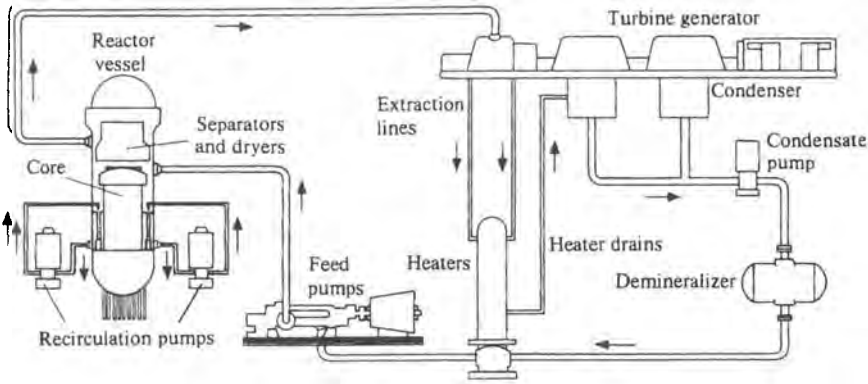


FIGURE 5 Boiling water direct cycle reactor system (General Electric)

To cool the fuel stored under water in the fuel building and the water in the upper containment, separate *fuel pool cooling pumps* are provided in an independent system.

The emergency core cooling system is in reality an array of subsystems providing the necessary features, including redundancy, to protect the core in case of a significant malfunction. The high-pressure core spray system uses a vertical *high-pressure core spray pump,* motor-driven but backed by a diesel generator in event of loss of electric power. This pump, a single unit, provides the initial response when a small pipe breaks or an equivalent malfunction occurs. Should this system be inadequate to maintain reactor water level, the reactor vessel is automatically depressurized and the *low-pressure core spray pumps* supply additional capacity. As an added safeguard, the *RHR pumps* are used in a

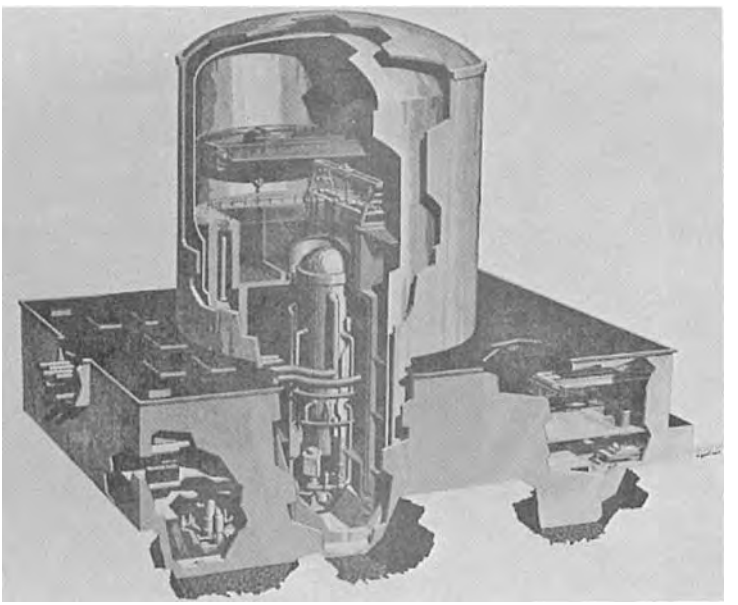


FIGURE 6 BWR containment and reactor vessel, showing one of two external recirculation pumps (General Electric)

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TABLE 2 Typical nuclear pump parameters in BWR plants

Pump	Number per plant	Flow, gpm (m^3/h)[a]	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)[a]	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Recirculation coolant	2	44000 (9993)	760 (231)	1675 (11.55)	575 (302)	8000 (5970)	Vert.	240 (6096)	1800	
RHR service water	4	7300 (1658)	115 (35)	150 (1.03)	150 (65)	300 (224)	Vert.	200 (5080)	900	
RHR	3	8520 (1935)	275 (84)	450 (3.10)	212 (100)	900 (670)	Vert.	350 (8890)	1800	
High-pressure core spray	1	1465 (333)	2600 (792)	1600 (11.03)	212 (100)	3000 (2240)	Vert.	500 (12700)	1800	
Low-pressure core spray	1	6000 (1363)	280 (85)	550 (3.79)	212 (100)	1750 (1310)	Vert.	400 (10160)	1800	
Closed cooling water	2	2040 (463)	110 (34)	150 (1.03)	212 (100)	60 (45)	Horiz.	100 (2540)	1800	
Reactor core isolation cooling	1	700 (159)	2600 (792)	1500 (10.34)	212 (100)	800 (597)	Horiz.	123 (3124)	4000	Turbine-driven variable-speed
Fuel pool cooling and cleanup	2	600 (136)	300 (91)	150 (1.03)	150 (65)	75 (56)	Horiz.	95 (2413)	1800	
Reactor water cleanup	2	150 (34)	500 (152)	1400 (9.65)	560 (293)	50 (37)	Horiz.	78 (1981)	3600	

(*Continued*)

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TABLE 2 Typical nuclear pump parameters in BWR plants (*Continued*)

Pump	Number per plant	Flow, gpm (m^3/h)[a]	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)[a]	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Control rod drive hydr. system	2	95 (22)	3500 (1067)	1750 (12.07)	150 (65)	300 (224)	Horiz.	168 (4267)	1800	
Standby liquid control	2	40 (9)	2800 (853)	1400 (9.65)	150 (65)	40 (30)	Horiz.	60 (1524)	1800	Reciprocating pumps
Jet	20	10000 (2271)	80 (24)	N.A.	575 (302)	N.A.	Vert.	250 (6350)	N.A.	
Waste evaporator	2	9000 (2044)	50 (15)	50 (0.35)	274 (134)	150 (112)	Horiz.	120 (3048)	720	
Resin tank precoat	1	255 (58)	85 (26)	150 (1.03)	150 (65)	10 (7.5)	Horiz.	65 (1651)	1800	

[a]Vary depending on reactor power rating.
Source: General Electric.

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secondary-mode operation to inject cooling water directly into the reactor vessel. If steam should enter the containment region, the *RHR pumps* operate in another mode—as containment spray pumps—and are manually operated to condense the steam and thus reduce any potential pressure build up in the containment. The *RHR pumps* function when needed to limit the temperature of the water in the suppression pool. The turbine-driven *reactor core isolation cooling pumps,* in a redundant and independent system, inject cool water into the reactor vessel. The *standby liquid control system pumps* inject boron solution into the reactor for alternative shutdown.

Additional Equipment In addition to these systems, a number of support systems exist, most of which require some form of pumping. Without attempting to describe their systems, there are, for example, pumps for feedwater, condensate, chilled water, booster service, condenser service, demineralized water transfer, condensate transfer, dry-well drain, containment drain, concentrated borated water tank, water leg, precoat, radwaste, and sample station.

MAIN COOLANT PUMPS

The term *main coolant pumps* as used here includes both the *recirculation pumps* in BWR plants and the *reactor coolant pumps* in the primary systems of PWR plants. As shown typically in Figures 7, 8, 9, and 10, main coolant pumps use a vertical shaft with the impeller at the bottom and a drive motor coupled to the top end of the shaft.

Bearings Main coolant pumps are usually single-bearing units, although one manufacturer provides a second guide-and-thrust oil-lubricated bearing just below the coupling. Within the pump, hydrodynamic water-lubricated bearings are conventionally used (one supplier using a hydrostatic type). The hydrodynamic bearing generally consists of a hardened sleeve journal shrunk on the shaft and a carbon-lined bearing with some type of self-aligning feature. Bearing diametral clearances are approximately 1.5 mi/in of bearing diameter (0.0015 mm/mm). This rather large clearance is desirable because of thermal transient conditions. Where carbon bearings are used, a cooling mechanism must be provided because the carbon is not suited to long exposure in a hot environment. Many methods are available to accomplish this.

With bearings adequately sized and operating in clean water, virtually trouble-free performance can be expected even with relatively frequent starts and stops.

The hydrostatic, or pressurized, bearing is used where it is not convenient or desirable to provide bearing cooling, for the hydrostatic bearing can be designed to operate in reactor temperature water. Usually it will be larger than its hydrodynamic equivalent and somewhat more sensitive to starting because of the metal-to-metal rubbing that occurs until rotative speed has built up a small pressure to provide a water film clearance. At normal operating speed, however, a hydrostatic bearing will have a significantly larger lubricating film than a hydrodynamic bearing and can tolerate larger particulate matter without wear.

An exception to the previous discussion occurs in both German and Swedish BWR designs, where the recirculation pumps are inverted and inserted directly into the bottom periphery of the reactor vessel and use wet winding motors. There are no jet pumps. Flow is varied by changing motor speed with solid-state power supplies, one for each recirculation pump.

Seals Pump seals are invariably of the pressure-balanced type because of the high pressures involved. Several suppliers use pressure breakdown techniques to distribute the pressure either equally or in some desired proportion between two or more seals in series (Figure 11). The technique is analogous to that of a potentiometer, where particular voltages may be obtained by selecting the proper point along an electrical resistance. Also, as with a potentiometer, the flow through the primary resistance path must exceed the tap off flow in order to maintain the system stability. For pressures in PWR systems, three series seals are frequently used (with each seal taking one-third of the overall pressure), and with lower-pressure BWRs, two series seals are usually sufficient. A margin of safety is built into the systems such that pump operation can be continued even if one of the series seals fails.

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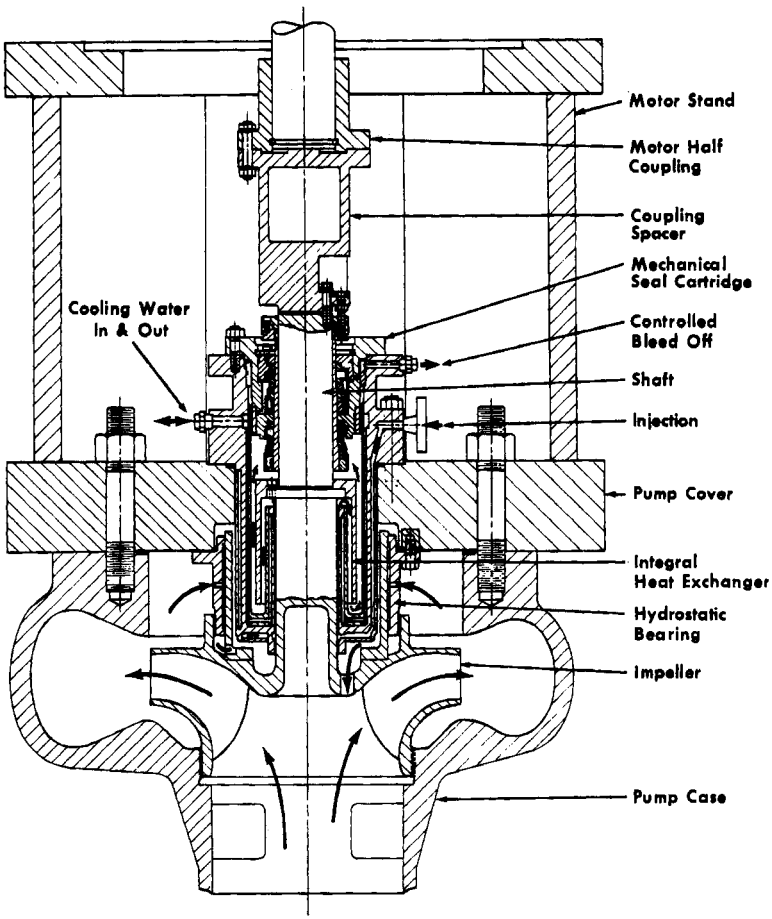


FIGURE 7 Main coolant pump (Flowserve Corporation)

In addition, a safety seal may also be installed as a backup. At least one manufacturer uses a high-pressure ceramic seal with a carbon seal backup followed by a vapor seal. Main coolant pump seals are usually supplied with reactor-grade water through a seal injection system. The advantage of using injection water is that it can be temperature controlled and filtered. Most pumps, however, can be operated without it for at least reasonable time periods.

PRINCIPAL TYPES OF PUMPS

Most of the pumps in nuclear service are one- or two-stage centrifugal motor-driven pumps. Both vertical and horizontal types are used. Charging, safety injection, feedwater, and other high-head pumps are usually multistage motor-driven centrifugal units. Some requiring high power are turbine driven. Double-suction designs are frequently used for RHR pumps, where service requires operation at low available NPSH. Reciprocating pumps find limited service in nuclear plants for makeup flow, seal injection flow, or

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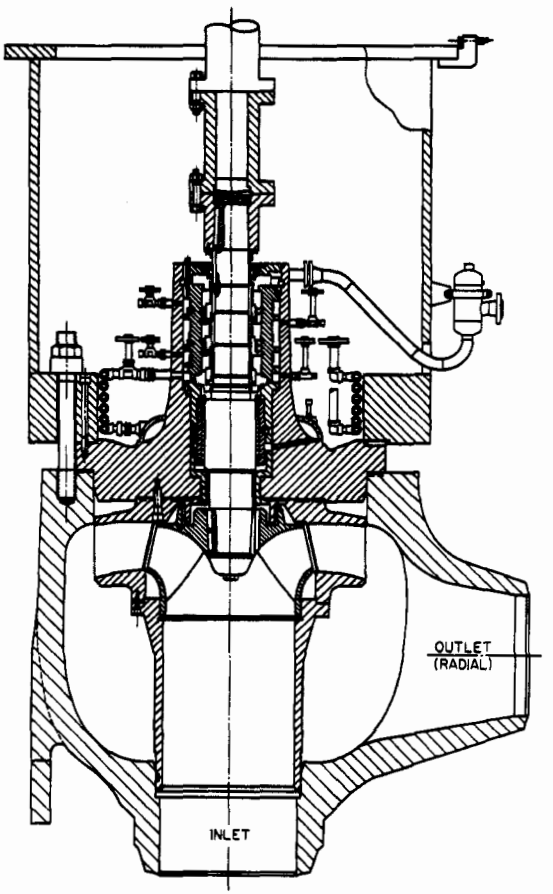


FIGURE 8 Main coolant pump (Sulzer Pumps)

chemical mixing service. Canned pumps are frequently used in subsystems where their zero-leakage capabilities can be exploited.

Figure 12 shows a multistage centrifugal pump used in charging and safety injection service.

Figure 13 illustrates a reciprocating pump often used for charging and hydrostatic test service. It is rated at 98 gpm (22 m^3/h) at 5800 ft (1768 m) head. In Figure 14, a vertical multistage unit is shown. This type of unit is common in heater drain service. Figure 15 shows a single-stage pump of a type used in many service functions in a nuclear plant. Typically, its flow is 75 gpm (17 m^3/h) with a developed head of 235 ft (71.6 m) at 3500 rpm.

SPECIAL REQUIREMENTS FOR NUCLEAR SERVICE PUMPS

It is in the area of special requirements that nuclear service pumps differ most widely from commercial products. These special requirements, described in greater detail later, far exceed the requirements of the general industrial field and illustrate the strong emphasis placed upon pressure integrity and pump operability.

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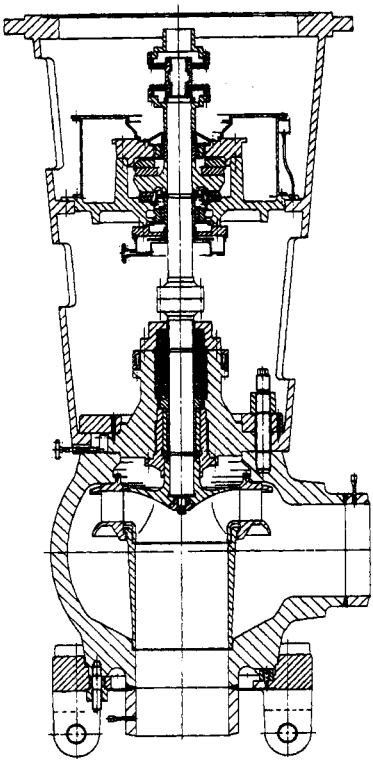


FIGURE 9 Main coolant pump (CE-KSB Pump)

Nuclear-grade pumps are designed, built, inspected, tested, and installed to rigid standards of the U.S. Nuclear Regulatory Commission, the American Society of Mechanical Engineers, the American National Standards Institute, and other regulatory agencies such as state and local jurisdictional bodies. Established rules can be divided into two distinct categories: (1) those controlling integrity of the pressure-retaining boundary and (2) functional considerations.

The hydraulic design of nuclear pumps is the same as that of pumps in conventional service. For recirculation and reactor coolant pumps, a radial discharge is preferred by some users because it tends to simplify certain aspects of the plant design.

Vibration characteristics of pumps in nuclear service are especially important because of the relative inaccessibility of the equipment for checking and servicing and because safety requirements permit only limited outage of pumps; otherwise, the plant must come to standby condition.

An analytical or experimental determination of lateral and torsional natural frequencies is routine for most pump-driver combinations, and occasionally a transient analysis may be required.

Design under ASME Code Rules Under the rules established by the ASME Boiler and Pressure Vessel Code, Section III (Nuclear Components), the owner, such as the plant designer or public utility, either directly or through an agent, prepares a design specification for a specific pump, which in code terms is a component. This specification must be approved by a licensed professional engineer and list the applicable ASME Code Year and Addenda.

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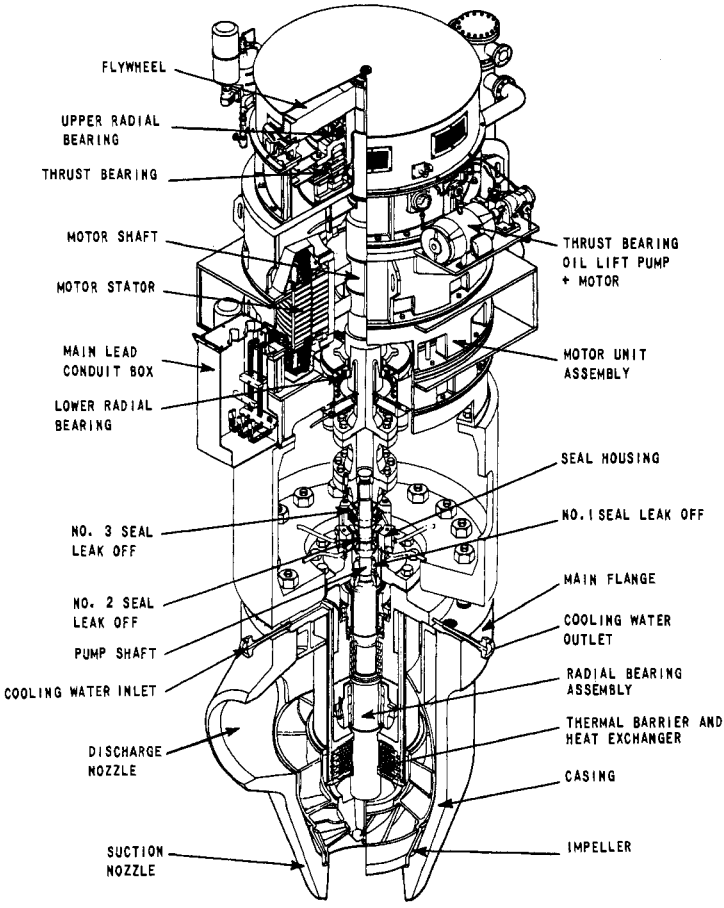


FIGURE 10 Main coolant pump (Westinghouse Electric)

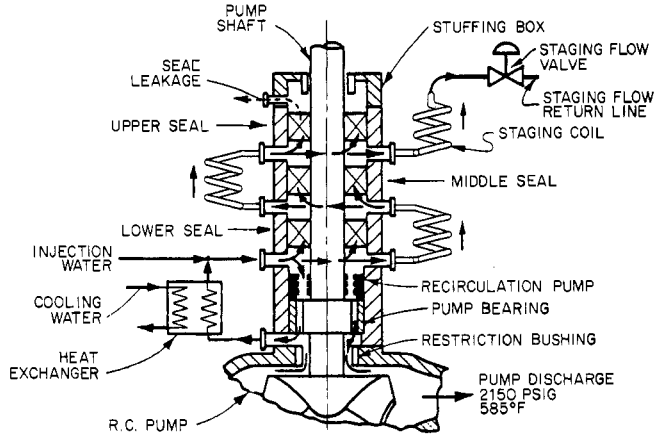


FIGURE 11 Flow schematic for a reactor coolant pump seal (Sulzer Pumps)

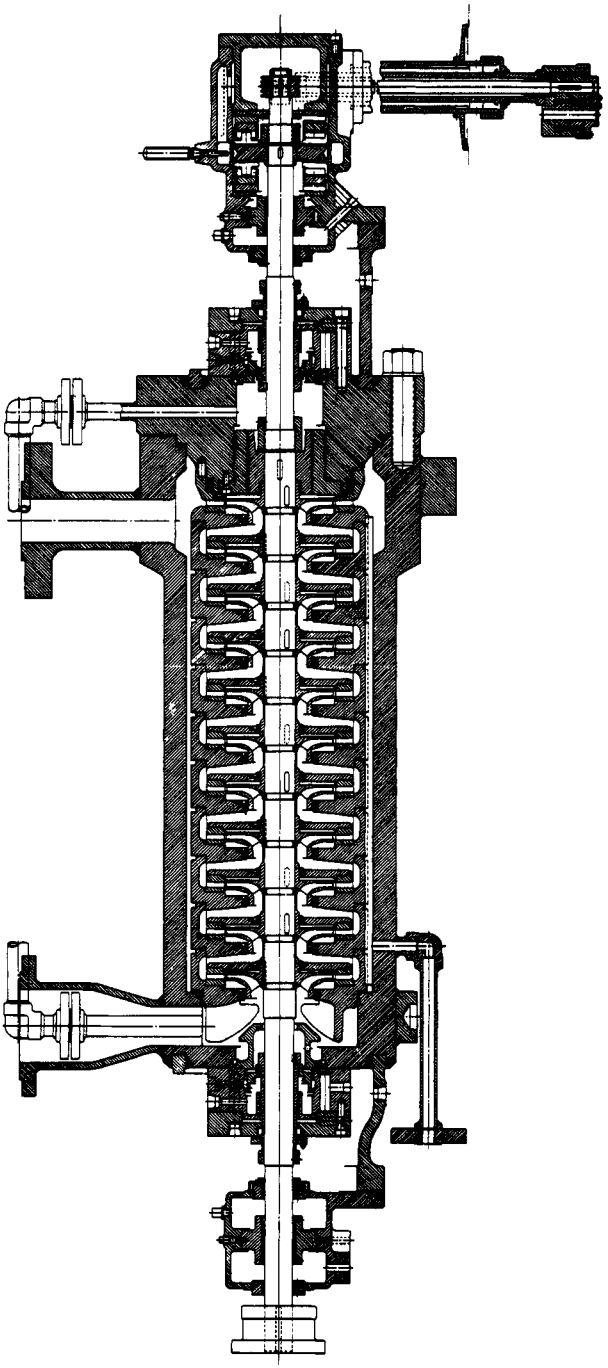


FIGURE 12 Pump for PWR charging and safety injection service (Flowserve Corporation)

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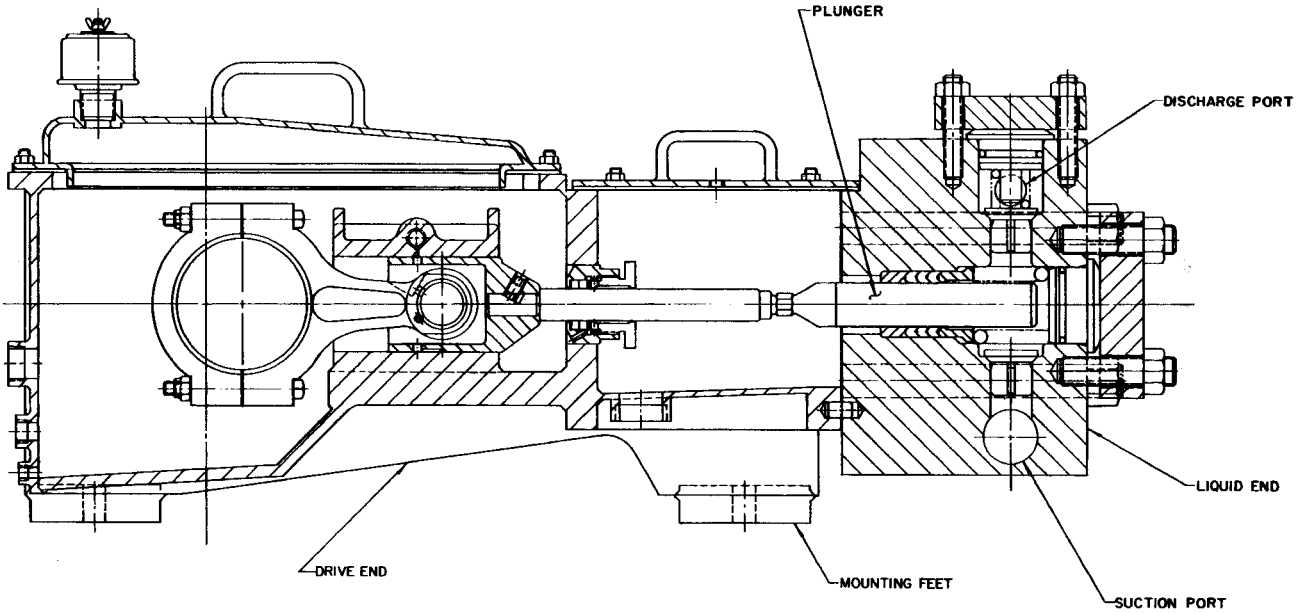


FIGURE 13 Reciprocating pump for nuclear service (Gaulin)

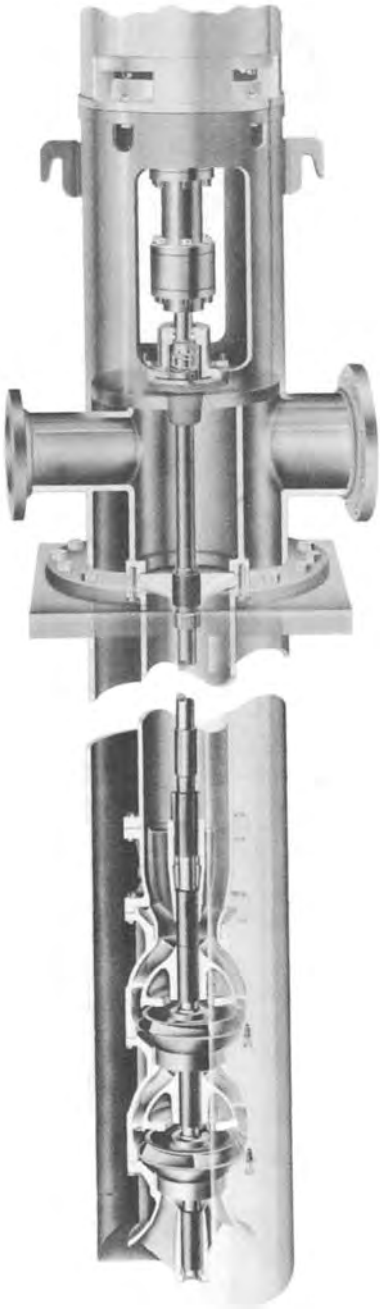


FIGURE 14 Vertical multiple-stage pump (Flowserve Corporation)

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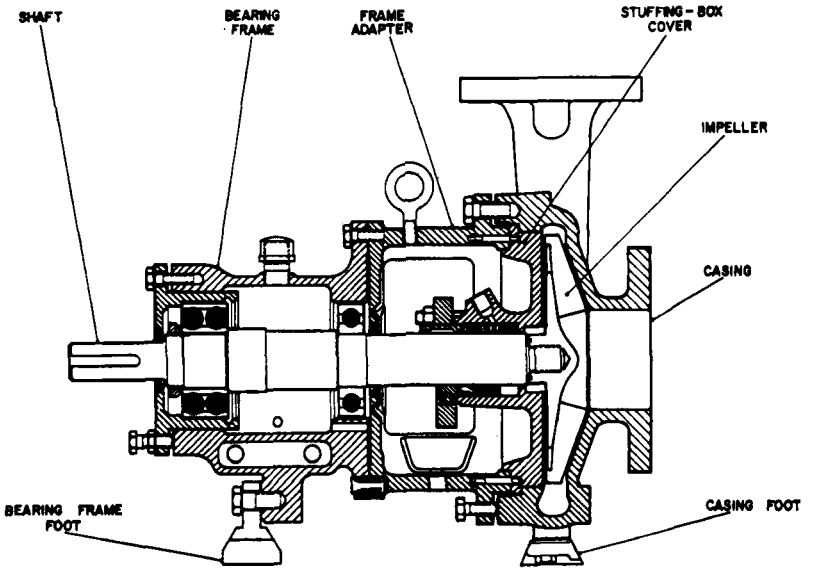


FIGURE 15 Single-stage pump used for many service functions in a nuclear plant (ITT/Gould Pumps, Inc.)

The manufacturer builds the pump to the design specification. Verification is provided through the combined efforts of the material supplier, manufacturer, insurance inspector, and state and local enforcement agencies.

The design specification requires compliance with the rules of the code with regard to the design of the pressure boundary and, in addition, includes supplementary requirements prescribed by the owner. Other standards are invoked as applicable to meet the safety and environmental requirements of the U.S. Nuclear Regulatory Commission (NRC). Functional needs may be included, and the code class to which the pump is to be built must be identified.

There are three ASME code classes for pumps. For the most critical service, a Class 1 pump is specified. Class 2 represents a pump serving a less critical system, and Class 3 is the lowest-class pump for nuclear service. It is the owner's responsibility to establish the pump class, with guidance provided by the Nuclear Regulatory Commission and the manufacturers.

For the pressure boundary evaluation, a Class 1 pump, by code rules, receives the most detailed analysis using verified and validated modern design techniques. These techniques must be supported, if necessary, by experimental stress analysis, and a certified Design Report must be submitted to the customer to document adherence to the code. Fatigue analysis of critical portions of the pump may be required, and behavior under all plant conditions, including accident, must be evaluated. Class 2 pumps require less analysis, but a certified design report is still required, documenting compliance with the customer's specification and the code. For Class 3 pumps, even wider latitude of design is permitted, but a certified design report must still be submitted to the customer. All three classes of pumps are to receive an ASME N-stamp by a qualified pump manufacturer upon successful completion of design, manufacture, inspection, test, and document review.

Additionally, nuclear service pumps are usually examined in design for thermal steady state and transient conditions, behavior under seismic disturbances, conditions of nozzle loadings imposed by system piping, means of support, and accessibility for service, in-service inspection, and replacement.

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Rules for quality control during material procurement, manufacture, and test are also contained in the code. Nondestructive examination and document control are detailed, and a quality assurance program requiring a formal procedure manual must be prepared and implemented by the manufacturer. Periodic surveys by the ASME verify adherence to code rules. Local jurisdictional authorities provide day-to-day inspection services as required by the manufacturer.

Design of Noncoded Parts Parts of the pump that are not classified as pressure boundary items or attachments to the pressure boundary are not covered by the ASME code. The owner's design specification may describe applicable requirements for these nonpressure parts, referencing other documents or excluding use of certain objectionable materials. If the pump is in critical service or is needed in emergency situations, certain ANSI standards may be invoked. Accompanying these rules will be extensive quality assurance and documentation to demonstrate compliance.

MATERIALS OF CONSTRUCTION

Material Limitations When pumps built to ASME standards are required, materials for the pressure-retaining parts must be selected from a list approved by the ASME. Section III (Nuclear Components) of the ASME Boiler and Pressure Vessel Code lists these materials, their allowable stresses, and the examination requirements that must be applied to ensure their suitability.

Typical Materials Examples of acceptable materials for pressure-retaining boundary parts are shown in Table 3. They are suitable for all three ASME code classes; however, other acceptable materials may be restricted to use for a particular class. Hazardous and porous materials are generally avoided, as are materials such as cobalt, which, though normally harmless, may become hazardous from radioactive considerations (see following text). Cobalt content is often limited in large stainless steel parts but is usually permitted in concentrated form of small areas, for example, where hard-facing is required.

Materials of construction should not be affected by the usual decontamination chemicals. Nonpressure boundary parts may be made of conventional materials but will usually require the buyer's approval. Certain elastomers, such as ethylene propylene, which has good radiation stability, are excellent for seal parts. Many fine grades of carbon-graphite are available for water-lubricated bearings and for mechanical seal facings.

TABLE 3 Materials for pressure-retaining boundary parts

Carbon steel	
Castings	SA-216, Gr WCA, WCB, WCC
Forgings	SA-105, Gr I, II
Plate	SA-515, Gr 55, 60, 65, 70
Bolting	SA-193, Gr B6, B7, B8, B16
Stainless steel	
Castings	SA-351, Gr CF8 (304), CF8M (316)
Forgings	SA-182, Gr 304, 316, 321, 347
Plate	SA-240, Gr 304, 316, 321, 347
Nonferrous	
A limited number of nonferrous materials are permitted	

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CONSIDERATIONS OF RADIOACTIVITY

Radioactivity may become a serious consideration in the design of nuclear pumps because of the need for servicing the equipment. The water used in the primary system becomes contaminated with metallic elements through solubility, corrosion, and erosion. When circulated through the core region, the metallic elements become radioactive because of interaction with neutrons. These radioactive contaminants may, if soluble, remain in solution in the water or, if insoluble, plate out on metal surfaces or become lodged in "crud traps," such as fit interfaces, screw threads, porous base metals, extremely rough surfaces, cracks, and certain types of weld configurations, such as socket welds. In one case, for instance, a pump impeller returned for overhaul defied attempts at decontamination until it was discovered that a repair had been made to a presumably integral wear ring by undercutting and shrinking on a new ring. The interface was barely perceptible, but once it had been found and the ring had been removed, the impeller was readily decontaminated.

Soluble contaminants are most easily removed by providing the pump with complete drainage features; that is, leaving no internal pockets that are not naturally drainable. Ease and speed of parts replacement are, therefore, also important items of design because they reduce the length of time service personnel are exposed to radiation.

The NRC provides specifications detailing the allowable radiation exposure to personnel. Any pump that produces radioactivity at rates that would exceed these limitations must either be repaired at a facility licensed to handle contaminated material or be decontaminated prior to being sent to a conventional repair shop.

After prolonged service in a nuclear plant, pumps may emit radioactivity at a rate in excess of 2 to 50 rem/h, far in excess of the NRC limits. Therefore, provisions to decontaminate nuclear pumps become an important design characteristic. Although not all nuclear pumps operate in highly radioactive environments, these pumps will usually require some degree of decontamination before they can be freely handled and repaired.

SEISMIC DESIGN

The ASME code requires that a seismic analysis be performed on all classes of nuclear pumps. Refer to Section 12.6.2 for more detailed information on this subject.

TESTING

Hydrostatic Testing Hydrostatic testing in accordance with ASME code rules is conventional except that specific documentation is required.

Performance Testing Performance of nuclear pumps is verified by procedures common to nonnuclear pumps. In addition, however, it may be necessary to demonstrate pump performance under some emergency condition, such as loss of cooling water or seal injection water. Also, a test under simulated total-loss-of-power conditions may be required, in which the reactor coolant pump must coast safely to a stop and withstand loss of both cooling and seal injection water for a finite time.

Periodic Testing The ASME code, under its rules for in-service inspection (Section XI), requires periodic testing of certain pumps installed in a nuclear plant. The pumps affected by these rules are those associated with the safety systems and those that may be required to function during an emergency or a reactor shutdown. Examples of such pumps are those for core spray, residual heat removal, boron injection, and containment spray. If pumps cannot be tested in their usual circuit, they must be supplied with bypass loops that can be valved off.

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The field testing procedure involves obtaining a baseline set of values for head, flow, speed (if variable), vibration, inlet pressure, and bearing temperatures. The flow quantity that sizes the bypass loop is not specified in the code, but in the practical sense, the flow must be adequate to prevent overheating of the pump and overloading of the bearings. Also, because it is an off-design point, consideration is given to the potential for rough operation due to impeller internal recirculation and possible low-flow cavitation damage. Code rules call for a 5-min running period every 3 months and longer periods where it is necessary for bearing temperatures to stabilize. If the differences between periodic operating data and baseline data exceed permissible limits, the cause for the differences must be sought and corrected. Record-keeping is therefore a significant part of pump in-service testing.

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12.6.2
NUCLEAR PUMP SEISMIC QUALIFICATIONS

D. NUTA
W. W. PARRY, JR.

Nuclear electric generating stations and their components shall be designed to resist the dynamic forces that could result from an earthquake at the site. This section covers the seismic qualification requirements for pumps classified according to function and importance based on safe operation and public safety.

SEISMIC CLASSIFICATION OF PUMPS

Pumps that must withstand a safe shutdown earthquake (SSE) and remain functional during and after such an event are designated seismic Category I. Pumps that are not required to remain functional during SSE are designated nonseismic Category I and may be classified as follows:

- Nonseismic Category I pumps are located so their failure or structural deformation might keep a seismic Category I piece of equipment or system from performing its intended function. The analysis methods and seismic input used on nonseismic Category I pumps to show that gross failure or excessive deformations will not occur are identical to those used for seismic Category I equipment.
- Nonseismic Category I pumps are located so their failure or structural deformation cannot affect any seismic Category I system or component. Seismic-related requirements for these pumps, if any, are established using a less stringent seismic environment with the aim of minimizing possible replacement expenses following a credible event.

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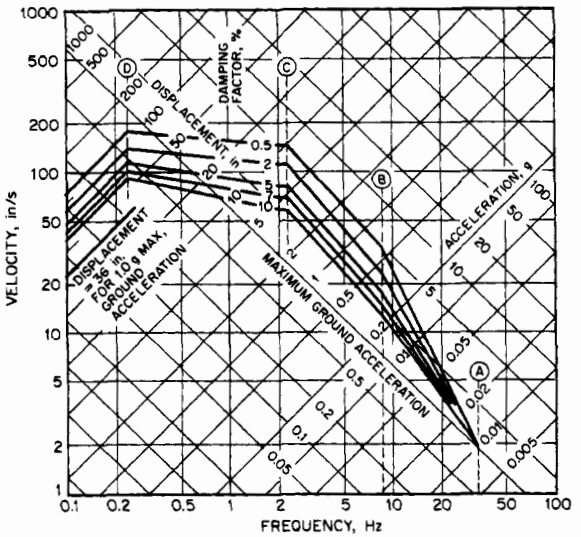


FIGURE 1 Horizontal design response spectra scaled to 1g ground acceleration (1 in = 2.54 cm; g = 32.17 ft/s^2 = 9.807 m/s^2)

DEFINITIONS

Critical Damping This is the minimum damping for which a vibrating system has no vibratory motion. For analysis purposes, damping is expressed as a percentage of critical damping. The amplification of input motion, or magnitude of response, is inversely proportional to the damping ratio of the vibrating system. Figure 1 shows responses obtained for damping ratios, which vary from 0.5 to 10% of critical damping.

Damping Damping is the dissipation of energy in a vibrating system. Damping is a function of many factors, including material characteristics, stress levels, and geometric configurations and takes place as a result of the transformation of input energy in the form of seismic motion to heat, sound waves, or other energy forms. For analysis purposes, damping is assumed to be viscous; that is, the damping force is proportional to velocity and acts in the opposite direction.

Degrees of Freedom The degrees of freedom are the coordinate(s) necessary to adequately describe the behavior of a structure or component. A node, or nodal point, may have as many as six degrees of freedom, consisting of three translations and three rotations (such as Δ_x, Δ_y, Δ_z, θ_x, θ_y, θ_z, as shown in Figure 2). This complex representation in terms of degrees of freedom is used when performing static analysis to obtain forces, stresses, or displacements.

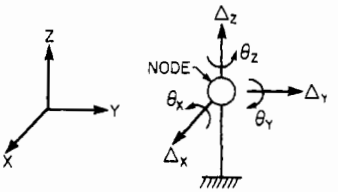


FIGURE 2 Nodal degrees of freedom

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Finite Element Method The finite element method of analysis involves the idealization of a structure or component, such as a pump, as an assemblage of elements of finite size. This mathematical model can be a lumped-mass model, a shell element model, a three-dimensional solid model, or a combination of all three. Figures 3 and 4 contain typical shell element models of horizontal and vertical pumps respectively.

Floor Acceleration This is the acceleration at a structural floor or any acceleration applicable at the equipment mounting location.

Ground Acceleration Ground acceleration is the acceleration of the ground associated with earthquake motion.

Modal Analysis Modal analysis is a type of dynamic analysis in which the response of a vibrating system is obtained as a combination of responses of the system's mode shapes.

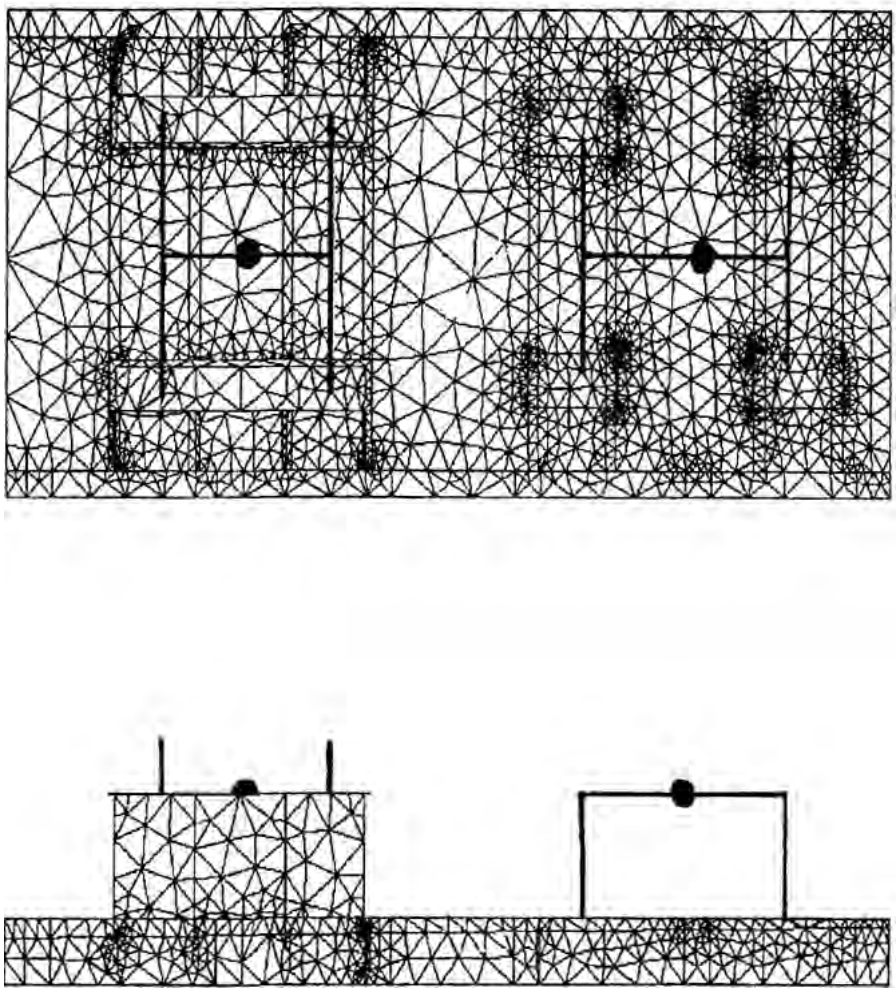


FIGURE 3 Typical FEM mesh of horizontal pump/motor assembly pump and motor are modeled as lumped-mass elements whereas bedplate is modeled with shell elements.

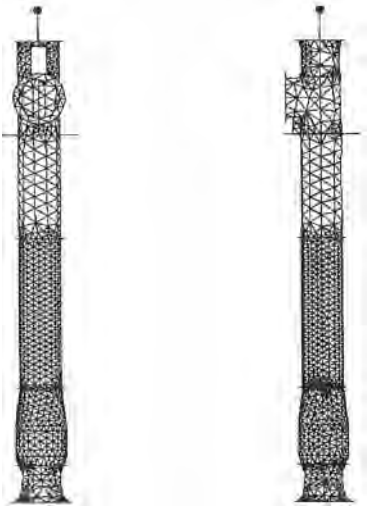


FIGURE 4 Typical FEM mesh of vertical circulating water pump. The pump is modeled with shell elements while the motor is modeled with lumped-mass elements.

(Specific requirements on the combination of modal responses is contained in U.S. Nuclear Regulatory Commission Guide 1.92.) Modal analysis may be used in conjunction with a response spectrum or time history analysis.

Natural Frequency The natural frequency is the frequency at which a body vibrates due to its own physical characteristics and to the elastic restoring forces developed when it is displaced in a given direction and then released while restrained or supported at specific points.

Operating Basis Earthquake (BE) An operating basis earthquake is an earthquake that could reasonably be expected to affect a plant site during the operating life of the plant. The systems and components necessary for continued operation of a nuclear plant without undue risk to public health and safety are designed to remain functional when subjected to the vibratory ground motion produced by an operating basis earthquake.

Required Response Spectrum (RRS) The RRS is the response spectrum for which the equipment (pump assembly) has to be dynamically qualified.

Response Spectrum The response spectrum is the plot of the maximum response (acceleration, velocity, displacement) of a series of damped single-degree-of-freedom (SDF) oscillators versus the natural frequencies (or, sometimes, periods) of the oscillators when subjected to the vibratory motion input (time history) at their supports. Figure 5 represents a plot of response spectra obtained for 0, 2, 5, and 10% of critical damping with an earthquake time history as input. The plot is on tripartite log paper, and only a limited number of SDF oscillators are shown (usually, more than 70 are needed to correctly define a spectrum).

In order to facilitate the understanding of how to use the Figure 5 plots, the 0% of critical damping response spectrum is presented in Figure 6. Assuming an SDF oscillator having a frequency of 2 cycles per second (Hz), the maximum displacement of the mass point when subjected to the earthquake record is 7 in (17.8 cm), the maximum velocity is 50 in/s (1.27 m/s), and the maximum acceleration is $2g$, where g = gravitational acceleration = 32.2 ft/s^2 (9.807 m/s^2). A line is drawn perpendicular to the frequency axis, and from the intersection of this line with the spectrum curve, a line parallel to the acceleration lines

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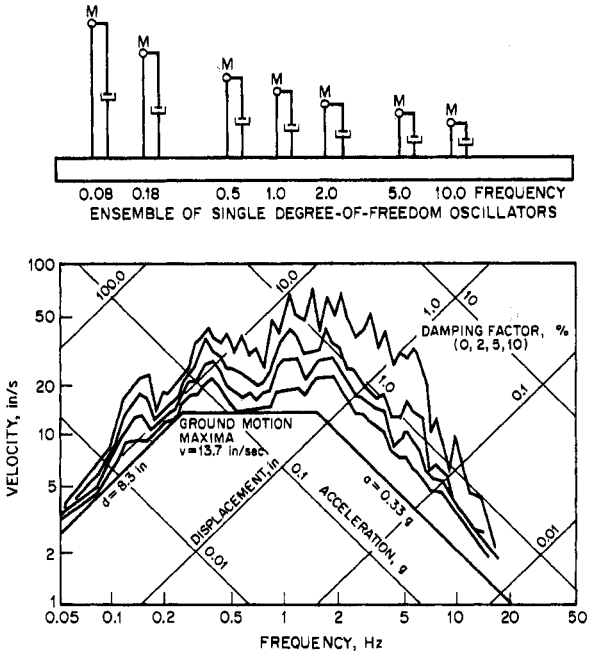


FIGURE 5 Response spectra plots for various dampings (1 in/s = 0.0254 m/s; 1 in = 2.54 cm; $1g$ = 32.17 ft/s² = 9.807 m/s²)

will provide the acceleration $a \approx 2g$, a line parallel to the displacement lines will provide the displacement d 7 ≈ in (17.8 cm), and a line perpendicular to the velocity axis will provide the velocity $v \approx 50$ in/s (1.27 m/s).

In general, response spectra are presented as plots of acceleration versus frequency. Given a time history of acceleration, such as the one presented in Figure 7, the response spectrum is obtained as described previously. The response acceleration at frequencies above

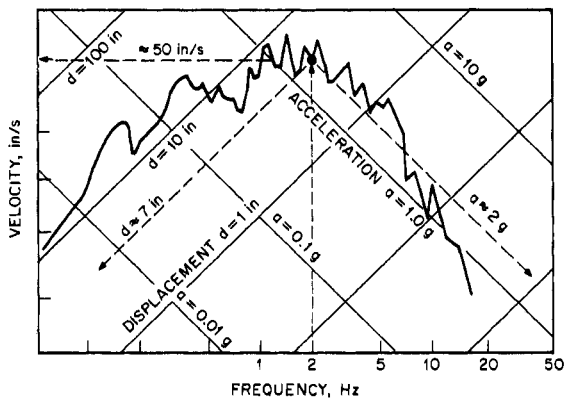


FIGURE 6 Response displacement, velocity, and acceleration versus frequency (1 in/s = 0.0254 m/s; 1 in = 2.54 cm; $1g$ = 32.17 ft/s² = 9.807 m/s²)

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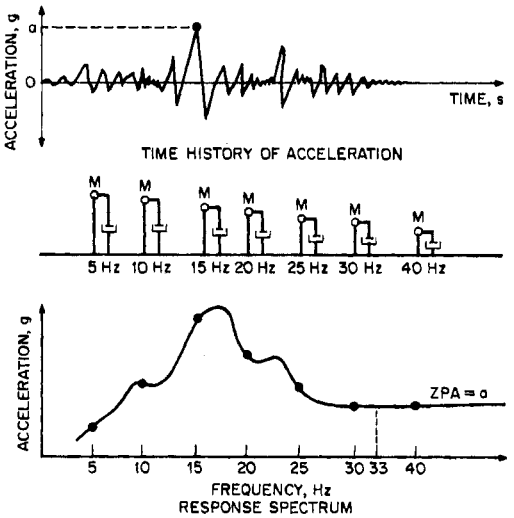


FIGURE 7 Development of acceleration response spectrum ($1g$ = 32.17 ft/s² = 9.807 m/s²)

33 Hz (also called zero period acceleration) is equal to the peak acceleration of the time history record regardless of the damping ratio used (the value a is specified in Figure 7).

Response Spectrum Analysis The response spectrum analysis is a modal dynamic analysis that uses response spectra as vibratory input at the pump attachment points.

Safe Shutdown Earthquake (SSE) An SSE is an earthquake that produces the maximum vibratory ground motion for which certain structures, systems, and components are designed to remain functional. These structures, systems, and components are those necessary to assure: (1) the integrity of the reactor coolant pressure boundary, (2) the capability to shut down the reactor and maintain it in a safe shutdown condition, or (3) the capability to prevent or mitigate the consequences of accidents that could result in potential offsite exposures to radioactive material. Various safety classes are assigned to structures, systems, and components as a function of which of these three categories they belong to.

Single-Degree-Of-Freedom Oscillator (SDF) The SDF is a vibrating system consisting of a single node point where the mass is concentrated and supported by a beam and spring element. If the single dynamic degree of freedom is along the axis of the beam, the frequency of the oscillator is

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

where f = frequency, cycles per second or Hz

$K = EA/L$ = spring stiffness, lb/in (N/mm)

E = Young's modulus of elasticity of support beam, lb/in² (Pa) A = cross-sectional area of support beam, in² (mm²)

L = length of support beam in (mm)

M = mass lb (s²/in (N · s²/mm)

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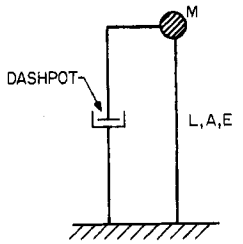


FIGURE 8 Single-degree-of-freedom oscillator

With this idealization, the frequency of the SDF oscillator may be varied by changing the length of the beam element and maintaining E, A, and M constant (Figure 8). A horizontal degree of freedom would be used in conjunction with beam bending and shearing.

Damping effects are simulated by a dashpot, which can be assigned different damping ratios. Although there is little effect on the frequency of the SDF oscillator when the dashpot is assigned damping ratios below 10 or 15% of critical damping, there is a significant effect when higher values are used. Thus, if the dashpot is assigned 100% of critical damping, the oscillator will behave as a rigid body, duplicating, without amplification, the support input motion.

Test Response Spectrum (TRS) The TRS is a response spectrum that is either theoretically constructed or derived using spectrum analysis based on the actual motion of the shake table (during dynamic testing).

Time History Analysis The time history analysis is an analysis performed in the time domain using time history vibratory input. A direct integration or modal analysis may be performed.

Time History of Acceleration The time history of acceleration is a variation of acceleration as a function of time during the postulated duration of the seismic event.

Zero Period Acceleration (ZPA) The ZPA is a response acceleration at frequencies above 33 Hz.

Although these definitions will facilitate the understanding of the seismic considerations discussed herein, the reader is encouraged to review the works listed at the end of this section for a more complete list of terms and definitions used in the process of dynamic qualifications.

SEISMIC QUALIFICATION REQUIREMENTS

The seismic qualification of seismic Category I pumps is performed to demonstrate the ability of the equipment to perform its intended function during and after the time it is subjected to an SSE, which could occur before or after several OBEs. The value of the SSE or OBE response spectra shall be supplied by the equipment buyer (nuclear facility). The simultaneous effects of the seismic accelerations in the two orthogonal horizontal directions and the vertical direction have to be considered.

The seismic qualification can be achieved by

- Analytical methods (static or dynamic analysis)
- Testing the equipment under simulated seismic conditions
- Using a combination of tests and analysis

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The information that is provided in the IEEE 344 standard on these qualification methods is applicable to pump assemblies. The information presented hereafter either emphasizes some of the IEEE 344 requirements or is pertinent to pumps and pump assemblies in particular.

Qualification by Analysis The initial step in qualifying a pump assembly by analytical methods is to determine the rigidity of the assembly. If the assembly has its lowest natural frequencies greater than the ZPA frequency, typically 33 Hz, the assembly is considered rigid. On the other hand, if one of its natural frequencies falls below the ZPA, the assembly is considered flexible. Therefore, prior to conducting the qualification analysis, the natural frequency of the pump assembly must be determined by either a modal analysis or an equipment "bump" test. The FEM models as shown in Figures 3 and 4 can be used for the modal analysis.

After the rigidity of the pump assembly has been determined, the type of qualification analysis can be selected. The allowable types of analysis are:

- **Static analysis** If it can be shown that the equipment is rigid, a static seismic analysis may be performed to determine the stresses and deformations due to the dynamic seismic loads. The dynamic forces shall be determined by multiplying the mass of the assembly times the maximum floor acceleration (ZPA from the response spectra). These forces are then applied through the center of gravity of the assembly. The resulting stresses from each acceleration force are combined by taking the square root of the sum of the squares (SRSS) to yield the total dynamic stress. The dynamic deflections shall be calculated in the same manner.
- **Simplified dynamic analysis** If it has been determined that the equipment is flexible and the customer specification permits, a simplified dynamic analysis may be completed applying the same method as the static analysis but using different values for the dynamic floor accelerations. The applicable floor acceleration shall be obtained by multiplying the acceleration values corresponding to the fundamental natural frequency from the appropriate response spectra curve by 1.5. If the fundamental natural frequency is not known, the maximum peak value of the response spectras shall be multiplied by 1.5. The 1.5 factor will conservatively account for possible participation of higher modes.
- **Detailed dynamic analysis** When justification for a static analysis cannot be provided, a detailed dynamic analysis is required, unless a conservative factor is used to account for the participation of higher modes (simplified dynamic analysis). A mathematical model of the equipment is required to determine the dynamic behavior of the equipment. This mathematical model may be a lumped-mass model, a shell model (see Figure 4), a solid element model, or a combination such as shown in Figure 3. The model should include, when applicable, the hydrodynamic mass that represents the contained water or the effects of submergence on vertical pumps. This model is then analyzed using either response spectra modal analysis or time-history (modal or step-by-step) analysis. The various modal contributions shall be combined by taking the SRSS of the individual modal stress and deformations.

These dynamic analyses are performed with computer programs in the public domain. Such programs include ANSYS, MSC/NASTRAN, ALGOR, and COSMOS/M, which incorporate acceptable methods of combining modal responses, or with a multitude of other finite element programs that are either commercially available or developed specifically for the analysis. Further information regarding specific items related to dynamic analysis, such as damping ratios, combination of modal responses, and modeling techniques may be obtained from the works listed at the end of this section. Although recent advancements in computer technology and software have made this type of analysis more readily available, when performing detailed dynamic analyses, a thorough understanding, obtained from experience, is imperative.

After these seismic stresses and deformations have been obtained, they must be combined with the other equipment stresses and deformations, resulting from all of the applicable loads, in order to determine the acceptability of the equipment.

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Qualification by Testing The equipment is tested by using the applicable seismic information (floor response spectrum or time history). The input test motion should have a TRS that closely envelopes the RRS, and the test should simulate the occurrence of five OBEs followed by one SSE.

Both the pump and the motor are tested under conditions that simulate as closely as possible actual operating conditions and operability. The operability of the pump, which is tested without the piping system attached and in the absence of any fluid, is established by the absence of undue large deflections during testing (which implies low stress levels) and by performance at the end of the test.

The input motion used must have a maximum acceleration equal to or greater than the ZPA and cannot include frequencies above the frequency associated with the ZPA. The input motion is applied to one vertical and one horizontal axis (resultant) simultaneously unless it can be proved that coupling effects are negligible. In general, multiaxis testing using independent random inputs is required.

Unless it has complete symmetry about its vertical axis, the equipment is rotated 90° and retested.

The mounting and support details used during testing should simulate the service mounting as closely as possible and should be designed so the mounting will not cause dynamic coupling to the test assembly or alter the input motion.

Combination of Test and Analysis If the motor assembly is tested separately and the pump is qualified by analysis, RRS at the motor assembly attachment points are developed from the pump assembly seismic analysis and used in the motor assembly testing. If both the pump assembly and the motor have to be tested but it is not feasible to test them together, dummy weights are used to replace the assembly part not included in the test and RRS at the location are established and used in testing the missing part.

DESIGN AND FUNCTIONAL REQUIREMENTS

The preceding sections provided the procedures for determining the seismic response of pumping equipment. However, in order to determine the acceptability of these results, they have to be combined with the pump's normal operational loads and compared to predetermined stress limitations. In accordance with the ASME code, the customer's design specification shall advise the applicable load combinations and allowable stress values.

Typically, the pump load combinations can be divided into four levels or service limits. Table 1 lists these service limits in ascending order dependent upon the magnitude of component deformation resulting from operation at the prescribed condition. In addition, Table 2 lists the ASME code allowable stress limits for each of these conditions. If no allowable stress limits are specified in the customer's design specification, no allowable stress increases should be used for the increasing load combinations, including those for OBE and SSE operation.

In addition to the specific requirements for stress levels or functionality of pressure containing components, the following items should be addressed as part of any seismic qualification of pump-motor assemblies:

- Secondary stress loading caused by differential movement of connected components should be determined.
- Displacements and deflections should be measured during testing or calculated, if qualified by analysis, in order to assure they are not excessive.
- Loads to be used in the design of the pump-motor assembly foundation and anchorage should be established.
- Seismic loads must be added to other normal shaft loads in order to verify stress levels, bearing loads, running clearances and possibly coupling misalignment.

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TABLE 1 Typical load combinations and stress limits

Plant Condition	Design/ Normal	Upset	Emergency	Faulted
Concurrent Loads	PD + SL	PD + SL + SOT	PD + SL + SOT + OBE	PD + SL + SOT + SSE
Design/Stress Limits	Level A	Level B	Level C	Level D

Where:

PD = Design pressure and temperature
SL = Sustained loads including dead weight and applied nozzle loads
SOT = System operating transients for upset, emergency, and faulted conditions per customer specifications
OBE = Operating basis earthquake
SSE = Safe shutdown earthquake
Level A = Those loadings that the pump may be subjected to in the performance of its specified function
Level B = Those loadings that the pump must withstand without damage requiring repair
Level C = Those loadings that may result in large deformations that necessitate the removal of the pump from service for inspection or repair
Level D = Those loadings that result in gross general deformations with some consequent loss of dimensional stability and damage requiring repair, which may require removal of the unit from service

TABLE 2 Stress limits

Service limits	Stress limits (maximum normal stress)
Design and Level	$\sigma_m \leq 1.0S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.5S$
Level B	$\sigma_m \leq 1.1S$
	$(z\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.65S$
Level C	$\sigma_m \leq 1.5S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.8S$
Level D	$\sigma_m \leq 2.0S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 2.4S$

σ_m = General membrane stress; equal to average stress across solid section under consideration; excludes discontinuities and concentrations and is produced only by pressure and other mechanical loads
σ_L = Local membrane stress; same as σ_m except that σ_L includes effect of discontinuities
σ_b = Bending stress; equal to linearly varying portion of stress across solid section under consideration; excludes discontinuities and concentrations and is produced only by mechanical loads
S = Allowable stress, given in Tables I-7.0 and I-8.0 of the ASME code; corresponds to highest metal temperature of section during condition under consideration

SOURCE: ASME Boiler and Pressure Vessel Code, Section III, Nuclear Power Plant Components, Division I, Subsection NC, 1992, New York.

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- For electric motors, deflections or component dislocations should be closely monitored in order to assure that:
  - the rotor will not grind into the stator.
  - the fan blades will not damage the winding insulation.
  - there is no loss of lubricant and no slipped winding or winding temperature detector.

DOCUMENTATION

Refer to IEEE 344 for a list of documents that might be required for dynamic qualification of the pump-motor assembly (by test or analysis) and for general guidance in preparing the pump assembly purchase specification.

FURTHER READING

U.S. Nuclear Regulatory Commission guides (available from NRC, Washington, D.C.).

Regulatory Guide 1.100, "Seismic Qualification of Electrical Equipment for Nuclear Power Plants."

Regulatory Guide 1.61, "Damping Values for Seismic Design of Nuclear Power Plants."

Regulatory Guide 1.70, "Standard Format and Content of Safety Analysis Reports of Nuclear Power Plants."

Regulatory Guide 1.89, "Qualification of Class 1E Equipment for Nuclear Power Plants."

Regulatory Guide 1.92, "Combining Modal Responses and Spatial Components in Seismic Response Analysis."

Industry Standards (Available from the Institute of Electrical and Electronics Engineers, New York.)

IEEE 323, "Qualifying Class 1E Equipment for Nuclear Power Generating Stations."

IEEE 344, "IEEE Recommended Practices for Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations."

Other Another source that may be helpful comes from the American Society of Civil Engineers.

American Society of Civil Engineers. "Structural Analysis and Design of Nuclear Plant Facilities." Manuals and Reports on Engineering Practices, No. 58, New York, 1980.

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SECTION 12.7
CHEMICAL INDUSTRY

GARY CLASBY
THOMAS C. SPENCE

SOLUTION CHARACTERISTICS

Chemical industries can be broadly defined as those that make, use, or dispose off chemicals. The pumps used in these industries are different from those used in other industries primarily in the materials from which they are made. Although cast iron, ductile iron, carbon steel, and aluminum- or copper-based alloys will handle a few chemical solutions, most chemical pumps are made of stainless steel, Hastelloy, the nickel-based alloys, or the more exotic metals, such as titanium and zirconium. Pumps are also available in carbon, glass, porcelain, rubber, and a variety of engineering polymers, including the thermoplastics, thermosets, epoxies, vinyl ester, and fluorocarbons. Each of these materials has been incorporated into pump design for just one reason—to eliminate or reduce the destructive effect of the chemical liquid on the pump parts.

Because the type of corrosive liquid to be pumped determines which of these materials is most suitable, a careful analysis of the chemical solution to be handled is the first step in selecting the proper materials for pump construction.

Major and Minor Constituents Foremost in importance in a study of the characteristics of any solution are the constituents of the solution. This means not only the major constituents but the minor ones as well, for in many instances the minor constituents will be the more important. They can drastically alter corrosion rates, and therefore a full and detailed analysis is most critical.

Concentration Closely allied to what the constituents are is the concentration of each. Merely stating "concentrated," "dilute," or "trace quantities" is basically meaningless because of the broad scope of interpretation of these terms. For instance, some interpret *concentrated* as meaning any constituent having a concentration of greater than 50% by weight, whereas others interpret any concentration above 5% in a like manner. Hence it is always desirable to cite the percentage by weight of each and every constituent in a given solution. This eliminates multiple interpretations and permits a more accurate evaluation. It is also recommended that the percentage by weight of any trace quantities be cited, even if this involves only parts per million. For example, titanium be completely suitable in a given environment in the absence of fluorides. If, however, the same environment contained even a few parts per million of fluorides, the titanium would suffer a catastrophic corrosion failure.

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Temperature Generalized terms such as *hot, cold,* or even *ambient* are ambiguous. The preferred terminology would be the maximum, minimum, and normal operating temperature. In general, the rate of a chemical reaction increases approximately two times with each 18°F (10°C) increase in temperature. Because corrosion can be considered a chemical reaction, the importance of temperature or temperature range is obvious.

A weather-exposed pump installation is a good illustration of the ambiguity of the term *ambient.* There could be as much as a 150°F (83°C) difference between an extremely cold climate and an extremely warm climate. If temperature cannot be cited accurately, the ambient temperature should be qualified by stating the geographic location of the pump. This is particularly important for materials that are subject to thermal shock in addition to increased corrosion rate at higher temperatures.

Acidity and Alkalinity More often than not, little consideration is given to the pH of process solutions. This may be a critical and well-controlled factor during production processing, and it can be equally revealing in evaluating solution characteristics for material selection. One reason the pH may be overlooked is that it generally is obvious whether the corrosive substance is acidic or alkaline. However, this is not always true, particularly with process solutions in which the pH is adjusted so the solutions will always be either alkaline or acidic. When this situation exists, the precise details should be known so a more thorough evaluation can be made. It is also quite important to know when a solution alternates between acidic and alkaline conditions because this can have a pronounced effect on materials selection. Some materials, although entirely suitable for handling a given alkaline or acidic solution, may not be suitable for handling a solution whose pH is changing.

Solids in Suspension Erosion-corrosion, velocity, and solids in suspension are closely allied in chemical industry pump services. Pump design is a very critical factor when the solution to be pumped contains solids. It is not uncommon for a given alloy to range from satisfactory to completely unsatisfactory in a given chemical application when hydraulic design is the only variable. Failure to cite the presence of solids on a solution data sheet is not an uncommon occurrence. The concentration of solids should be referred to as percent by volume or weight. This undoubtedly is the reason for many catastrophic erosion-corrosion failures because most corrosion resistant materials are soft and do not handle abrasion well.

Aerated or Nonaerated The presence of air in a solution can be quite significant. In some instances, it is the difference between success and failure in that it can conceivably render a reducing solution oxidizing and require an altogether different material for pump construction. A good example of this would be a self-priming nickel-molybdenum-alloy pump for handling commercially pure hydrochloric acid. This alloy is excellent for the commercially pure form of this acid, but any condition that can induce even slightly oxidizing tendencies renders this same alloy completely unsuitable. The very fact that the pump is a self-primer means that aeration is a factor to contend with, and extreme caution must be exercised in using an alloy that is not suitable for an oxidizing environment. The presence of air will not only affect the head-flow rate performance, but also the NPSHR. The maximum amount of air a conventional centrifugal pump can handle is approximately 5% by volume.

Transferring or Recirculating This item is important because of the possible buildup of corrosion product or contaminants, which can influence the service life of the pump. Such a buildup of contaminants can have a beneficial or deleterious effect, and for this reason it should be an integral part of evaluating solution characteristics.

Inhibitors or Accelerators Both inhibitors and accelerators can be intentionally or unintentionally added to the solution. Inhibitors reduce corrosivity, whereas accelerators increase corrosivity. Obviously, no one would add an accelerator to increase the corrosion rate on a piece of equipment, but a minor constituent added as a necessary part of a process may serve as an accelerator; thus the importance of knowing the presence of such constituents.

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Purity of Product Where purity of product is of absolute importance, particular note should be made of any element that may cause contamination problems, whether it be discoloration of product or solution breakdown. In some environments, pickup of only a few parts per billion of certain elements can create severe problems. This effect is particularly important in pump applications where velocity effects and the presence of solids can alter the end result, as contrasted with other types of process equipment where the velocity or solids may have little or no effect.

When a material is basically suitable for a given environment, purity of product should not be a problem. However, this cannot be an ironclad rule, particularly with chemical pumps.

Continuous or Intermittent Duty Depending upon the solution, continuous or intermittent contact can have a bearing on service life. Intermittent duty in some environments can be more destructive than continuous duty if the pump remains half full of corrosive during periods of downtime and accelerated corrosion occurs at the air-liquid interface. Stagnant conditions for certain chemicals can also cause pitting corrosion of some metals. Perhaps of equal importance is whether the pump is flushed or drained when not in service.

CORROSIVES AND MATERIALS

Metallic or Nonmetallic Materials for chemical industry pump applications can, in general, be divided into two very broad categories: metallic and nonmetallic. The metallic category can be further divided into ferrous and nonferrous alloys, both of which have extensive application in the chemical industry. The nonmetallic category can be further divided into natural and synthetic rubbers, polymers, ceramics and glass, carbon and graphite. These materials have definite application in the handling of heavy corrosives. In particular, polymers in recent years have gained widespread acclaim for their ability to handle chemicals. For a given application, a thorough evaluation of not only the solution characteristics but also the materials available should be made to ensure the most economical selection.

Source of Data To evaluate material for chemical pump services, various sources of data are available. The best source is previous practical experience within one's own organization. It is not unusual, particularly in large organizations, to have a materials group or corrosion group whose basic responsibility is to collect and compile corrosion data pertaining to process equipment in service. These sources should be consulted whenever a materials evaluation program is being conducted. A second source of data is laboratory and pilot-plant experience. Though the information from this source cannot be as valuable and detailed as plant experience, it certainly can be very indicative and serve as an important guide. The experience of suppliers can be a third source of information. Though suppliers cannot hope to provide data on the specific details of a given process and the constituents involved, they normally can provide assistance and materials for test to facilitate a decision. Technical journals and periodicals are a fourth source of information. A wealth of information is contained in these publications, but if an excellent information retrieval system is not available, it can be very difficult to locate the information desired.

Reams of information have been published in books, tables, charts, periodicals, bulletins, and reports pertaining to materials selection for various environments. It is not the intent of this section of the handbook to make materials recommendations. However, it is deemed advisable to provide some general comments and to point out a few applications having unusual characteristics. The Hydraulic Institute Standards present a very comprehensive guide for polymer material selection.

Sulfuric Acid This is the most widely used chemical in industrial applications today, and much time is spent in evaluating and selecting materials for applications involving sulfuric acid with and without constituents. The following are some of the applications that merit special consideration.

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DILUTION OF COMMERCIALLY PURE SULFURIC ACID When sulfuric acid is diluted with water, there is considerable evolution of heat. At times, the mixing of the acid and water takes place not in the mixing tank but in the pump transferring the acid. This means that heat is evolved as the solution is passing through the pump. Temperatures of 200°F (93°C) or higher are reached, depending upon the degree of dilution and the amount of mixing taking place in the pump. Thus, the heat evolved in the dilution would restrict material selection. Very few metallics or nonmetallics are resistant to 70% sulfuric acid at temperatures approaching 200°F (93°C). Refer to Section 8.2 for material guidelines for sulfuric acid.

SULFURIC ACID SATURATED WITH CHLORINE It is a well-known fact that any solution involving wet chlorine is extremely corrosive. In a solution containing sulfuric acid and chlorine, the specific weight percentage of sulfuric acid determines whether the solution will accelerate corrosion. Because of the hygroscopic nature of concentrated sulfuric acid, it will absorb moisture from the chlorine. Thus, when a sulfuric acid-chlorine solution contains at least 80% sulfuric acid, there need be little concern for the chlorine because dry chlorine is essentially noncorrosive. In such a case, a material selection can be made as if sulfuric acid were the only constituent. If the solution is saturated with chlorine but contains less than approximately 80% sulfuric acid; however, the material selection must be based not only on the sulfuric acid but also on the wet chlorine. This, of course, is a very corrosive solution, and extreme caution must be exercised in selecting the material to be used.

SULFURIC ACID CONTAINING SODIUM CHLORIDE It is quite apparent that the addition of sodium chloride to sulfuric acid will result in the formation of hydrochloric acid and thus necessitate a material that will resist the corrosive action of hydrochloric acid also. Though this may seem obvious, it is amazing how often it is ignored. This is particularly true in 10 to 15% sulfuric acid pickling solutions to which sodium chloride has been added to increase the rate of pickling, with little or no consideration being given to the destructive effect of the salt on the process equipment handling the pickling solution.

PIGMENT MANUFACTURE A slurry of titanium dioxide in sulfuric acid is one of the processing steps in the manufacture of pigment. A variety of metallics and nonmetallics would be suitable for this application in the absence of the titanium dioxide solids, but the presence of the solids circulating in a pump renders practically all of the normal sulfuric acid-resistant pump materials unsuitable. Special consideration must be given to materials that will resist the severe erosion-corrosion encountered in this type of service.

SULFURIC ACID CONTAINING NITRIC ACID, FERRIC SULFATE, OR CUPRIC SULFATE The presence of these compounds in sulfuric acid solutions will drastically alter the suitability of materials that can be used. Their presence in quantities of 1% or less can make a sulfuric acid solution oxidizing, whereas it would normally be reducing. Their presence, singly or in combination, could serve as a corrosion inhibitor, thus in certain instances allowing a stainless steel, such as type 316, to be used. On the other hand, the same compounds could serve as a corrosion accelerator for a nonchromium bearing alloy, such as nickel-molybdenum alloys, and thus render it completely unsuitable.

Nitric Acid In the concentrations normally encountered in chemical applications, nitric acid presents fewer problems than sulfuric acid. The choice of metallic materials for various nitric applications is somewhat broader than the choice of nonmetallic materials. Nitric acid, being a strongly oxidizing acid, permits the use of stainless steel quite extensively, but its oxidizing characteristics restrict the application of nonmetallics in general and plastics in particular. Requiring special evaluation are such aggressive solutions as fuming nitric acid; nitric-hydrofluoric; nitrichydrochloric (some of which fall into the aqua regia category); nitric-adipic combinations; and practically any environment consisting of nitric acid in combination with other constituents. Invariably, additional constituents in nitric acid result in more aggressive corrosion; hence material selection becomes quite critical.

Hydrochloric Acid Both commercially pure and contaminated hydrochloric acid present difficult situations in selecting pump materials. The most common contaminant that

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creates problems is ferric chloride, the presence of which can render this otherwise reducing solution oxidizing and thus completely change the material of construction that can be used. Addition of a very few parts per million of iron to commercially pure hydrochloric acid can result in the formation of enough ferric chloride to cause materials such as nickel-molybdenum, nickel-copper, and zirconium to be completely unsuitable. Conversely, the presence of ferric chloride can make titanium completely suitable. Nonmetallics find extensive application in many hydrochloric acid environments. Often the limiting factors for the nonmetallics are temperature, mechanical properties, and suitability for producing pump parts in the design desired. With the nonmetallics, the near-complete immunity from corrosion in such environments subordinates corrosion resistance to other factors. Refer to Section 8.2 for material guidelines for hydrochloric acid.

Phosphoric Acid The increasing use and demand for all types of fertilizers have made phosphoric acid a very important commodity. In the wet process of producing phosphoric acid, the phosphate rock normally contains fluorides. In addition, at various stages of the operation the solution will also contain sulfuric, hydrofluoric, fluosilicic, and phosphoric acids as well as solids. In some instances, the water used in these solutions may have an exceptionally high chloride content, which can result in the formation of hydrochloric acid, which further aggravates the corrosion problem. It is also common for certain of these solutions to contain solids, which of course create an erosion-corrosion problem. Pure phosphoric and superphosphoric acids are relatively easy to cope with from a material standpoint, but when the solution contains all or some of the aforementioned constituents, a very careful materials evaluation must be conducted. Such environments are severely corrosive in the absence of solids and cause severe erosion-corrosion and a drastically reduced service life when solids are present. This is particularly significant with any type of chemical pump.

Chlorine Little need be said about the corrosivity of chlorine. Wet chlorine, in addition to being extremely hazardous, is among the most corrosive environments known. Dry chlorine is not corrosive and carbon steel is commonly used, but there are those who contend that dry chlorine does not exist. A common definition of dry chlorine for carbon steel use is less than 50 ppm water. Chlorine vapor combined with the moisture in the atmosphere, for instance, can create severe corrosion problems. In any case, selecting the most suitable material for any type of chlorine environment requires very careful evaluation.

Alkaline Solutions With some exceptions, alkaline solutions, such as sodium hydroxide or potassium hydroxide, do not present serious corrosion problems at temperatures below 200°F (93°C). However, in certain applications, purity of product is of utmost concern, necessitating selection of a material that will have essentially no corrosion rate. Among the exceptions to the rule that alkaline solutions are relatively noncorrosive are bleaches, alkaline brines, and other solutions containing chlorine in some form.

Organic Acids Organic acids are much less corrosive than inorganic acids. This does not mean, however, that they can be taken lightly. For instance, acetic, lactic, formic, and maleic acids all have their corrosive characteristics and must be treated accordingly when evaluating metallics and nonmetallics.

Salt Solutions Normally considered neutral, salt solutions do not present a serious corrosion problem. In some instances, process streams adjust pH to maintain a slightly alkaline environment, and such solutions are even less corrosive than when they are neutral. On the other hand, when a process stream has a pH adjustment to maintain a slightly acidic environment, the liquid becomes considerably more corrosive than neutral salt solutions. This condition requires that more effort be expended in evaluating the solution before making a material selection.

Organic Compounds Most organic compounds do not present corrosion problems of the same magnitude as inorganic compounds. This does not mean that any material arbitrarily selected will be a suitable choice. It does mean that there will be more materials

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available to choose from, but each application should be considered on its own merits. Of particular concern in this area are chlorinated organic compounds and those that will produce hydrochloric acid when moisture is present. Plastics, categorically, possess excellent corrosion resistance to inorganic compounds within their temperature limitations, but they do exhibit some weaknesses in their corrosion resistance to organic compounds.

Water Water is less corrosive than most of the other mediums encountered in the chemical and allied industries. For the term *water* to be meaningful, however, it is extremely important to know the specific kind of water: demineralized, fresh, brackish, salt, boiler feed, mine. These waters and the various constituents in them can demand a variety of materials, indicated, for example, by the spectrum of materials being studied and used in desalination programs. Because they are likely to have a very pronounced effect on our total economy, precise materials evaluation and selection are integral parts of these programs.

TYPES OF PUMP CORROSION

The types of corrosion encountered in chemical pumps may at first appear to be unusual compared with those found in other process equipment. Nevertheless pumps, like any other type of chemical process equipment, experience basically only eight forms of corrosion, of which some are more predominant in pumps than in other types of equipment. It is not the intent here to describe in detail these eight forms of corrosion, but it is desirable to enumerate them and provide a brief description of each so they can be recognized when they occur.

General, or Uniform, Corrosion This is the most common type, and it is characterized by essentially the same rate of deterioration over the entire wetted or exposed surface. General corrosion may be very slow or very rapid, but it is of less concern than the other forms of corrosion because of its predictability. However, predicting the general corrosion rate in a pump can be a difficult task because of the varying velocities of the solution in the pump.

Concentration Cell, or Crevice, Corrosion This is a localized form of corrosion resulting from small quantities of stagnant solution in areas such as threads, gasket surfaces, holes, crevices, surface deposits, and the underside of bolt and rivet heads. When concentration cell corrosion occurs, the concentration of metal ions or oxygen in the stagnant area is different from the concentration in the main body of the liquid. This causes electric current to flow between the two areas, resulting in severe localized attack in the stagnant area.

Pitting Corrosion This is the most insidious form of corrosion, and it is very difficult to predict. It is extremely localized and manifested by small holes, and the weight loss due to the pits will be only a small percentage of the total weight of the equipment. Chlorides in particular are notorious for inducing pitting of stainless steels. Pitting is common in areas other than stagnant areas, whereas concentration cell corrosion is basically confined to areas of stagnation.

Stress Corrosion Cracking This is localized failure caused by a combination of tensile stresses in a specific alloy in a specific medium. Fortunately, castings, because of their basic overdesign, seldom experience stress corrosion cracking. Corrosion fatigue, which can be classified as stress corrosion cracking, is of concern in chemical pump shafts because of the repeated cyclic stressing. Failures of this type occur at stress levels below the yield point as a result of the cyclic application of the stress.

Intergranular Corrosion This is a selective form of corrosion at and adjacent to grain boundaries. It is associated primarily with stainless steels but can also occur with other

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alloy systems. In stainless steels, it occurs when the material is subjected to heat in the 800 to 1600°F (427 to 871°C) temperature range. Unless other alloy adjustments are made, this form of corrosion can be prevented only by heat-treating. It is easily detectable in castings because the grains are quite large relative to those in wrought material of equivalent composition. In some instances, uniform corrosion is misinterpreted as intergranular corrosion because of the etched appearance of the surfaces exposed to the environment. Even in ideally heat-treated stainless steels, very slight accelerated attack can be noticed at the grain boundaries because these areas are more reactive than the grains themselves. Care should be taken to avoid confusing general and intergranular corrosion. Stainless steel castings will never encounter intergranular corrosion if they are properly heat-treated after being exposed to temperatures in the 800 to 1600°F range (427 to 871°C).

Galvanic Corrosion This occurs when dissimilar metals are in contact or are otherwise electrically connected in a corrosive medium. Corrosion of the less noble metal is accelerated, and corrosion of the more corrosion-resistant metal is decreased. The farther apart the metals or alloys are in the electromotive series, the greater the possibility of galvanic corrosion. When it is necessary to have two dissimilar metals in contact, the total surface area of the less resistant metal should far exceed that of the more resistant material. This tends to prevent premature failure by simply providing a substantially greater area of the more corrosion-prone material. This form of corrosion is not common in chemical pumps but may be of some concern with accessory items in contact with pump parts and exposed to the environment.

Erosion-Corrosion This type of failure is characterized by accelerated attack resulting from the combination of corrosion and mechanical wear. It may involve solids in suspension or high velocity. It is quite common with pumps where the erosive effects prevent the formation of a passive surface on alloys that require passivity to be corrosion resistant. The ideal material for avoiding erosion-corrosion in pumps would possess the characteristics of corrosion resistance, strength, ductility, and extreme hardness. Few materials possess such a combination of properties.

Selective Leaching Corrosion This, in essence, involves removal of one element from a solid alloy in a corrosive medium. Specifically, it is typified by dezincification, dealuminumification, and graphitization. This form of attack is not common in chemical pump applications because the alloys in which it occurs are not commonly used in heavy chemical applications.

TYPES OF CHEMICAL PUMPS

The second step in selecting a chemical pump is to determine which type of pump is required, based on the characteristics of the liquid and on the desired head and flow rate. It should also be noted that not all types are available in every material of construction, and the final selection of pump type may depend on the availability of designs in the proper material. Centrifugal, regenerative turbine, rotary, and diaphragm pumps are commonly used in the chemical industry.

Centrifugal Pumps Centrifugal pumps (Figure 1) are the most commonly used type in the chemical industry because of their suitability in practically any service. They are available in a wide range of corrosion-resistant materials. Centrifugal pumps with flow rates as high as 7000 gpm (1600 m^3/h) are commonplace. Heads range as high as 990 ft (300 m) at 60 Hz 2-pole electric motor speeds. Centrifugal pumps are normally mounted in the horizontal position, but they may also be installed vertically, suspended in a tank, or hung in a pipeline similar to a valve. They are simple, economical, dependable, and efficient. Disadvantages include reduced performance when handling liquids of more than 500 SSU (108 cSt) viscosity and the tendency to lose prime when comparatively small amounts of air or vapor (3 to 5%) are present in the liquid.

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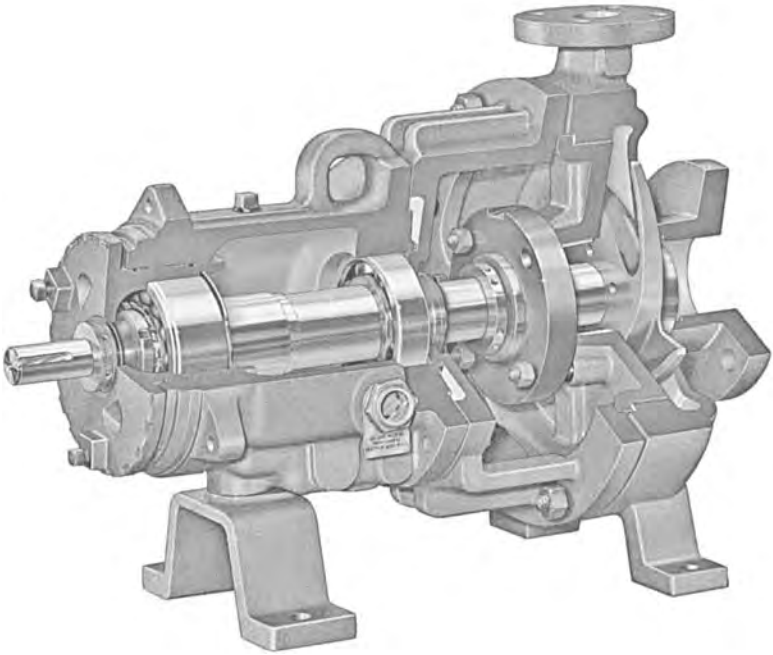


FIGURE 1 Cutaway of a typical centrifugal chemical pump (Flowserve Corporation)

Regenerative Turbine Pumps Flow rates up to 390 gpm (88 m^3/h) and heads up to 6900 ft (2100 m) are available with this type of pump at 60 Hz 2-pole electric motor speeds. When it is used for chemical service, the internal clearances must be increased to prevent rubbing contact, which results in decreased efficiency. These pumps are generally unsuitable for solid-liquid mixtures of any concentration.

Rotary Pumps Gear, screw, flexible-member, sliding-vane, piston, lobe, and cam types are particularly adept at pumping liquids of high viscosity or low vapor pressure. Because they operate on the positive displacement principle, they are inherently self-priming and their positive displacement makes them ideal for use in metering controlled volumes of liquid. The types that have close running internal clearances are generally limited to use on clear, nonabrasive liquids, and when built of materials that tend to gall or seize on rubbing contact, the clearances between mating parts must be increased, resulting in decreased efficiency.

Diaphragm Pumps These units are classed as positive displacement sealless pumps because the diaphragm acts as a limited displacement piston. Pumping action is obtained when the diaphragm is forced into reciprocating motion by mechanical linkage, compressed air, or oil from a pulsating external source. This type of construction eliminates any connection between the liquid being pumped and the source of energy and thereby eliminates the possibility of leakage. This characteristic is of great importance when toxic or very expensive liquids are being handled. Disadvantages include a limited selection of corrosion-resistant materials, limited head and capacity range, and the necessity of using check valves in the suction and discharge nozzles. Although air-operated diaphragm pumps are displacement pumps, they are not positive displacement pumps. The maximum pumping pressure cannot exceed the pressure of the compressed air powering the pump. Refer to Section 3.6 for more details on diaphragm pumps.

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CHEMICAL PUMP DESIGN CONSIDERATIONS

Casting Integrity Practically all the major components of chemical pumps are castings. There is probably more concern in chemical pump applications than in any other type of service because leakage, loss of product, and downtime can be extremely costly, as well as very dangerous.

Mechanical Properties There are several factors that determine whether a certain material can be utilized for a particular design. Materials may possess outstanding corrosion resistance but may be completely impossible to produce in the form of a chemical pump because of their mechanical properties. Accordingly, it is advisable to be aware of the mechanical properties of any material being considered in a corrosion evaluation program. Most materials are covered by ASTM or other specifications; such sources can be used for reference.

Nonmetallic Construction A number of nonmetallic materials have been used extensively in chemical pump construction (refer to Section 8.2). Their excellent chemical resistance makes them competitive with stronger metal alloys.

Weldments Welded construction should impose no limitation, provided the weldment is as good as or better than the base material. Materials requiring heat treatment to achieve maximum corrosion resistance must be treated after welding, or other adjustments must be made to make certain corrosion resistance is not sacrificed.

Section Thickness Pressure-containing parts are generally made thicker than required for handling a noncorrosive liquid so that full capability will be maintained even after the loss of some material to the corrosive medium. Parts that are subject to corrosion from two or three sides, such as impellers, must be made considerably heavier than their counterparts in water or oil pumps. Pressure-containing parts are also made thicker so they will remain serviceable after a specified amount of corrosive deterioration. Areas subject to high velocities, such as the cutwater of a centrifugal pump casing, are reinforced to allow for the accelerated corrosion caused by the high velocities.

Threads Threaded construction of any type in the wetted parts must be avoided whenever possible. The thread form is subject to attack from two sides, and a small amount of corrosive deterioration can reduce the holding power of the threaded joint.

Gaskets Gasket materials must resist being corroded by the chemical being handled. Compressed synthetic fibers and elastomers have been used extensively for corrosive services. Fluorocarbon resins are used because of their almost complete corrosion resistance.

Power End This assembly, consisting of the bearing housing, bearings, oil or grease seals, and bearing lubrication system, is normally made of iron or steel components; thus it must be designed to withstand a severe chemical plant environment. For example, where venting of the bearing housing is required, special means of preventing the entrance of water, chemical fumes, or dirt must be incorporated into the vent design.

The bearing that controls axial shaft movement is usually selected to limit shaft movement to 0.002 in (0.05 1 mm) or less. Endplay values above this limit have been found detrimental to impeller and mechanical seal operation.

Water jacketing or fan cooling of the bearing housing may be necessary under certain conditions to maintain bearing temperatures below 180°F (82°C), the limit used in most applications.

Maintenance Maintenance of a chemical pump in a corrosive environment can be a very costly and time-consuming item. When evaluating materials and design factors,

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maintenance aspects should be high on the priority list. The ease and frequency of maintenance are critical items and should be considered part of a preventive maintenance program. Such a program can be the most effective way of eliminating emergency shutdowns caused by pump failure. Furthermore, the knowledge gained in a routine preventive maintenance program can be of unlimited value when a breakdown does occur because repair personnel will have acquired a thorough knowledge of the construction details of the pump.

SEALING

The area around the sealing chamber probably causes more chemical pump failures than all other parts combined. The problem of establishing a seal between a rotating shaft and the stationary pump parts is one of the most intricate and vexing problems facing the pump designer.

Packings Braided fluorocarbon resins, aluminum, graphite, and many other materials or combinations of materials have been used to establish a seal (discussed in more detail in Section 2.2.2). A small amount of liquid must be allowed to seep through the packing to lubricate the surface between packing and shaft. This leakage rate is hard to control, and usually the packing is overtightened and the leakage is stopped. The unfortunate results of this condition is rapid scoring of the sealing surface, making it much harder to adjust the packing.

Mechanical Seals Mechanical shaft seals as described in Section 5.2 are used extensively on chemical pumps. The majority of chemical pumps in service today do not use packing. The primary consideration is selection of the proper materials for the type of corrosive liquid being pumped. Stainless steels, ceramics, graphite, and fluorocarbon resins are used to make most seal parts. Seals consist of rotating and stationary elements that, in the past, have been assembled into the pump as separate pieces. However, cartridge seals have now become an industry standard, where the seal, sleeve, and gland are preassembled as a unit, with no adjustment required, to simplify installation in the pump and improve seal operating life and reliability.

Seal Chambers Most chemical pumps are available with traditional convertible sealing chambers that can accept packing or mechanical seals. These "stuffing boxes" have been sized for packing and require narrow cross-section mechanical seals that are prone to early failure because of excessive temperature rise and trapped air. Chemical pumps are now available with oversized bore sealing chambers with additional liquid for cooling the seal and tapered bore sealing chambers with excellent seal cooling from fluid exchange with the pump and no trapped air, as they are self-venting.

Temperature One of the most important factors affecting the sealing medium is their operating temperature. Increased temperatures can result from high process temperatures and from heat generated by the sealing device. High temperature can increase the corrosive attack on the sealing chamber, the packing or the mechanical seal. Seal parts not designed for high temperature can also distort or crack and fail.

One answer to the heat/temperature problem is to cool the sealing chamber with a water jacket that surrounds it and the seal gland. These jackets tend to be poor heat exchangers, however, and water can be expensive—or nonexistent—in many locations. There is also the risk that the cooling water source will fail while the pump is running. These factors are contributing to the requirement that chemical pumps operating between 200 and 500°F (93 and 260°C) be selected with mechanical seals that can handle full process temperature. These pumps are then applied with oversized sealing chambers with product flush or tapered sealing chambers to maximize seal cooling and to avoid trapped air. In many cases, cooling water is no longer required.

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Pressure Seal chamber pressure varies with suction or discharge pressure depending on its location in the pump and impeller design. Variations in impeller design include those using vertical or horizontal seal rings in combination with balance ports, or the use of back vanes, pump-out vanes, or pump-out slots. All impeller designs depend upon a close running clearance between the impeller and the stationary pump parts. This clearance must be kept as small as possible to prevent excessive recirculation of the liquid and the resulting loss of efficiency. Unfortunately, most chemical pump materials tend to seize when subjected to rubbing contact, and the running clearances must therefore be increased considerably above those used in pumps for other industries.

At pressures above 100 lb/in^2 (690 kPa), packing is generally unsatisfactory. Mechanical seals incorporating a pressure balancing feature to relieve high face loading are the best means of sealing at pressures above 100 lb/in^2 (690 kPa).

Shaft Pump shaft bending—or deflection—can create additional sealing problems. Undersize shafts, or those made of materials that bend readily, will deflect from their true center in response to radial thrust on the impeller.

Mechanical seal operation is impaired when the shaft is bent or deflected during operation. Because the flexible member of the seal must adjust with each revolution of the shaft, excessive deflection results in shortened seal life. If the deflection is of more than nominal value, the flexible seal member will be unable to react with sufficient speed to keep the seal faces together, allowing leakage at the mating faces.

A limit of 0.002 in (0.05 mm) at the seal faces has been established as the maximum allowable shaft deflection consistent with good pump design and seal life. Operation of the pump outside the allowable operating flow region can also increase radial thrust and shaft deflection, shortening seal life.

Shaft Surface In the seal chamber region, the shaft surface must have corrosion resistance at least equal to and preferably better than that of the wetted parts of the pump. In addition, this surface must be hard enough to resist the tendency to wear under the packing or mechanical seal parts. Further, it must be capable of withstanding the sudden temperature changes often encountered in operation.

Often it is cost effective to make the shaft from high-strength carbon steel, and then add protective sleeve of stainless steel, plastic, carbon, glass, or a coating in the seal chamber area. Cylindrical sleeves are made so they can be removed and replaced when they become worn. Other designs utilize sleeves that are permanently bonded to the shaft. Where possible, shafts are increasingly being made from solid stainless steel to maximize the shaft diameter under the seal, reducing bending and increasing seal life. This is most effective when cartridge seals are used as they include their own sleeve.

Another method of obtaining a hard surface in this region is the welded overlay or spray coating of hard metals onto the base shaft. Ceramic materials applied by the plasma spray technique possess excellent corrosion resistance but cannot achieve the complete density required to protect the underlying shaft.

Composite shafts utilizing carbon steel for the power end and a higher alloy for the wet end have been used extensively where the high-alloy end has acceptable corrosion resistance. The two ends are joined by various welding techniques, and the combination of metals is therefore limited to those that can be easily welded together. On such assemblies, the weld joint and the heat-affected zone must be outside the wetted area of the shaft.

Sealless Pumps Elimination of the sealing chamber and its associated problems has been the objective of several pump designs. Refer to Section 2.2.3 for sealless canned motor pumps and magnetically driven pumps. Diaphragm pumps previously mentioned are sealless pumps.

Vertical immersion pumps utilize sleeve bearings in the area immediately above the impeller to limit the flow of liquid along the shaft. For chemical service, the problem of materials associated with this bearing and its lubrication has to be addressed on an application basis.

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CONSTRUCTION OF NONMETALLIC PUMPS

A number of nonmetallic materials have been used extensively in chemical pump construction (refer to Section 8.2). Their excellent chemical resistance makes them competitive with stronger metal alloys.

CHEMICAL PUMP STANDARDS

In 1962, a committee of the Manufacturing Chemists Association (MCA) reached agreement with a special committee of the Hydraulic Institute on a proposed American Standards Association (ASA) standard for chemical process pumps. This document was referred to as the American Voluntary Standard or the Manufacturing Chemists Association Standard. In 1971, it was accepted by the American National Standards Institute (ANSI) and issued as ANSI Standard B123.1. This ANSI standard was renumbered to ANSI B73.1 in 1974, to ANSI/ASME B73.1M in 1984, to ASME B73.1M in 1991, and to ASME B73.1 in 2001.

It is the intent of this specification for horizontal end suction centrifugal pumps for chemical process that pumps of similar size from all sources of supply shall be dimensionally interchangeable with respect to mounting dimensions, size, and location of suction and discharge nozzles, input shafts, base plates, and foundation bolts. Table 1 lists the pump dimensions that have been standardized, and a cross-sectional assembly of a pump meeting these criteria is shown in Figure 1.

It is also the intent of this standard to outline certain design features that will minimize maintenance problems. The standard states, for instance, that the pump shaft should be sized so the maximum shaft deflection shall not exceed 0.005 in (0.127 mm) at the centerline of the impeller when the pump is operating within its design region. It does not specify shaft diameter because impeller diameter, shaft length, and provision for operation with liquids of high specific gravity would determine the proper diameter.

The standard also states that the minimum bearing life shall be 17,500 h within the design region of the pump. Bearing size is not specified but is to be determined by the individual manufacturer and will be dependent upon the load to be carried.

Additional specifications in the standard include hydrostatic test pressure, shaft finish at rubbing points, packing space, and seal chamber space.

ASME B73.2 is the specification for vertical in-line centrifugal pumps for chemical process. Dimensional criteria are shown in Table 2, and a pump meeting these requirements is shown in Figure 2.

ASME B73.3 is the specification for sealless horizontal end suction centrifugal pumps for chemical process. It gives dimensional and other technical requirements for both magnetic drive and canned motor pumps.

ASME B73.5 is the specification for thermoplastic and thermoset polymer material horizontal end suction centrifugal pumps for chemical process. A typical composite pump meeting these requirements is shown in Figure 3.

Standards for acceptance tests are given in ANSI/HI 1.6, the American National Standard for Centrifugal Pump Tests.

The International Organization for Standardization (ISO) also publishes standards for chemical pumps. ISO 2858 gives the nominal duty point and dimensions for end suction horizontal centrifugal pumps. ISO 3069 gives the dimensions of cavities for mechanical shaft seals and soft packing for pumps conforming to ISO 2858. ISO 3661 gives the dimensions of baseplates for pumps conforming to ISO 2858. ISO 5199 gives the technical specifications of pumps conforming to ISO 2858.

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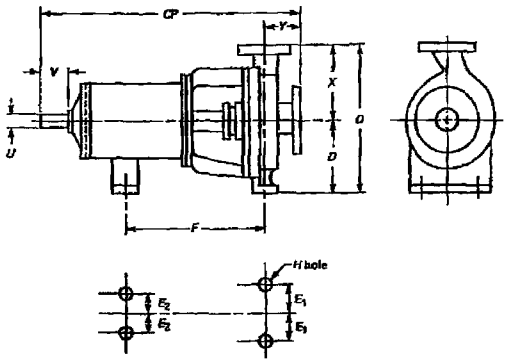


TABLE 1 ASME B73.1 Horizontal pump dimensions

Dimension Designation	Size; Suction × Discharge × Nominal Impeller Diameter	CP	D	$2E_1$	$2E_2$	F	H	O	U [Note (1)] Diameter	U [Note (1)] Keyway	Vmin	X	Y
AA	1.5×1×6 (40×25×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AB	3×1.5×6 (80×40×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AC [Note (2)]	3×2×6 (80×50×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AA [Note (2)]	1.5×1×8 (40×25×200)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AB [Note (2)]	3×1.5×8 (80×40×200)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
A10	3×2×6 (80×50×150)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.5 (420)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.25 (210)	4 (102)
A50	3×1.5×8 (80×40×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A60	3×2×8 (80×50×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	17.75 (450)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	9.5 (242)	4 (102)
A70	4×3×8 (100×80×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	19.25 (490)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11 (280)	4 (102)
A05 [Note (2)]	2×1×10 (50×25×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A50	3×1.5×10 (80×40×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A60	3×2×10 (80×50×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	17.75 (450)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	9.5 (242)	4 (102)

(*Continued*)

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TABLE 1 ASME B73.1 Horizontal pump dimensions (*Continued*)

Dimension Designation	Size; Suction × Discharge × Nominal Impeller Diameter	*CP*	*D*	$2E_1$	$2E_2$	*F*	*H*	*O*	*U* [Note (1)] Diameter	*U* [Note (1)] Keyway	*V*min	*X*	*Y*
*A*70	4×3×10 (100×80×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	19.25 (490)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11 (280)	4 (102)
*A*40	4×3×10 (100×80×250)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	22.5 (572)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	12.5 (318)	4 (102)
*A*50 [Note (3)]	6×4×10 (150×100×250)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	23.5 (597)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	13.5 (343)	4 (102)
*A*20 [Note (2)]	3×1.5×13 (80×40×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	20.5 (520)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	10.5 (266)	4 (102)
*A*30	3×2×13 (80×50×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	21.5 (548)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11.5 (292)	4 (102)
*A*40	4×3×13 (100×80×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	22.5 (572)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	12.5 (318)	4 (102)
*A*80 [Note (3)]	6×4×13 (150×100×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	23.5 (597)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	13.5 (343)	4 (102)
*A*90 [Note (3)]	8×6×13 (200×150×330)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*100 [Note (3)]	10×8×13 (250×200×330)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*105 [Note (3)]	6×4×15 (150×100×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*110 [Note (3)]	8×6×15 (200×150×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*120 [Note (3)]	10×8×15 (250×200×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	33.5 (851)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	19 (483)	6 (152)
*A*105 [Note (3)]	6×4×17 (150×100×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*110 [Note (3)]	8×6×17 (200×150×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*120 [Note (3)]	10×8×17 (250×200×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	33.5 (851)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	19 (483)	6 (152)

GENERAL NOTES:

(a) Dimensions in parentheses are approximate equivalents in millimeters.

(b) All other dimensions are in inches.

NOTES:

(1) "*U*" Diameter may be 1.625 in (41.28 mm) in *A*05 through *A*80 sizes to accomodate high torque values.

(2) Discharge flange may have tapped bolt holes.

(3) Suction flange may have tapped bolt holes.

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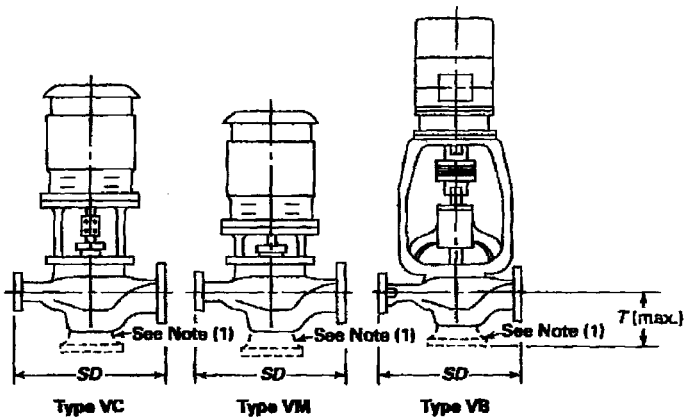


TABLE 2 ASME B73.2 Vertical in-line pump dimensions

Dimensions, mm					Dimensions, in				
Standard Pump Designation [Note (2)]	Metric Flange Sizes		SD 2.5 –2.0	T, Max.	Standard Pump Designation [Note (2)]	ANSI 125, 150, 250, or 300 Flange Sizes		SD +0.10 –0.08	T, Max.
	Suction	Discharge				Suction	Discharge		
50-40-380	50	40	380	175	2015/15	2	1.5	14.96	6.89
50-40-430	50	40	430	175	2015/17	2	1.5	16.93	6.89
50-40-480	50	40	480	175	2015/19	2	1.5	18.90	6.89
80-40-380	80	40	380	200	3015/15	3	1.5	14.96	7.87
80-40-480	80	40	480	200	3015/19	3	1.5	18.90	7.87
80-40-610	80	40	610	200	3015/24	3	1.5	24.02	7.87

(*Continued*)

TABLE 2 ASME B73.2 Vertical in-line pump dimensions (*Continued*)

Dimensions, mm					Dimensions, in				
Standard Pump Designation [Note (2)]	Metric Flange Sizes		SD 2.5 –2.0	*T*, Max.	Standard Pump Designation [Note (2)]	ANSI 125, 150, 250, or 300 Flange Sizes		SD +0.10 –0.08	*T*, Max.
	Suction	Discharge				Suction	Discharge		
80-50-430	80	50	430	200	3020/17	3	2	16.93	7.87
80-40-510	80	50	510	200	3020/20	3	2	20.08	7.87
80-50-610	80	50	610	200	3020/24	3	2	24.02	7.87
100-80-560	100	80	560	225	4030/22	4	3	22.05	8.86
100-80-635	100	80	635	225	4030/25	4	3	25.00	8.86
100-80-710	100	80	710	225	4030/28	4	3	27.95	8.86
150-100-610	150	100	610	250	6040/24	6	4	24.02	9.84
150-100-710	150	100	710	250	6040/28	6	4	27.95	9.84
150-100-760	150	100	760	250	6040/30	6	4	29.92	9.84

NOTES:

(1) Optional separate pedestal.

(2) Pump designation defines design, flange sizes, and SD dimension (e.g., VC, VB 50-40-380).

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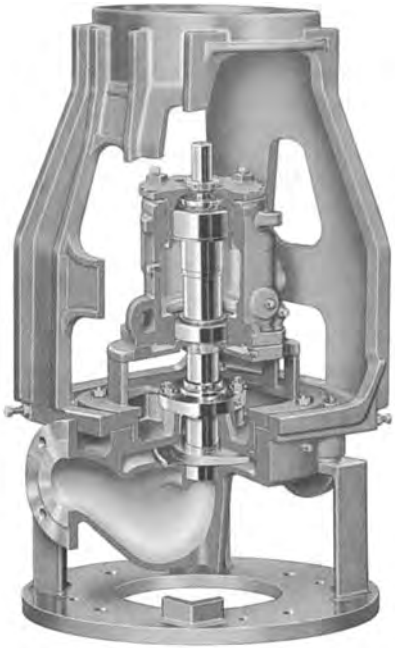


FIGURE 2 Vertical in-line centrifugal pump for chemical process applications (Flowserve Corporation)

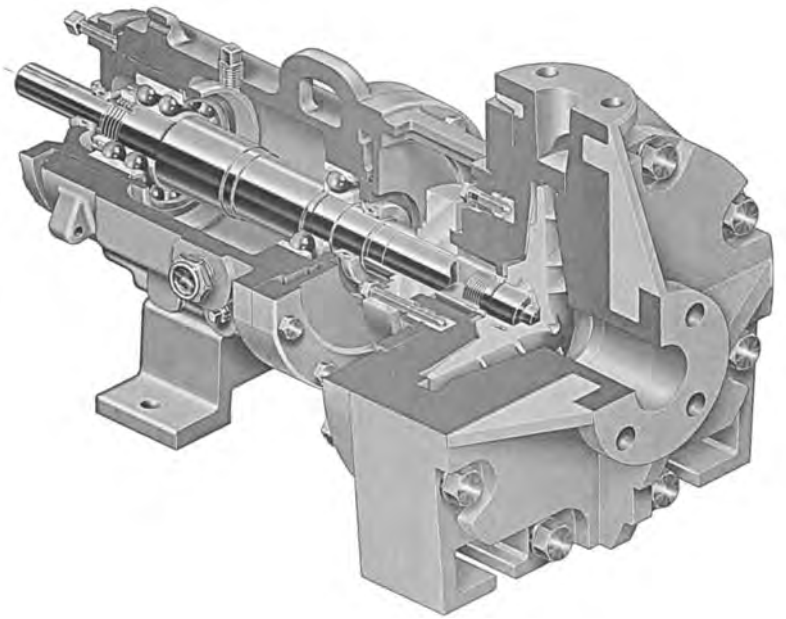


FIGURE 3 A typical composite pump (Flowserve Corporation)

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SECTION 12.8
PETROLEUM INDUSTRY

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Every phase of petroleum production, transportation, and refining requires a variety of pump types.

Production pumps include reciprocating units for mud circulation during drilling operations and sucker-rod, hydraulic rodless, and motor driven submersible centrifugal units for lifting crude to the surface. The most common use of centrifugal pumps in production is for water flooding (secondary recovery, subsidence prevention, or pressure maintenance).

Transportation pumps include units for gathering, on and offshore production, pipelining crude and refined products, loading and unloading tankers, tank cars, barges, or tank trucks, and servicing airport fueling terminals. The majority of the units are centrifugal-type pumps.

While refining units use all types of pumps, the vast majority of them are centrifugal as well. They vary from single-stage centrifugal units to horizontal and vertical multistage barrel-type pumps handling a variety of products over a full range of temperatures and pressures. Centrifugal pumps are also used for auxiliary services, such as boiler feedwater, cooling water, and rainwater transfer.

Except for some comments about the use of displacement pumps for handling viscous liquids, this section is restricted to centrifugal pumps. It also includes an overview of the requirements for some of the principal types of centrifugal pumps.

REFINERY PUMPS

Major refinery processes include crude distillation, vacuum tower separation, catalytic cracking, alkylation, hydrocracking, catalytic reforming, coking, and hydrotreatment for the removal of sulfur and nitrogen. The products resulting from these processes include motor gasoline, commercial jet fuel and kerosene, distillate fuel oil, residual fuel oil, and lubricating oils. The American Petroleum Institute (API) lists over a thousand distinct refinery products. The American Petroleum Institute Standard 610, "Centrifugal Pumps

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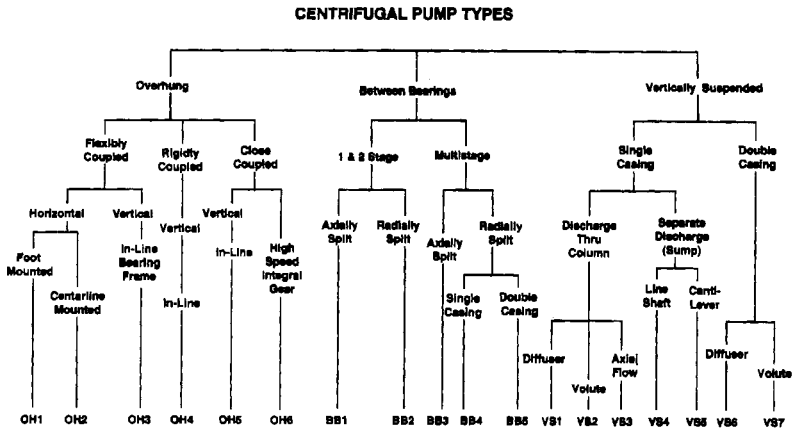


FIGURE 1 Pump classification type identification (Courtesy of the American Petroleum Institute[1])

for Petroleum, Petrochemical and Natural Gas Industries (API 610), has established specifications for the design features required for centrifugal pumps used for general refinery service.

API 610 first edition was published in October of 1954. In its initial form, the document contained only 15 pages of text and applied to single stage overhung pumps only. Since the original publication, the standard has been updated on a regular basis, approximately every five to six years. The tenth edition (2004), contains 89 pages of text, 91 pages of annexes, and covers the specification of as many as 18 different pump types. The eighth edition (1995) showed, for the first time, a "family tree" (Figure 1) of pump types that are potentially covered by the standard and a document organization that makes it clear which requirements apply to what pumps.

The tenth edition of API 610 is technically identical to the ninth edition other than editorial and formatting changes. These latter changes were implemented to allow the standard to be published as ISO 13709, making it for the first time a true, globally accepted international standard.

Construction Figure 2 illustrates the details of a single-stage overhung refinery process pump meeting API 610 requirements. Such a pump is referred to as a Type OH2. The suction nozzle may be located either at the end or at the top. A spacer coupling with a minimum 5 inch long spacer is used between the pump shaft and the driver shaft so the bearing bracket and cover may be removed without disconnecting the suction or discharge piping. This is referred to as a back pull-out design. Figure 3 illustrates the details of a between bearings, two-stage, radially split pump of type BB2. Type BB2 pumps can have one or two stages. Figures 4 and 5 show horizontal multistage pumps. Figure 4 is an axially split machine, Type BB3, and Figure 5 is a radially split or barrel-type pump, Type BB5. It is a requirement of all present day horizontal refinery pumps that the mechanical seal can be changed and the rotor removed without disconnecting the piping or moving the driver.

API 610 requires standardized seal chamber dimensions for all pumps. For overhung and vertically-suspended designs, there is only one seal chamber and seal to prevent leakage to the atmosphere. For between bearings designs, there are two seal chambers and two seals. In single-stage overhung pumps, the seal chamber pressure during pump operation is between suction and discharge pressure, depending on the design of the wear rings and impeller balance holes. Between-bearings pumps of two or more stages usually have a breakdown bushing with bleed-off to suction at the high-pressure end to balance the pressure at the shaft seals. Vertically suspended pumps are usually also equipped with a leak-off arrangement to reduce seal chamber pressure from discharge pressure.

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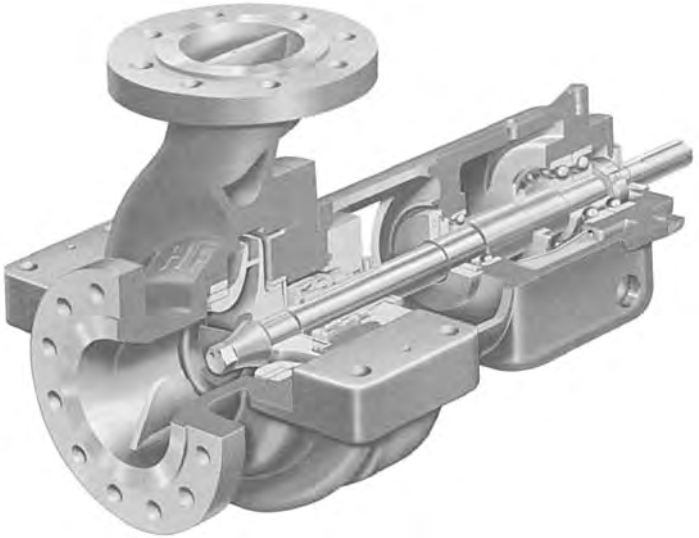


FIGURE 2 Single-stage overhung process pump—API Type OH2 (Flowserve Corporation)

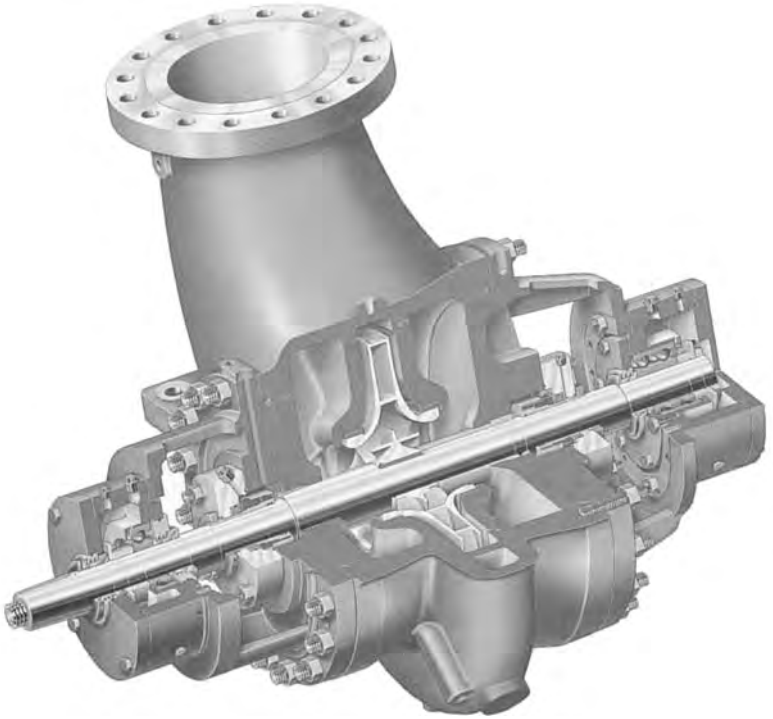


FIGURE 3 Between bearings, single-stage, double suction, radially-split process pump—API 610 Type BB2 (Flowserve Corporation)

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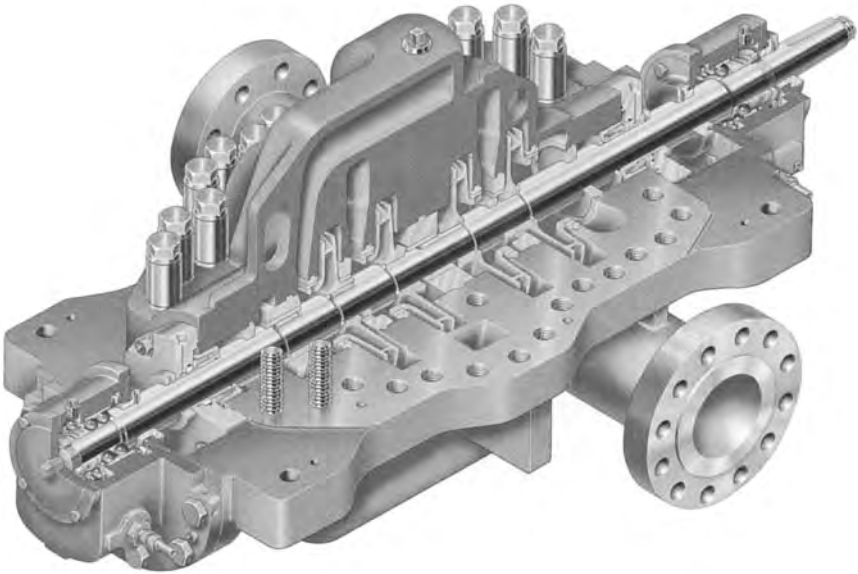


FIGURE 4 Axially split multistage pump API 610 Type BB3 (Flowserve Corporation)

In API 610 ninth edition, mechanical seal piping and lubricating systems (when separate from the pump itself) were removed from the standard. The mechanical seal requirements were moved into API 682, "Shaft Sealing Systems for Centrifugal and Rotary Pumps" and the piping and lubrication system requirements were moved into API 614 "Lubrication, Shaft Sealing, and Control-Oil Systems and Auxiliaries for Petroleum, Chemical, and Gas Industry Services". These API standards have been adopted by the International Standards Organization (ISO), API 682 as ISO 21049, and API 614 as ISO 10438.

Although most refiners continue to strongly prefer single seals for cost and reliability reasons, environmental requirements increasingly drive them toward dual seals. From the mid 90s to the present approximately 40% of refinery pumps in hydrocarbon services have been specified to have dual seals. In the last few years this percentage has climbed to the point that a new refinery project in detailed design has specified more than 90% of hydrocarbon pumps to have dual seals.

Regardless of whether the seal is a single or dual, for each seal, one of the sealing rings is made of carbon, and the mating ring is either silicon carbide or tungsten carbide. API 610/ISO 13709 requires that seals and their support systems comply with API 682/ISO 21049. API 682/ISO 21049 gives standardized auxiliary piping plans for the support of all liquid-type mechanical seals. Dual mechanical seals have proved progressively more reliable while the seals and their support systems have become progressively more expensive. It is not unusual for the mechanical seal and its support system in a single-stage overhung pump to cost more than the bare pump, driver, and baseplate.

The thrust bearing for single-stage overhung pumps is subject to axial loading caused by exposure of one end of the shaft to suction pressure and forces on the impeller due to the differential between suction pressure and discharge pressure. When single-stage overhung pumps are used for suction pressures in excess of 250 psig (17 MPa), it is normal practice to "unbalance" the impeller wear rings. This is done by making the diameter of the back wear rings smaller than that of the suction side wear rings, or by eliminating the back wear rings completely. This produces an unbalanced axial thrust load in the opposite direction to that created by high suction pressure and, thus, decreases the net

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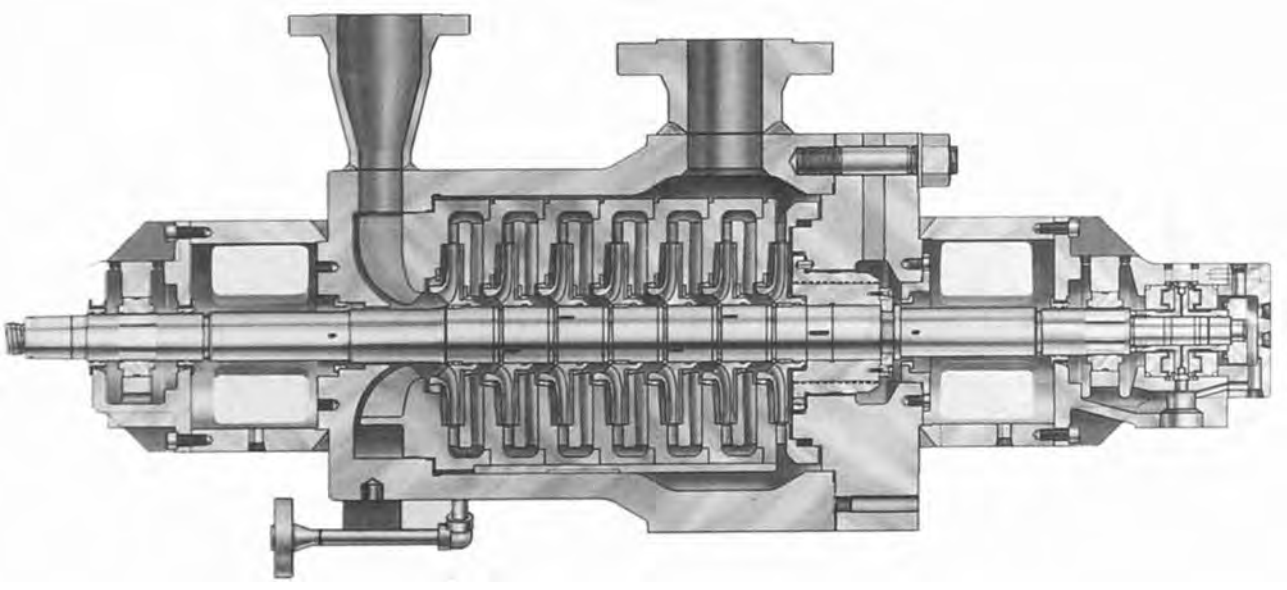


FIGURE 5 Radially split or barrel type pump—API 610 Type BB5 (Flowserve Corporation)

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axial load on the thrust bearing. In the case of multistage pumps, the differential pressure across the impellers builds with each stage and some method must be employed to reduce the load on the thrust bearing. In some designs, axial thrust is balanced by means of a balance piston/leak-off arrangement; in others, an opposed impeller arrangement helps to balance axial loads. In all cases, to meet API 610 requirements, the bearings must be designed for a minimum life of 25,000 hours of continuous operation at rated conditions and 16,000 hours at maximum radial and axial loads, at rated speed.

All single-stage overhung pumps and most one and two stage between bearings pumps are designed with "families" of bearing brackets/housings. This results in the bearings being selected for some "worst case" or "highest load" set of hydraulics. Fatigue bearing life is rarely a concern in API 610 pumps and bearing life is generally a function of lubricant quality and operational effectiveness of the owner.

It is well recognized in the oil and gas industry that for expensive sealing systems to deliver very low fugitive emissions and high reliability, radial movement at the seal faces must be held to a minimum. API 610 addresses this issue with a number of requirements. Among these are:

- Shaft deflection at the seal faces is limited to 0.002 in under the worst conditions.
- One and two stage pumps are required to have a dry bending first critical above 120% of maximum continuous speed.
- Seal chamber dimensions are standardized.
- Bearings must be designed for a minimum life of 25,000 hours at rated conditions, 16,000 hours at maximum radial and axial loads and rated speed.

The shaft deflection criteria, the dry critical criteria and the bearing minimum life criteria all drive shaft stiffness up. The seal chamber dimensions set the minimum overhang, the pump can have and limit how stiff the shaft can be. Note that Hydraulic Institute Standard 1.3 2000 Section 1.3.4.7.2 provides standard calculations to check shaft deflection and dry bending critical speed as well as bearing loads. With bearing loads known the several bearing manufacturers provide very simple tools for calculating bearing life on their websites.

Having noted the importance of minimizing shaft deflection at the seal faces, API 610 has no requirement for pumps to be designed with double volutes. However, commercial interests of pump manufacturers make it almost universal that single-stage overhung pumps have double volutes over a certain size, typically a 3-in discharge and, for larger pumps, even some 3-in discharge sizes are available with a double volute. The most significant exceptions are the smaller pumps, where it is simply not cost effective to cast a double volute.

To aid in maintaining alignment at various operating temperatures, pump mounting feet are located on the case at the same centerline as the shaft for all horizontal pumps. Earlier editions of API 610 also suggested using bearing or seal chamber cooling at elevated pumping temperatures. However, the industry has moved toward less and less cooling water usage because of cost and availability of quality water sources in many areas. Currently, API 610 leaves cooling as an item to be jointly agreed upon by the purchaser and the pump vendor. It should also be noted that the quality and stability of lubricating oils has significantly improved over the last few decades, and oils simply perform better and more adequately without cooling than before.

In general, pumps meeting API 610 are designed for operation at elevated temperatures. In all services, but particularly in high temperature services, pumps are subject to loads at the nozzles due to thermal movements of piping, fabrication errors in the piping, and movements of the piping supports. These forces and moments tend to "move" the pump relative to its driver and result in misalignment of the pump and driver shafts. This misalignment can increase vibration levels and impose forces on the shafting that contribute to poor reliability. API 610 prescribes minimum forces and moments, by nozzle size, that complying pumps must be able to withstand. Further, it outlines a test procedure to allow verification that a pump meets these minimum nozzle load requirements. The coordinate system for orientation of nozzle loads is in accordance with ISO 1503 standard convention.

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Inline pumps have been developed, in part, to avoid the problem of misalignment due to nozzle loads. Figure 6 shows a single stage, overhung, inline pump with a separate bearing frame, Type OH3. Figure 7 shows a high-speed integrally geared inline pump, Type OH6. For a given flange size, inline pumps are required to withstand twice the magnitude of nozzle loads allowed for horizontal pumps. In addition, installation is simplified and less expensive because a block foundation is not required, and the pump mounts in a pipeline



FIGURE 6 Single-stage, overhung, vertical inline pump with separate bearing frame—API 610 Type OH3 (Flowserve Corporation)

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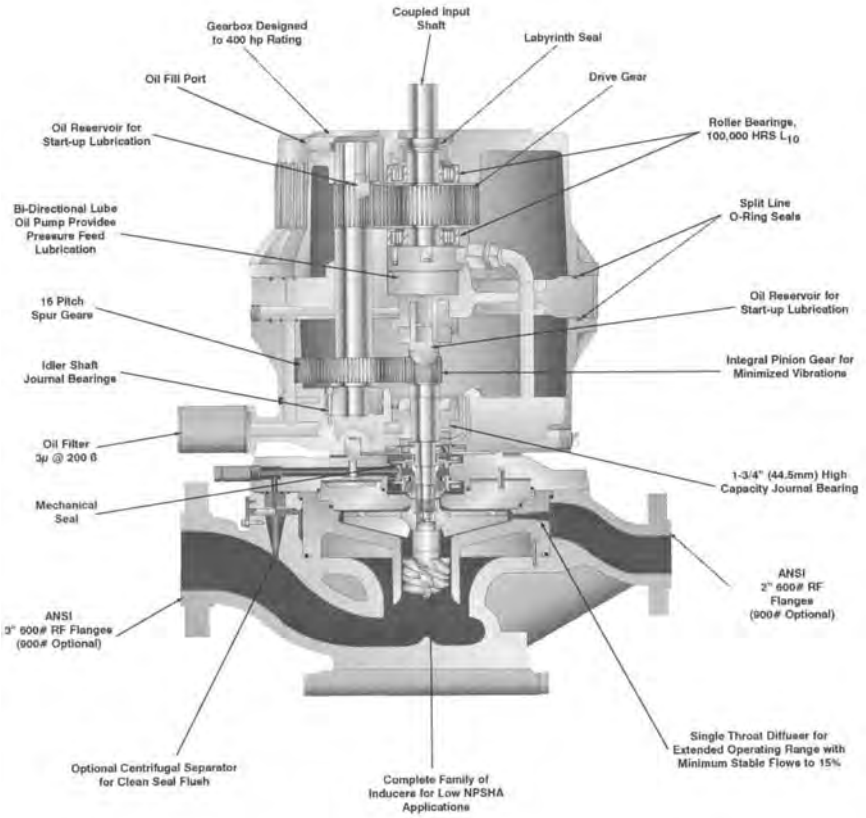


FIGURE 7 Integrally geared inline pump—API 610 Type OH6 (Sundyne Corporation)

similar to a valve. It is noted that an in-line pump is the equivalent of a very heavy valve, and most users supply a support pad or other support arrangement beneath the pump. Furthermore, the vertical arrangement causes the pump to take up much less space. The Type OH6 pump (Figure 7) has a gear box that increases the speed at which the impeller spins and typically produces high heads at relatively low flows. This type of design has many advantages in certain low flow, high head services. Some alternative inline pump designs utilize high-speed motors that eliminate the need for the gearbox. High-speed pumps may also be supplied with inducers in front of the typically radially bladed impellers to improve suction (NPSH) performance. When this is done, the range of operation of the pump may be restricted to avoid off-design flow instability. This should be recognized and the required range of pump operation should be addressed during the applications stage of pump selection for such pumps.

Pumps of overhung shaft construction are nominally limited by most manufacturers to drivers rated below approximately 500 HP (375 kW). Units with greater power requirements are usually designed with bearings on both ends of the shaft and the impeller—or impellers—in between the bearings (designated by API 610 as between bearings, or Type BB, pumps). Ball bearing construction, in compliance with requirements of API 610, is used to a limit of a bearing Nd_m factor of 500,000. The Nd_m factor is the product of the pump operating speed (N), in revolutions per minute and the mean bearing diameter (d_m), equal to the bearing bore plus the bearing outside diameter, divided by 2 (all dimensions in millimeters). For values of Nd_m above 500,000, or where the bearing basic life rating (L_{10h}) does not meet the minimum life requirements mentioned earlier, hydrodynamic

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bearings are required. This may result in a pump with sleeve radial and rolling element thrust bearings, or with both hydrodynamic radial and thrust bearings. API 610 also establishes an energy density limit of four million, above which hydrodynamic radial and thrust bearings are always required.

To ensure safety and reliability, it is an API 610 requirement that pump pressure casings and flanges be designed for maximum pump discharge pressure plus allowances for head increases at the rated pumping temperatures. API 610 also establishes minimum pressure ratings for all covered pumps. For horizontal radially split pumps, the minimum pump pressure rating is that of an ANSI/ASME B16.5 Class 300 (ISO 7005-1 PN50) steel flange of the material grade corresponding to that of the casing, or 600 psig (4 MPa), whichever is less. Furthermore, all pumps are subjected to a hydrostatic test at 1.5 times the case rating. It is a requirement for one- and two-stage pumps that the suction and discharge flanges have the same pressure rating and that the casing have a minimum pressure rating equivalent to the flange rating.

Performance One- and two-stage centrifugal pumps for API services cover a wide range of operating conditions—up to 1200 ft (365 m) of head and 11,000 gpm (2500 m^3/h), varying with manufacturer. Vertical in-line pumps have a range similar to that of their horizontal single-stage counterparts. Multistage centrifugal pumps are available for both large flow and extremely high heads, probably exceeding anything we could state here by the time this handbook is published.

API 610 requires a performance test in accordance with Hydraulic Institute (HI) standards for all pumps. The only deviation from HI standards is that efficiency is calculated for information and is not a rating or guarantee value. Power is the guarantee quantity and includes all losses (such as for mechanical seals and bearings). Vibration levels are taken during the test and acceptance criteria are established for both bearing housing and shaft measurements on the pump manufacturer's test stand. In general, refiners have lowered vibration levels in their plants through conscientious maintenance practices, particularly in the areas of balance and alignment. The vibration acceptance criteria are based upon approximately 150% of the average vibration of spared pumps in one of the larger refineries in North America in the early 1990s. Therefore, the vibration acceptance criteria are somewhat more liberal than typical plant values because of the temporary nature of a manufacturer's performance test setup. If critical service suction conditions warrant it, an NPSH test may also be conducted.

Materials Refinery pumps handle a variety of products with specific gravities ranging from 0.3 to 1.3 and viscosities from values below that of water to as high as 15,000 SSU (3240 centistokes) for centrifugal pumps and even higher for rotary pumps, over a wide range of temperatures. The product may be as innocuous as a lubricating-type oil or extremely corrosive. Many materials are utilized to satisfy these varied product requirements; the most common material classes are given in Table 1 (API 610 Table G-1: Material Class Selection Guide). Material classes and specifications are given in Table 2 (API 610 Table H-1: Materials for Pump Parts).

Drivers Drivers for refinery pumps are usually either electric motors or steam turbines. Centrifugal pumps running backward as hydraulic turbines are also used. The vast majority of refinery pumps are run at a fixed, 2-pole speed of either 3600 rpm in 60 cycle locations or 3000 rpm in 50 cycle locations.

PIPELINE PUMPS

Centrifugal pumps are used on every major liquid pipeline in the world to transport a variety of fluids, including crude oil, motor gasoline, fuel oil, jet fuel, liquefied petroleum gases, and anhydrous ammonia. Pump efficiency is of primary importance because of the power required to transport the liquid. Most pipeline systems install pumps in series because the differential head requirement is primarily energy loss due to friction, and an

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TABLE 1 Material class selection guide (Courtesy of the American Petroleum Institute)[1]

Service	Temperature range °C	Temperature range (°F)	Pressure range	Materials class	Ref. note
Fresh water, condensate, cooling tower water	<100	<212	All	I-1 to I-2	
Boiling water and process water	< 120	<250	All	I-1 or I–2	a
	120 to 175	250 to 350	All	S-5	a
	>175	>350	All	S-6, C-6	a
Boiler feed water					
Axially split	>95	>200	All	C-6	
Double-casing (barrel)	>95	>200	All	S-6	
Boiler circulator	>95	>200	All	C-6	
Foul water, reflux drum water, water draw, and hydrocarbons containing these waters, including reflux streams	<175	<350	All	S-3 or S-6	b
	>175	>350	All	C-6	
Propane, butane, liquefied petroleum gas, ammonia, ethylene, low-temperature services (minimum metal temperature)	230	<450	All	S-1	
	> −46	> −50	All	S-1(LCB)	h
	> −73	> −100	All	S-1(LC2)	h
	> −100	> −150	All	S-1(LC3)	h, i
	> −196	> −320	All	A-7 or A-8	h, i
Diesel oil; gasoline; naphtha; kerosene; gas oils; light, medium, and heavy lubricating oils; fuel oil; residuum; crude oil; asphalt; synthetic crude bottoms	<230	<450	All	S-1	
	230 to 370	450 to 700	All	S-6	b, c
	>370	>700	All	C-6	b
Noncorrosive hydrocarbons, e.g., catalytic reformate, isomaxate, desulfurized oils	230 to 370	450 to 700	All	S-4	c
Xylene, toluene, acetone, benzene, furfural, MEK, cumene	<230	<450	All	S-1	
Sodium carbonate	<175	<350	All	I-1	
Caustic (sodium hydroxide), concentration <20%	<100	<212	All	S-1	d
	>100	>200	All	–	e
Seawater	<95	<200	All	–	f
Sour water	<260	<470	All	D-1	
Produced water, formation water, and brine	All	All	All	D-1 or D-2	f

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Sulfur (liquid state)	All	All	All	S-1	
FCC slurry	<370	<700	All	C-6	
Potassium carbonate	<175	<350	All	C-6	
	<370	<700	All	A-8	
MEA, DEA, TEA stock solutions	<120	<250	All	S-1	
DEA, TEA-lean solutions	< 120	<250	All	S-1 or S-8	d,g
MEA-lean solution (CO_2 only)	80 to 150	175 to 300	All	S-9	d
MEA-lean solution (CO_2 and H_2S)	80 to 150	175 to 300	All	S-8	d,g
MEA-, DEA-, TEA-, rich solutions	< 80	175	All	S-1 or S-8	d
Sulfuric acid concentration > 85%	<38	<100	All	S-1	b
85% to < 1%	<230	<450	All	A-8	b
Hydrofluoric acid concentration > 96%	<38	<100	All	S-9	b

The materials for pump parts for each material class are given in Annexe H.

Specific materials recommendations should be obtained for services not clearly identified by the service descriptions listed in this table.

Cast-iron casings, where recommended for chemical services, are for nonhazardous locations only. Steel casings (5.12.1.6) should be used for pumps in services located near process plants or in any location where released vapor from a failure could create a hazardous situation or where pumps could be subjected to hydraulic shock, for example, in loading services.

[a]Oxygen content and buffering of water should be considered in material selection.

[b]The corrosiveness of foul waters, hydrocarbons over 230°C (450°F), acids, and acid sludges may vary widely. Material recommendations should be obtained for each service. The material class indicated above is satisfactory for many of these services, but shall be verified. S-8 materials may also be considered for operating temperatures below 95°C (200°F).

[c]If product corrosivity is low, Class S-4 materials may be used for services at 231°C to 370°C (451°F to 700°F). Specific material recommendations should be obtained in each instance.

[d]All welds shall be stress-relieved.

[e]UNS N08007 of Ni-Cu alloy pump material should be used.

[f]For seawater, produced water, formation water, and brine services, the purchaser and the vendor should agree on the construction materials that best suit the intended use.

[g]The vendor shall consider the effects of differential material expansion between casing and rotor and confirm suitability if operating temperatures are to exceed 95°C (200°F).

[h]Materials selected for low-temperature services shall meet the requirements of 5.12.4 and 5.12.1.6. Casting alloy ASTM A 352, Grades LCB, LC2, & LC3 is shown only for reference. Use equivalent materials for wrought alloys.

[i]Material alloys based on aluminium, bronze, aluminium bronze, and nickel, may also be considered for temperatures as low as −196°C (−320°F).

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TABLE 2 Materials for pump parts (Courtesy of the American Petroleum Institute)[1]

PART	Full compliance materials[b]	Material classes and abbreviations												
		I-1	I-2	S-1	S-3	S-4	S-5	S-6	S-8[l]	S-9[l]	C-6	A-7	A-8	D-1[j]	D-2[j]
		CI[a]	CI	STL	STL	STL	STL	STL	STL	STL	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
		CI	BRZ	CI	NI-RESIST	STL	STL 12 % CHR	12 % CHR	316 AUS	Ni-Cu Alloy	12 % CHR	AUS[c, d]	316 AUS[d]	Duplex	Super Duplex
Pressure casing	Yes	Cast iron	Cast iron	Carbon steel	Carbon steel	Carbon steel	Carbon Steel	Carbon Steel	Carbon Steel	Carbon Steel	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
Inner case parts: (bowls, diffusers, diaphragms)	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	Carbon Steel	12 % CHR	316 AUS	Ni-Cu Alloy	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
Impeller	Yes	Cast iron	Bronze	Cast Iron	Ni-Resist	Carbon steel	Carbon Steel	12 % CHR	316 AUS	Ni-Cu Alloy	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
Case wear rings[k]	No	Cast iron	Bronze	Cast Iron	Ni-Resist	Cast iron	12 % CHR Hardened	12 % CHR Hardened	Hard-faced 316AUS[e]	Ni-Cu Alloy	12 % CHR Hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced Duplex[e]	Hard-faced Super Duplex[e]
Impeller wear rings[k]	No	Cast iron	Bronze	Cast Iron	Ni-Resist	Cast iron	12 % CHR Hardened	12 % CHR Hardened	Hard-faced 316AUS[e]	Ni-Cu Alloy	12 % CHR Hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced Duplex[e]	Hard-faced Super Duplex[e]
Shaft[d]	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	AISI 4140	AISI 4140[f]	316 AUS	Ni-Cu Alloy	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
Throat bushings[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12 % CHR Hardened	12 % CHR Hardened	316 AUS	Ni-Cu Alloy	12 % CHR Hardened	AUS	316 AUS	Duplex	Super Duplex
Interstage sleeves[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12 % CHR Hardened	12 % CHR Hardened	Hard-faced 316AUS[e]	Ni-Cu Alloy	12 % CHR Hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced Duplex[e]	Hard-faced Super Duplex[e]
Interstage bushings[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12 % CHR Hardened	12 % CHR Hardened	Hard-faced 316AUS[e]	Ni-Cu Alloy	12 % CHR Hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced Duplex[e]	Hard-faced Super Duplex[e]
Case and gland studs	Yes	Carbon steel	Carbon steel	AISI 4140 steel	AISI 4140 Steel	AISI 4140 Steel	AISI 4140 Steel	AISI 4140 Steel	AISI 4140 Steel	Ni-Cu Alloy Hardened[i]	AISI 4140 Steel	AISI 4140 Steel	AISI 4140 Steel	Duplex[i]	Super Duplex[i]
Case gasket	No	AUS, Spiral wound[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	316 AUS Spiral wound[g]	Ni-Cu Alloy, Spiral wound, PTFE filled[g]	AUS, Spiral wound[g]	AUS, Spiral wound[g]	316 AUS Spiral wound[g]	Duplex SS Spiral wound[g]	Duplex SS Spiral Wound[g]

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Discharge head/ suction can	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel [i]	AUS	AUS	316 AUS	Duplex	Super Duplex
Column / bowl shaft bushings	No	Nitrile butadiene [h]	Bronze	Filled carbon	Nitrile butadiene [h]	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon
Wetted fasteners (bolts)	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	316 AUS	316 AUS	316 AUS	Ni-Cu Alloy	316 AUS	316 AUS	316 AUS	Duplex	Super Duplex

[a] The abbreviations in the upper part of the second row indicate the case material; the abbreviations in the lower part of the second row indicate trim material. Abbreviations are as follows: BRZ = bronze, STL = steel, 12 %, CHR = 12 % chromium, AUS = austenitic stainless steel, CI = cast iron, 316 AUS = Type 316 austenitic stainless steel

[b] See 5.12.1.4

[c] Austenitic stainless steels include ISO Types 683-13-10/19 (AISI Standard Types 302, 303, 304, 316, 321, and 347).

[d] For vertically suspended pumps with shafts exposed to liquid and running in bushings, the standard shaft material is 12 % chrome, except for Classes S-9, A7, A-8, and D-1. The standard shaft material for cantilever pumps (Type VS5) is AISI 4140 where the service liquid allows (see Annex G, Table G.1).

[e] Unless otherwise specified, the need for hard-facing and the specific hard-facing material for each application is determined by the vendor and described in the proposal. Alternatives to hard-facing may include opening running clearances (5.7.4) or the use of non-galling materials or non-metallic materials, depending on the corrosiveness of the pumped liquid.

[f] For Class S-6, the standard shaft material for boiler feed service and for liquid temperatures above 175 °C (350 °F) is 12 % chrome (see Annex G, Table G.1).

[g] If pumps with axially split casings are furnished, a sheet gasket suitable for the service is acceptable. Spiral-wound gaskets should contain a filler material suitable for the service. Gaskets other than spiral wound, may be proposed and furnished if proven suitable for service and specifically approved by the purchaser.

[h] Alternative materials may be substituted for liquid temperatures greater than 45 °C (110 °F) or for other special services.

[i] Unless otherwise specified, AISI 4140 steel may be used for non-wetted case and gland studs.

[j] Some applications may require alloy grades higher than the Duplex materials given in Table H.2.

"Super Duplex" material grades with pitting resistance equivalency (PRE) values greater than 40 may be necessary.

PRE $\geqslant$ 40, where PRE is based on actual chemical analysis.

PRE = $\%Cr_{free}$ + (3,3 × % molybdenum) + (2 × % copper) + (2 × % tungsten) + (16 × % nitrogen)

= [(% chromium - (14,5 × % carbon)] + (3,3 × % molybdenum) + (2 × % copper) + (2 × % tungsten) + (16 × % nitrogen)

Note that alternative materials such as "super austenitic" may also be considered.

[k] Non-metallic wear part materials, proven to be compatible with the specified process fluid, may be proposed within the applicable limits shown in Table H.4. Also see 5.7.4 c).

[l] The vendor shall consider the effects of differential material expansion between casing and rotor and confirm suitability if operating temperatures are to exceed 95 °C (200 °F).

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outage of one of the units would result in only a partial loss in the throughput capacity. For pipelines where the differential head is mostly static, such as in mountainous areas, pumps are often installed in parallel. A series installation arrangement in a static system would be unsuitable because the differential head required could not be obtained unless all of the units were operating. In certain cases, however, where differential head requirements exceed levels practical for a single multistage pump, a combination of series and parallel arrangements may be desirable.

Construction A single-stage double-suction pump, a two-stage double-suction pump, and a four-stage pipeline pump are shown in Figures 8, 9, and 10, respectively. All the units are double volute and are of axially split construction. The single-stage double-suction unit (Figure 8) is typical of those used in series operation on large diameter pipelines with capacities as high as two million barrels per day. One pipeline alone has installed more than one hundred 5000 HP (3730 kW) pumps of this construction. The two-stage double-suction pump (a smaller unit is shown in Figure 9 with an engine driver) is typical for higher head applications, and sizes are available for power ratings to 20,000 HP (15,000 kW). For lateral and smaller diameter pipeline pumping stations, multistage units (Figure 10) are used. The number of stages chosen corresponds to the differential head required. Multistage units may be destaged initially to produce lower heads, with a subsequent power saving, or the nozzling may be arranged for series or parallel operation of a portion of the stages. The exact arrangement depends on the system characteristics and initial and ultimate capacities to be pumped.

All units are equipped with sleeve radial bearings and either ball or hydrodynamic thrust bearings, as dictated by power (size) and rotating speed. The single- and two-stage double suction pumps are inherently balanced axially. The multistage units utilize opposed impeller design to obtain axial balance. All modern units are equipped with mechanical seals in the seal chamber. Because most pipeline stations are now designed for unattended outdoor service, a number of safeguard controls are used. These include warnings for low suction pressure, high discharge pressure, high bearing or case temperature vibration, and excessive seal leakage. Tank farms utilize single- and double-suction inline vertical pumps or, if the tank is to be pumped dry, vertical canned pumps.

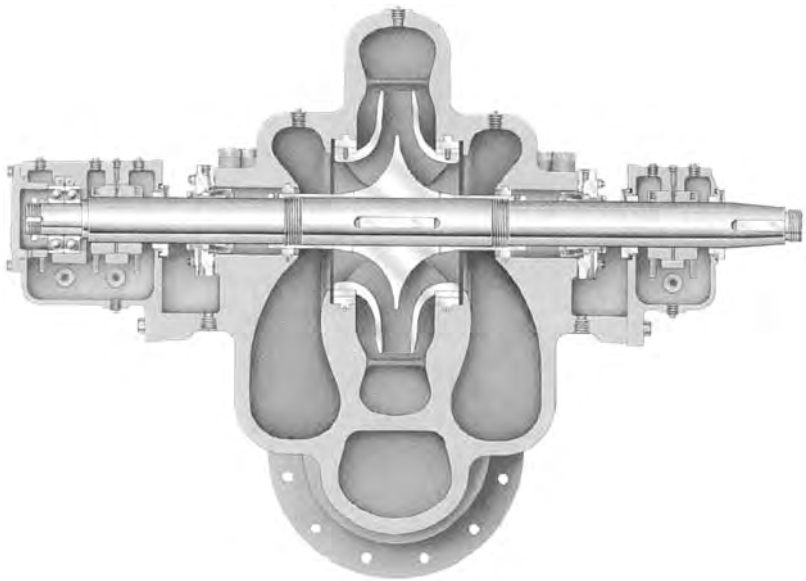


FIGURE 8 Single-stage double suction, axially split pump (Flowserve Corporation)

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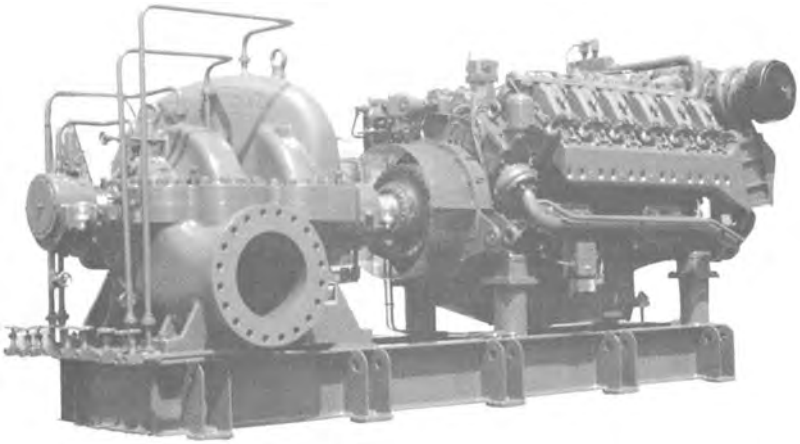


FIGURE 9 Two-stage, double suction pump with engine drive (Flowserve Corporation)

Performance The typical available performance range for single- and two-stage pipeline pumps is up to 1200 ft (365 m) of head and 8000 gpm (1800 m^3/h). For multistage units, the range is up to 8000 ft (2,300 m) and 10,000 gpm (2400 m^3/h). Pipeline units are custom designed, and the range is constantly being extended. Electric motors are normally used as drivers. With the utilization of gas turbines, the maximum power employed has increased dramatically in the last 30 years. Another use of gas turbines on existing pipelines is for peak capacity booster station service, in which case pumps are direct-driven at operating speeds as high as 6000 rpm.

Materials Pump part materials commonly specified for crude and product pipeline services are shown in Table 2 under columns S-1 through S-9. For the more aggressive services and liquids, the higher numbered material columns are preferred. All these material columns utilize carbon steel pump casings; the impellers, wear rings, and shaft materials are changed as the material grades increase in alloy and hardness (wear rings) with advancing material column grade number.

Pumping Stations The typical piping of a main line pump station in which the units are arranged in series is shown in Figure 11. As pipeline capacities have increased, one of the major problems has been the pressure loss at each station. For the station shown, three discharge valves (one 16 in (406 mm) ball valve manifolded in parallel with two 24 in (610 mm) control valves) were used. This arrangement limited the calculated pressure drop to that of a length of 36 in (914mm) pipe equal to the distance across the manifold and still allowed the use of control valves of proven size. When line conditions require throttling, the 16 in (406 mm) ball valve is first completely closed. If additional throttling is needed, the two 24 in (610 mm) control valves are closed to produce the required pressure drop. In order to move scrapers or batch separators through each station without interrupting flow through the pumps, signals (PIG SIG), hydraulically operated 24-in (610 mm) valves, and sequence control wiring are used. The distance between PIG SIG 1 and PIG SIG 2 represents the volume of station loop to be displaced. The tripping of PIG SIG 3 and PIG SIG 4 controls the valve opening and closing required to divert flow from the station discharge piping to behind the scraper or batch separator, thus forcing it to leave the station in the same relative position as it entered. Elbows at pump suction are arranged to avoid uneven flow distribution in the inlet of the double suction impeller.

Pipeline pumps are tested in accordance with HI standards. Reduced speed tests may be required because of the power limitations of manufacturing plants and available drivers, and such tests have proven to be extremely accurate indicators of full speed performance.

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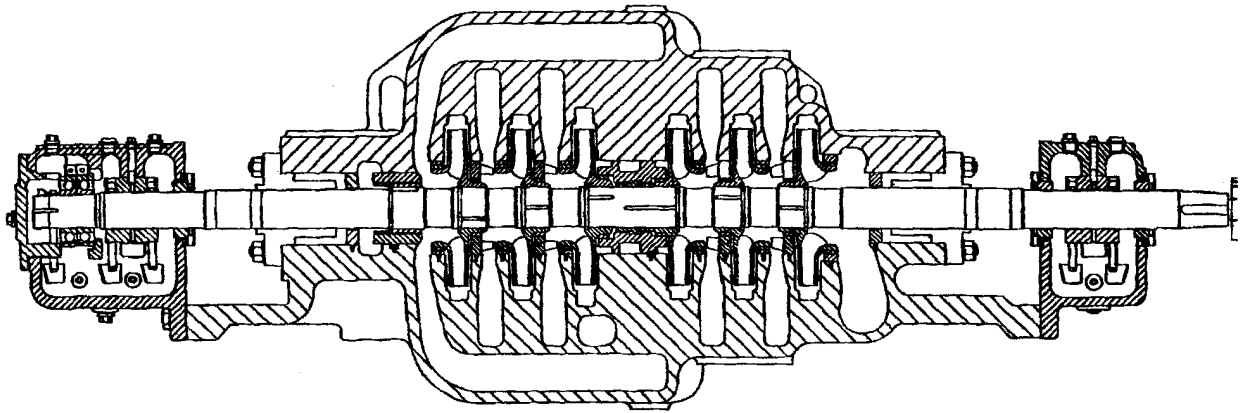


FIGURE 10 Single-suction, double-volute, axially split, six-stage pipeline pump (Flowserve Corporation)

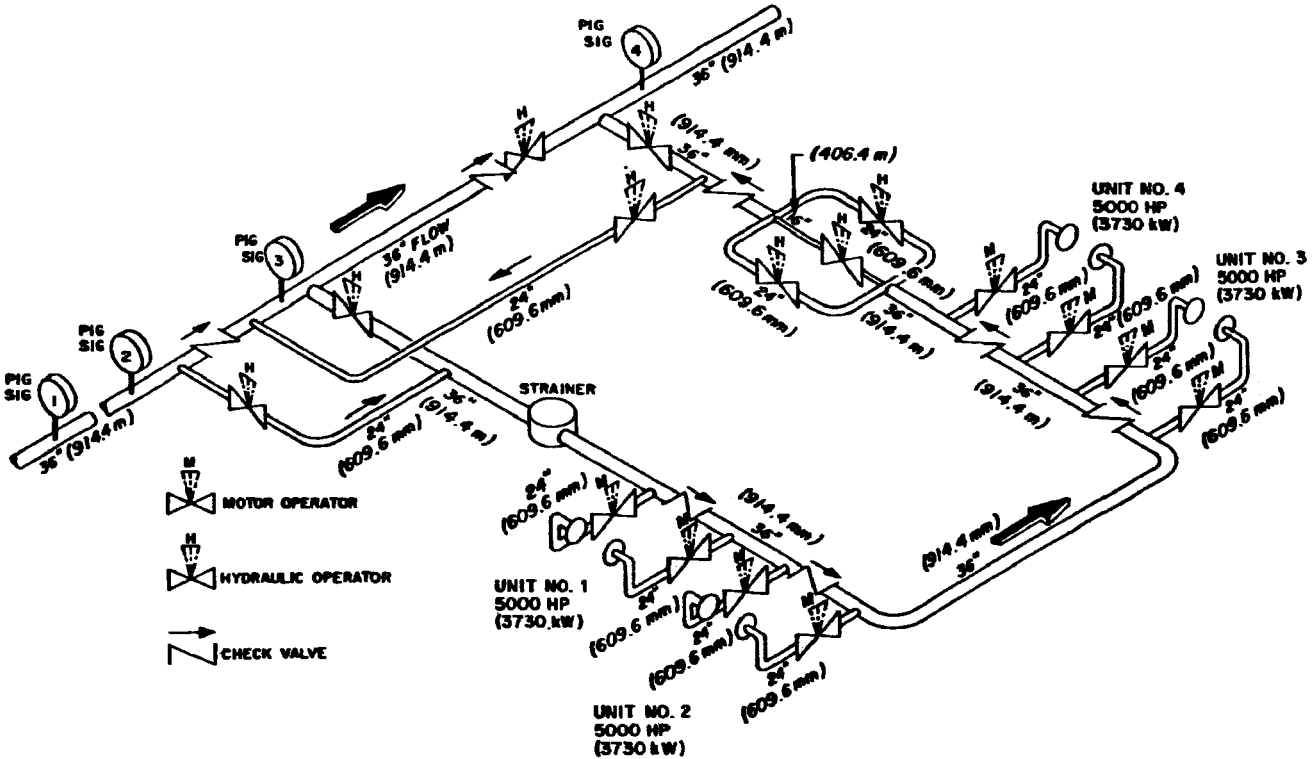


FIGURE 11 Main line pumping station piping layout

SPECIAL SERVICES

Waterflood Centrifugal pumps are also used for water injection service when the capacity required is greater than 10,000 barrels per day. Injection pressures requirements vary considerably from initial startup with operating time. For high-pressure services, usually above 2500 lb/in² (17Mpa), barrel-type pumps are used.

Waterflood, or reinjection, pumps are among the most technically interesting API pumps built in significant quantities today. These pumps push the envelope of what has been accomplished in centrifugal pumps. Barrel-type waterflood pumps (Figure 12) have been build up to 8500 lb/in² (586 MPa) discharge (equivalent to 20,000 ft/6000 m of head), 16,000 gpm (3600 m³/h), with power reaching 36,000 HP (27 mW), at speeds up to 11,000 rpm. The barrel construction is typically of forged steel with an austenitic stainless steel welded overlay applied in areas of high velocity. Barrel-type casings are hydrostatically tested to 1.5 times the rated pressure so to as much as 12,750 lb/in² (863 MPa). At these high pressures, slightly lower test pressures are sometimes negotiated.

Most waterflood injection pressure requirements, however, are such that axially split multistage pumps of the design shown in Figure 4 may be used. As many as 14 stages in one pump are often utilized to obtain the required differential head. While many examples of very reliable pumps with these large numbers of stages can be found, most users are moving away from pumps with more and eight or ten stages. In addition, if still greater pressures are required, the units may be piped in series. Axially split pumps have been hydrostatically tested to 14,000 lb/in² (96 MPa). For such instances, it is becoming more common to hydrostatically test at 1.3 times the maximum allowable working pressure of the pump, following the latest ASME code requirements.

For applications where injection liquid is highly corrosive, materials of construction may include austenitic and duplex stainless steels. Commonly used materials for major pump components in waterflood applications are shown in Table 2 under material columns A-8 and D-1. Seal chambers for high pressure and high-speed services may contain a limited leakage breakdown bushing preceding a mechanical seal for increased reliability.

Reactor Feed Pumps One of the most difficult services encountered in the petroleum industry is the high-temperature, high pressure reactor feed, or charge pump. Type BB5 pumps, such as the one shown in Figure 5, are most commonly used for these services. Reactor feed pumps can be subject to operating temperatures in excess of 800°F (425°C) with discharge pressures up to 3000 lb/in² (20 MPa) gauge. To produce such high pressures requires either a large number of stages or high speed. Historically, refiners have preferred 3600 rpm (3000 rpm in 50 Hz areas) pumps with large numbers of stages. However, the trend more recently is for shorter, more robust pumps, with fewer stages operating at higher speeds. The trend for waterflood pumps is identical.

The high temperatures and corrosiveness of the pumped fluids often require the use of a number of dissimilar metals. Compensation must be made for the different rates of expansion of the various components at operating temperatures. This presents a critical design problem for the manufacturer and a significant operability issue for the refinery. Extreme care is required in the startup and operation of these units. Even so, it is common for refiners to buy only a single pump for such services (no spare), and for these pumps to run continuously for five or more years without maintenance.

LPG Pumps Liquefied petroleum gases (LPG) are pumped from their point of production to points of use. The pump services involved are product pumps (from production into a pipeline), pipeline (located along the pipeline as necessary to maintain the desired flow), and reactor feed when the particular LPG is the feed for a petrochemical process. API 610 type BB3 and BB5 pumps are used, typically BB5 for product and reactor feed, BB3 for pipeline. Rated discharge pressure for product and pipeline pumps today is typically 1800 to 2000 psig (125 to 140 MPa). Reactor feed pumps rated for discharge pressures on the order of 4500 psig (310 MPa) are in service. The difference in pump type with service is determined by location: on plot installations, product, and reactor feed, usually follow API

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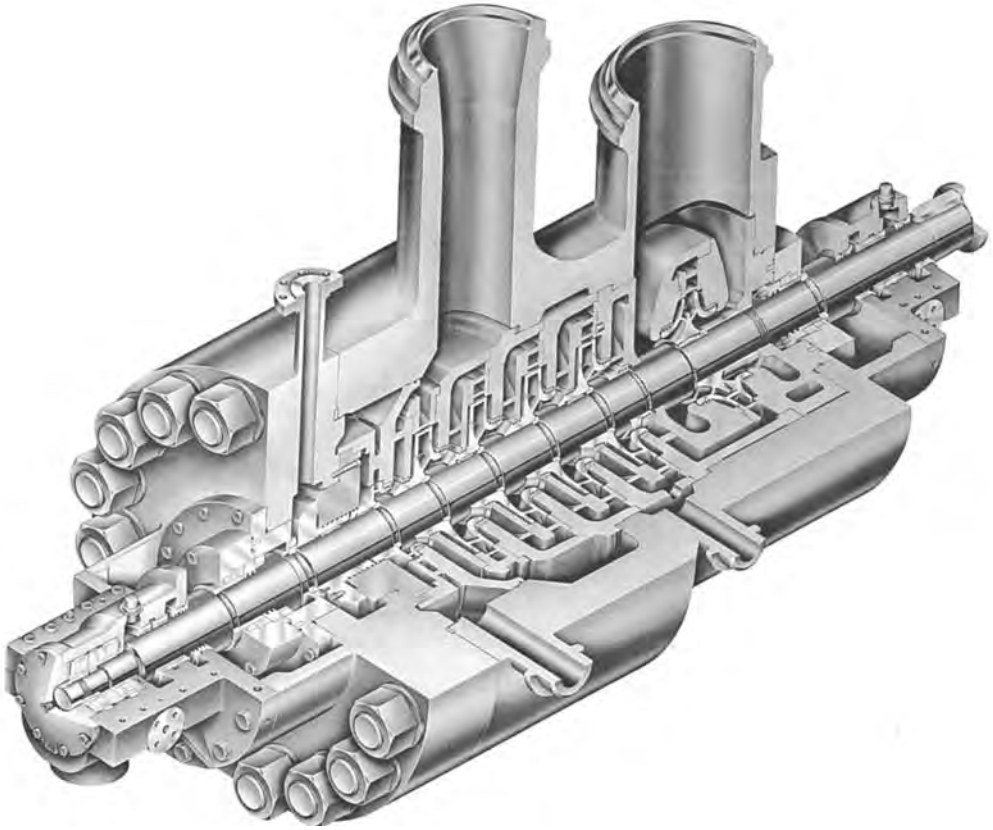


FIGURE 12 Double-casing multistage barrel-type pump with radially split inner casing (Flowserve Corporation)

610's guidelines on casing construction for pressure, SG and temperature quite faithfully, whereas pipeline operators have successfully used BB3 pumps for rated pressures to 2000 psig (140 MPa), SG down to 0.35. Beyond casing construction, there are two other critical aspects of LPG pumps: rotor design and shaft seal type.

Rotor design falls into two classes, slender shaft and large shaft. The former have static deflection greater than the internal running clearances and, therefore, rely on the Lomakin effect for support during operation. Because the rotor rubs in its running clearances during start-up and shutdown, and LPG has poor lubricity, special materials are used for the running clearances to tolerate the rubbing. The alternative is a large-shaft rotor designed to operate between its first and second dry bending critical speeds and have a finite clearance with its running clearances when the pump is shutdown. Provided one surface of the running clearances is serrated to accommodate a "touch," normal pump materials can be used. Large-shaft rotors are the same as those being employed in modern design reactor feed pumps for refining.

Shaft sealing of LPG has been rendered far simpler than it used to be since the development of vaporizing liquid dry gas seals. The usual design is a tandem seal with filtered seal gas upstream of the primary seal, nitrogen sweep through a labyrinth between the secondary and primary seals with primary seal leakage plus portion of the nitrogen to flare. The secondary seal normally runs on nitrogen. Seal gas, nitrogen, and primary seal leakage flows are monitored to alarm on high leakage, shut down the pump on high-high leakage. These seals, correctly engineered into the pump, have demonstrated 5 years operation at sealed pressure on the order of 1000 to 1500 psig (70 to 105 MPa).

PUMPING OF VISCOUS LIQUID

Viscosity of the pumped liquid is a very important factor that must be considered to properly select and size pumps in many petroleum industry applications. A thorough understanding of the relationship between liquid viscosity and pump performance is essential to proper sizing of both the pump and its driver.

Centrifugal pumps are routinely applied on services with liquids having viscosities below 3000 Saybolt Seconds Universal (SSU) or 660 centistokes (cSt), and may be used up to at least 15,000 SSU (3300 cSt). (For background information on viscosity, refer to Section 8.1.) Centrifugal pumps are sensitive, however, to changes in viscosity and will exhibit significant reductions in capacity and head and rather drastic reductions in efficiency at moderate to high values of viscosity. The extent of these effects may be seen in Table 3.

The pump manufacturer nearly always specifies centrifugal pump performance basis pumping clean, cold water, even when the pump has been specifically designed for petroleum industry applications. Cold water is a universal pumping medium. Pump selections, however, must necessarily be made to satisfy viscous conditions of service and require application of these correction factors in the reverse direction.

TABLE 3 Effect of viscosity on performance of a typical centrifugal pump operating at best efficiency point

Viscosity SSU (cSt)	Capacity gpm (m^3/h)	Total head ft (m)	Efficiency %	Brake power bhp (kW)[a]
Nil	3000 (681)	300 (91)	85	241 (180)
500 (110)	3000 (681)	291 (89)	71	279 (208)
2000 (440)	2900 (658)	279 (85)	59	312 (233)
5000 (1100)	2670 (606)	264 (80)	43	373 (278)
10,000 (2200)	2340 (531)	243 (74)	31	417 (311)
15,000 (3300)	2100 (477)	228 (69)	23	473 (353)

[a]All values of brake power based on liquid having a specific gravity of 0.90.

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The Hydraulic Institute Standard 9.6.7 (Reference 5) may be used to calculate viscous effects on pump performance for radial flow centrifugal and vertical pumps with open or closed impellers, handling Newtonian liquids within the pump's normal operating range and within the capacity limits stated in the standard.

DISPLACEMENT PUMPS Both rotary and reciprocating pumps are also frequently used for pumping viscous fluids, and some types are well suited for use where viscosities are beyond the limits for centrifugal pumps. In fact, many common designs are suitable only for use with liquids that are at least moderately viscous because they depend on the viscosity to maintain the lubricating and sealing films between various internal parts of the pumps. Most gear pumps and screw pumps are in this category.

As with centrifugal pumps, the performance of displacement pumps may be significantly affected by changes in liquid viscosity, but the nature of these changes may be quite different. At constant speed, changes in viscosity are likely to have little or no effect on pump capacity. Total head, or differential pressure across the pump, would probably increase with increasing viscosity because of higher system resistance. Thus, the brake power required would increase with even though pump efficiency would not suffer nearly so drastically, if at all, as for a centrifugal pump.

The influence of pump design on how performance varies with viscosity is greater in displacement pumps than in centrifugal pumps. Because there are so many designs, it is not practical to attempt to provide a general means of determining their viscous performance here. This should be done only on the basis of information provided by the pump manufacturer.

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SECTION 12.9
PULP AND PAPER MILLS

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GLOSSARY

accepts Pulp which has passed the screening process and is usable.

additive An additive is any material such as clay, filler, or color added to stock to contribute specific properties.

air dry Pulp which contains 10% moisture.

alkaline pulping Also refers to kraft process pulping or use of caustic; contrasts with the acid sulfite process.

alum Hydrated salts of aluminum sulfate used in papermaking process.

black liquor Liquor containing dissolved organic materials and residual chemicals resulting from alkaline cooking process.

bleaching Bleach is the process of removing the lignin and other color-forming substances from the stock to render it white.

blow Pulp stream discharged from digester when under pressure to atmospheric pressure.

bone dry Also referred to oven dry; contains no moisture.

brightness Indicates degree of whiteness; the reflectivity of a sheet of pulp or paper based on standardized conditions.

brown stock Pulp before bleaching.

cellulose Cellulose is a carbohydrate that is the primary substance in plant fibers. It is extracted from the plant to make paper products.

chest A chest is a vessel for storing pulp.

consistency Consistency is the proportion by mass of fibers in a mixture of fibers and water.

cooking Cooking is the action of the chemical used to break down lignin bond between cellulose fiber in wood and other organic matter.

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deinking Deinking is the process of removing ink particles from recycled paper such as newsprint or magazines.

digester A digester is a pressure vessel used to contain chemical action of cooking chemical and raw cellulose material. Digesters may be of the batch or continuous flow type.

dry end The end of the paper machine where the paper is dried, calendered (imparting a surface finish), and reeled.

evaporator In the recovery of chemicals from the cooking liquor in a chemical pulp mill, an evaporator is used to concentrate the liquor (remove water) so the concentrated liquor can be incinerated in the recovery furnace.

fiber Fibers are the vascular bundles found in plants that are extracted and used to make paper.

fourdrinier Fourdrinier is the continuous wire upon which pulp or paper sheet is produced.

freeness Freeness is a measure of degree of refinement of stock and, hence, of its ability to drain water.

furnish Mixture of raw materials which includes pulp and chemicals used to make a specific grade of paper.

groundwood Groundwood is a mechanical pulp formed by simple grinding to break down wood structure.

headbox A headbox is a specialized nozzle that takes flow from the fan pump and discharges a uniform layer of white water (stock) onto the forming wire of the paper machine.

kraft process The kraft process is a method of separating cellulose fiber from lignins by using caustic soda in presence of sulfur radical.

lignin Lignin is a generic term used to refer to the complex organic matter present in wood that acts as binding agent for cellulose fibers.

liquor *Black liquor:* Solution of water plus residual organic matter (or lignins) in the wood after washing of raw stock. *Green liquor:* Solution of smelt from recovery furnace when dissolved in either water or weak washed liquor. *White liquor:* Solution of caustic soda (or other alkali), sometimes in the presence of a sulfur radical; this is the liquor charged to the digester for cooking.

market pulp Pulp sold as a raw material.

neutral sulfite Neutral sulfite is made through a cooking process using a solution of about 10% sodium sulfite and 5% caustic soda mixed for cooking wood or agricultural fiber to produce high-yield pulp.

oven dry Also referred to as bone dry; contains no moisture.

paper Paper is a sheet of cellulose or other fibers formed on a screen from a mixture of fibers in water. Thin absorbent paper is known as tissue; thicker grades of paper are called paperboard.

pulpwood Logs trimmed to size for use by a pulp mill.

refining Refining is a mechanical process carried out on stock to improve the ability of the fiber to form paper sheet. Different techniques are employed—some designed to shorten the fiber, others to increase the amount of fibrils (or "whisker") on the fiber.

shives Bundles of fibers which have not been separated during the pulping process.

soda process A soda process is similar to kraft process but uses caustic soda without presence of sulfur.

stock Stock is a generic term for the suspension of cellulose fiber in water or chemicals. *Bleached stock:* The same after bleaching. *Brownstock:* The same before bleaching. *Raw stock:* The product discharged from digester(s) before any washing or other treatment. It is also referred to as pulp.

sulfite process A sulfite process is the method of separating cellulose fiber from lignin with acid. This method is becoming obsolete because of its inherent detrimental impact on the environment.

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vat A vat is a semicylindrical mold or container for holding stock during washing or sheet formation.

yield Yield is the percentage of cellulose fiber in the form of pulp produced from a given weight of wood or other raw material. *High yield:* The same when some lignins are allowed to remain in the finished product.

Common Conversion Factors

AD =	Air Dry Stock (Contains 10% water)
OD =	Oven Dry Stock (All water removed) Also called Bone Dry (BD)
AD =	1.11 × OD
OD =	0.90 × AD
ADT/D =	1.11 × ODT/D
ODT/D =	0.90 × ADT/D
AD Consistency =	1.11 × OD Consistency
OD Consistency =	0.90 × AD Consistency
T/D or TPD or ST/D =	Short Tons per Day (Short ton = 2000 lb)
MT/D =	Metric Tons per Day (Metric ton = 2205 lb)
ADST/D =	Air Dry Short T/D (adstpd)
ADMT/D =	Air Dry Metric T/D (ADMtpd)
ADST/D =	1.1025 × Metric T/D
ADST/D =	1.11 × BDST/D
ADST/D =	1.225 × BDMT/D
Production rate normally expressed as ADT/D	
Consistency normally expressed as OD% (or BD%)	

$$\text{Flow Rate (GPM)} = \frac{P \times 15}{C}$$

where

P = Production Rate in ADST/D (adstpd)

C = % Consistency (OD or BD)

$$\text{Flow Rate (l/s)} = \frac{P \times 1.04}{C}$$

where

P = Production Rate in ADMt/d

C = % Consistency (OD or BD)

GENERAL

Apart from the petrochemical industry, there are few continuous-process plants with pumps as much in evidence and with reliability as essential as in the pulp and paper industry.

Before paper leaves the machine room, 100 to 200 tons of water will have been pumped to the mill for every ton of paper produced. These figures represent only the basic amount of water taken from a river and rejected as effluent; the amount of liquid circulated is several times greater.

Although many attempts have been made to utilize a dry process for papermaking, both pulping (the separating of crude fibers from the raw material), and papermaking (actual treatment of the fiber and mechanical formation of the sheet) still require water as the medium to convey fiber. Throughout the mill, pumps are used to transfer or circulate

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fibers suspended in water (stock), chemicals or solids in solution (liquors), or residues and waste matter as slurries, as well as to supply water for general services.

There are at least 150 pumps installed in a modern pulp mill and another 50 or so in a paper mill. About 1000 kW · h is required for the production of one ton of pulp, and a further 500kW · h for the finished paper. Of this total, about 25% is used in pumping. This means that even in a medium-size mill, the installed power required for pumps alone is approximately 10,000 HP (7457 kW), and often it is much higher.

PULPING PROCESS

Raw Materials In the past, the traditional fibrous raw materials for the manufacture of paper were cloth and agricultural residues; today the vast bulk of cellulose pulp is made from wood. Although over half of this is produced from softwoods (long-fibered), an increasing amount is being now produced from hardwoods (short-fibered). Traditional raw materials such as linen, cotton waste, straw, and agricultural residues are still used in small quantities, particularly where the paper sheet requires special properties. These distinctions are important in the selection of pumps, for the liquors have different characteristics. For example, straw black liquor is much more viscous than wood black liquor, and the proper corrections must be made in calculating the pump performance and pipe frictional losses.

Recycle Fiber Many grades of paper include fiber recovered from waste paper—both pre- and post-consumer. Recently, because of environmental concerns, much emphasis has been placed on recycling post-consumer paper products. Specialized plants exist for the recovery and processing of recycled fiber before it is used for papermaking. Some of the different types of recycled fiber are ONP (old newsprint), OCC (old corrugated containers), and MOW (mixed office waste). Special care is needed in the selection of pumps to handle recycled stock because of the large amount of foreign matter present—rope, string, metal, synthetic fibers, and adhesive materials—all of which can cause problems in the process and pumping. Deinking is also widely used in the processing of recycled fibers. Flotation cell deinking, a popular method, uses large amounts of entrained air, which has a dramatic effect on centrifugal pump performance and must be accounted for.

Groundwood Pulp This type of pulp is produced by simply grinding away wood by mechanical action. Almost all of the wood is used in the pulp, including many of the resins and other complex organic compounds. The fibers are bruised so the pulp has inferior strength.

Large amounts of water are required for cooling and for carrying away the groundwood pulp; the latter is usually acidic (pH 4 to 5), and so corrosion-resistant materials must be used. The pulp is used primarily for newsprint and magazines. Depending on the end use, some mechanical treatment (refining) may be required to alter the characteristics of the pulp, particularly the viscosity. In some cases, a mild bleach may be used to improve the color.

Refiner Mechanical Pulp The refiner mechanical pulping process utilizes a disc refiner to reduce wood chips to fibers. It produces a longer fibered pulp than conventional groundwood, but not as long as the chemical pulps. The pulp is therefore somewhat stronger and freer than SGW, but not nearly as strong as kraft pulp. RMP is actually the first of many processes utilizing the disc refiner to produce pulp, The processes vary as to the number of refining stages, refining pressure, temperature, and pre-treatment of the chips (steaming, chemical pre-treatment, and so on). Two of the more popular versions are TMP (thermomechanical pulp), in which chips are pre-softened with steam, and CTMP (chemithermomechanical pulp), in which they receive an additional chemical pretreatment. RMP pulps do not require a rigorous bleaching process in contrast to chemical pulps.

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Chemical Pulping Wood is a complex, nonuniform material containing about 50% by weight cellulose fiber, 30% lignins, and 18 to 20% carbohydrate. The remainder is proteins, resin, and other complex organic compounds that vary from one species to another. Cellulose resists attack from most chemicals, whereas the carbohydrates and other organic materials generally form compounds with the chemical cooking liquor. Some paper products can use the carbohydrate fraction to contribute bulk to the sheet, and for such papers groundwood and RMP and other mechanical pulps are used. Where high strength is required, cooking is necessary to separate the fibers completely from the remainder of the wood.

Most cooking of wood is done in a pressure vessel at high temperature and pressure in the presence of an acid or alkali.

There is considerable tradition in chemical pulping, and a number of different processes are used. For many years the traditional method of producing pulp for high-grade papers was the acid sulfite process. This has been largely superseded in recent years by an alkaline process using sodium-based liquors in the presence of a sulfur radical; this is known as the sulfate or kraft process. The main reasons for the change to the sulfate process have been lower corrosion rates, ease of chemical recovery, and a stronger pulp. The properties of the liquids pumped in the two processes are different, and the pumps require different materials of construction.

Typical Sulfate Process Pulpwood logs are first chipped to about $\frac{3}{4}$ by $\frac{1}{8}$ in (19 by 3 mm) and then charged into either a continuous digester or a series of batch digesters. Digester capacities range from approximately 100 air-dried tons per day to over 2000 air-dried tons per day, necessitating a wide range of hydraulic coverage for digester pumps. Cooking liquor (NaOH plus up to 30% Na_2S) is then allowed to react with the wood chips for 2 to $2\frac{1}{2}$ h at a temperature up to 350°F (177°C) and a pressure in the digester of 80 to 100 lb/in^2 (551 to 689 kPa). In many mills, the heating of the chips and cooking liquor is by direct steam injection to the digester. In others, some form of indirect heating is used with a closed liquor recirculation system. In the latter case, the digester circulating pumps are a critical item because they must handle hot caustic solutions and entrained solid matter in a closed, pressurized circuit. After cooking, the contents of the digester are discharged to atmospheric pressure into a vessel called the blow tank, where the sudden expansion causes the fibers to separate from the liquid, which is now known as black liquor.

At this point, the process splits into two streams—one for fiber processing and the other for chemical recovery. The fiber is washed and screened and then formed into a pulp or paper sheet. The black liquor is washed from the pulp and treated for chemical recovery. Because the most troublesome liquors are to be found in the recovery process and bleach plant, the selection of these pumps is critical for the successful operation of the process.

The chemistry of the recovery process is as follows: After concentration of the black liquor in multiple-effect evaporators to about 50% total solids, the final concentration to 60 to 65% is done by direct contact with hot flue gas from the waste heat or recovery boiler. The 65% concentration black liquor is mixed with salt cake (Na_2SO_4) before being sprayed into the furnace under pressure generated by high-pressure pumps. The furnace atmosphere is maintained with a minimum of excess air so the Na_2SO_4 is reduced to Na_2S, and sodium carbonate (Na_2CO_3) is formed in the process. These molten chemicals run out as a smelt and are dissolved in a tank to form green liquor. This liquor is then causticized with lime to form caustic soda (NaOH), with the Na_2S still present along with other residual chemicals, thus forming the regenerated cooking liquor known as white liquor. The calcium carbonate ($CaCO_3$) formed is burned in a lime kiln for reuse in causticizing. Various lime slurries and residues are formed during this process. The white liquor is then clarified and reused in the digesters, completing the cycle, as shown in Figure 1.

There are a variety of other pulping processes in use, but the sulfate process offers so many advantages that almost all recent installations have been of this type.

Bleaching Bleaching may be considered an extension of the cooking process, the object being to remove the coloring matter, carbohydrate, and lignins to that the remaining pulp

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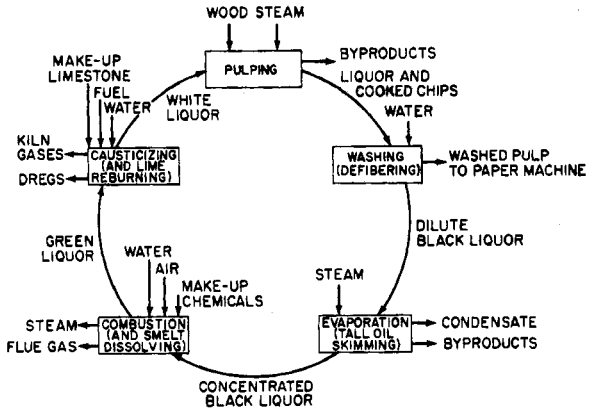


FIGURE 1 The recovery cycle in the sulfate process (Courtesy of ITT/Goulds Pumps, Inc.)

contains a maximum percentage of alpha cellulose, which is the purest cellulose form and the one most resistant to attack from normal chemicals. Bleaching is carried out to reach a degree of reflectance of monochromatic light and called pulp brightness. Because of resistance to attack, special highly reactive chemicals must be used for bleaching. The traditional chemical used for bleaching is chlorine, which is highly selective at attacking the lignin while resisting attack of the cellulose. Chlorine has a tendency, however, to form pollutants in the bleach plant effluent and is therefore being replaced with other chemicals such as chlorine dioxide, hydrogen peroxide, and gaseous oxygen and ozone. Chlorine dioxide produces highly corrosive liquors in the bleach plant.

LIQUIDS PUMPED IN A MILL

There are, broadly, three categories of liquids to be pumped in a paper mill:

1. Water and similar fluids
2. *Liquors* and *slurries*—mainly chemicals and solids in solution or suspension
3. *Stock*—suspension of cellulose fibers in water

Water Apart from the quantities involved, there are no special requirements concerning the water in pulp and paper mills because operating conditions are well within normal limits. However, iron and carbon steel piping should not be used in bleach pulp mills because of iron pickup.

Process water treatment is frequently used to purify process water for the mill and to remove undesirable elements such as iron. Higher-quality water is required for chemical reparation in the bleach plant and for boiler feedwater where demineralizer plants are used. Rubber- or epoxy-resin-lined pumps are used for those components in contact with the demineralized water.

PUMPS FOR MILL WATER For the majority of pumps, standard cast iron or stainless steel fittings are used except as noted for demineralized water. In many mills, however, stainless-steel-fitted pumps are standard because this permits a minimum number of spares to be held in stock for other duties.

In the paper mill, water used to form the sheet on the paper machine has a very low fiber content—$\frac{1}{2}$ to 1% consistency—and is known as white water. Fiber contents this low usually do not cause any pumping problems except in wear ring areas where flashing or slotting is used to keep leakage paths open and free from binding.

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Much of this water is recirculated, and where bleached products are produced, pumps must be constructed an austenitic stainless steel.

Liquors and Slurries Depending on the process and the particular point in that process, the liquor characteristics may require special pumps or special materials. Although the liquor cycle is a difficult one as far as the pumps are concerned, standard designs should be used whenever possible because this reduces the number of different types of pumps in the mill. In some cases, it may be necessary to use a higher material specification than necessary to achieve interchangeability.

Liquor and slurry pumps may be grouped as follows:

Group A—Standard designs suitable for most process uses where corrosion or erosion is not a major factor. Impellers are typically stainless steel with casings of cast iron.

Group B—Standard end-suction designs suitable for corrosive liquors. All liquid end components are typically 316 stainless steel. Duplex stainless steels may be used where erosion may be a factor.

Group C—Standard or nonstandard designs suitable for special services. Pumps are similar to group B for most applications but are of 317 or 317L stainless steel. For most corrosive services, glass-reinforced epoxy, resin, titanium, super austenitic stainless steels are used for both impeller and casing. Mechanical seals in place of packed boxes or dynamic seals are usually fitted to these pumps.

Recommendations for liquor and slurry pumps are

1. All liquor pumps should be classified as slurry type with open nonshrouded impellers of the end-suction and back pull-out type. Simply supported, double suction pumps are also used for fibrous slurries (stock)—particularly $\frac{1}{2}$% to 3% consistency stock on the paper machine. This includes most fan and cleaner pump applications.
2. On group A and B pumps, sealing is accomplished with dynamic seals, mechanical seals, or packed stuffing boxes.
3. For group C pumps, in particular, it may be necessary to depart from a standard design or type of centrifugal pump. For example, if a positive displacement characteristic is required, a screw-type pump may be used with confidence. In addition, all pumps handling stock with consistency above 6% must be regarded as nonstandard types.

After the pumps are grouped, it becomes necessary to decide which pump may be used for specific liquors. Requirements for individual mills will differ in detail, but the following may be taken as an indication of current practice, particularly in modern sulfate (kraft) mills. In every case, manufacturers should be made aware of the liquor characteristics and of the location of the pump in the process.

COOKING LIQUOR (WHITE LIQUOR—SULFATE PROCESS) This is essentially an alkaline solution made by causticizing green liquor. The liquor is prepared at concentrations over the range of 50 to 100 g/liter depending on the wood species, and the amount of active alkali (expressed as Na_2O) may be from 14 to 30% of the dry wood weight. White liquor is mainly sodium hydroxide, with a small percentage of sodium sulfide which depends on the mill sulfidity. Higher values of active chemical are used in bleached pulp mills. The term *sulfidity* is used to denote the ratio of chemicals present; it is frequently expressed as Na_2O and calculated from the expression

$$\frac{Na_2S}{NaOH + Na_2S}$$

The sulfidity value commonly used is from 20 to 30%; the higher values usually denote better chemical recovery. The specific gravity of the liquor will be approximately 1.2, and after clarification only small quantities of grit should be present. The liquor must be considered an abrasive that produces a high rate of wear on pump rotating elements. White liquor has a tendency to crystallize on internal surfaces of pipes and pumps, but there are

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no special viscosity problems and a pump head loss allowance of about 10% above that of water should be adequate. Group B pumps are recommended.

BLOW TANK DISCHARGE As the liquor introduced with the chips into the digester combines with the noncellulose and hemicellulose fractions of the wood, it changes from white liquor to black liquor before reaching the blow tank. In addition, the sudden release of pressure frees the cellulose fibers from the other matter, so the blow tank contains both raw stock (pulp) and black liquor. Pulp from the blow tank is often entrained with air, sand, and other contaminants. A stock pump, therefore, is required for this duty because the stock concentration is quite high.

BLACK LIQUOR For convenience these pumps are divided into three groups.

Weak Black Liquor (Total Solids Up to 20%) During washing, hot water is used to dissolve away the surplus organic matter from the pulp, and the liquor produced is termed *black liquor.* This liquor is a mixture of the lignins and carbohydrates in the original wood plus the cooking chemicals: it is alkaline with a solids content of 14 to 16% in a sulfate mill. The temperature will be about 180 to 190°F (82 to 87°C), and the specific gravity about 1.08. Washing is usually carried out with a minimum of three countercurrent stages, and the solids content given previously is representative of the liquor leaving the stages nearest the inlet; that is, where it is most concentrated. The quantity of recirculated liquor is quite high, and many mills have found group A pumps with stainless trim to be satisfactory and economical. With the low solids content, there are no special viscosity problems. This may be seen from Figure 2.

Black Liquor with Total Solids of 20 to 50% This liquor is formed by the evaporation of water from weak black liquor. The concentration is accomplished in multiple-effect evaporators, which usually discharge liquor with about 50% total solids at close to 200°F (93°C). In some odor-free installations, the solids concentration is much higher. Because of the nature of the evaporation, special pumps are usually required.

Liquor containing up to 50% solids is reasonably easy to pump, but allowance must be made for viscosity effects. In noting the values in Figure 2, it should be remembered that the plant must often start up cold, so cold liquor with a higher viscosity may have to be pumped. The liquor-specific gravity rises during evaporation from about 1.1 to 1.25. Group B pumps are recommended.

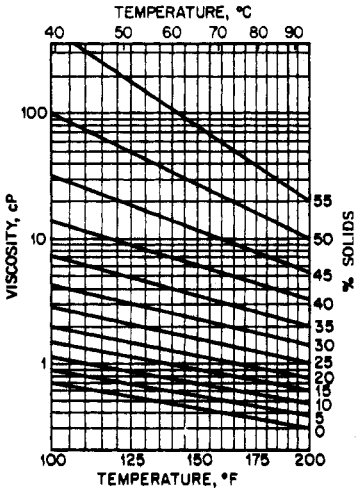


FIGURE 2 Black liquor viscosity (Courtesy of ITT/Goulds Pumps, Inc.)

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Black Liquor with Total Solids of 50 to 65% This is often referred to as heavy black liquor because the specific gravity rises to 1.35. The liquor is formed by further evaporation, either in the multiple-effect units or by contact evaporators using hot flue gas.

From a pumping standpoint, this liquor is probably the most difficult of all liquids to pump satisfactorily in pulp and paper mills. Continuous operation requires careful attention to pump sealing and maintenance. Steaming out at regular intervals of the evaporator and piping is particularly important to prevent solids buildup affecting NPSHA to the pumps.

No accurate figures are available for the viscosity of liquids with a solids concentration above 55% because there is wide variation in the liquors produced from different wood species and also in the liquors from the same wood of different ages. Hardwood species produce a more viscous liquor, especially eucalyptus, as well as more liquor per ton of pulp produced. Black liquor produced from straw pulping is even more viscous and, in addition, causes the deposition of silica on the walls of pumps and piping. An approximation of the viscosity of straw mill heavy black liquor may be determined from published figures, which give viscosities up to 2000 centistokes. This is probably at least 50% higher than liquor from normal long-fibered softwood.

During recovery, liquor is sprayed into the furnace for evaporation to dryness and burning. Prior to this, the make-up chemical (sodium sulfate or salt cake) is added and reduced to Na_2S in the reducing atmosphere of the furnace.

Little is known with certainty about heavy black liquor, but it does not seem to be very corrosive, and carbon steel is often used for pipework, although stainless steel pumps are fairly common. The pumps are subjected to severe duties—notably high heads, lumpy material, high temperature and pressure, and continuous service. Group B pumps are almost universally specified, often with casings of more wear-resistant material such as Alloy 20 or duplex stainless steels. In some cases, mills making straw pulp have not found suitable centrifugal pumps and have had to resort to gear pumps because of the very high viscosity of the liquor.

GREEN LIQUOR Green liquor is a solution of sodium carbonate and sodium sulfide plus other elements and compounds. One of these other compounds is iron sulfide in a colloidal form, which produces a greenish color. The liquor is formed by dissolving smelt from the causticizing process. Severe erosion takes place in green-liquor pumps, primarily because of the violent action inside the dissolving tank but also because of the gritty matter always present. Green liquor also builds up on the walls of pumps and piping, causing high frictional losses. The specific gravity is usually about 1.2, and an allowance of about 20% should be made for viscosity. Group B pumps are recommended for this service.

Liquor containing up to 50% solids is reasonably easy to pump, but allowance must be made for viscosity effects. In noting the values in Figure 2, it should be remembered that the plant must often start up cold, so cold liquor with a higher viscosity may have to be pumped. The liquor-specific gravity rises during evaporation from about 1.1 to 1.25. Group B pumps are recommended.

LIME SLURRIES In causticizing, various solutions and slurries are present that, apart from causing excessive wear in standard pumps, do not cause any problems. Thus, any normal slurry pump should prove satisfactory. In sulfate mills, the lime mud formed during green liquor causticizing presents the most serious problem, for approximately 1000 lb (500 kg) of mud may be formed for each ton (1000 kg) of pulp produced. Solid loads above 35% can occur, and frequent blockages are likely unless pumps are selected for minimizing downtime. For mild slurry duty, group B pumps with duplex stainless steel construction are satisfactory. For harsher applications, hard iron pumps like those used in the mining industry are employed.

BLEACH PLANT LIQUOR Most bleached pulp mills today use at least four stages of bleaching, and often six or more. Bleaching is used to remove residual lignins or to convert them to compounds that are stable regarding color and heat. The stages used include chlorination, either by hypochlorite, gaseous chlorine (both becoming obsolete), or chlorine dioxide

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(usually two stages), with an alkali extraction washing stage between. On occasions oxygen is also used. Bleach plant chemicals are usually prepared in the mill so solutions such as chlorine water, sulfuric acid, sodium chlorate, sodium chloride, sodium hydroxide, calcium hypochlorite, and chlorine dioxide all have to be pumped.

It cannot be emphasized too strongly that materials of construction are of vital importance in the chemical preparation area of the bleach plant.

In addition to the standard chemicals, some of the common pulp mill bleach substances, together with some chemical preparation systems, are as follows.

CHLORINE This is usually delivered to the mill in tank cars but is always vaporized to a gas before use.

CHLORINE WATER (HYPOCHLOROUS AND HYDROCHLORIC ACID) Concentrations cover the range from pH 2 to 10 for bleaching pulp. In some cases, the gas is mixed directly with pulp in special mixers. Group C lined pumps are essential.

SODIUM HYPOCHLORITE AND CALCIUM HYPOCHLORITE This mixture is made in the mill by permitting chlorine to react with either sodium or calcium hydroxide concentrated caustic (70%) diluted to 5 to 6% before chlorination. Calcium hypochlorite is made from a 10% solution of slaked lime at temperatures up to 150°F (66°C), but not normally exceeding 70°F (21°C). These liquors are corrosive to steel, and group C or lined pumps are necessary when handling solutions *to* the bleach plant; *after* bleaching the filtrate may still have residual hydrochloric acid.

CHLORINE DIOXIDE This is the most common bleach solution used because it gives an excellent brightness to the pulp and, despite corrosion problems, is usually cheaper than other bleach solutions.

After generation of the gas, during which absolute cleanliness is vital, the gas is stripped in a packed tower as an aqueous solution and stored in plastic tanks made of special resins that resist chemical attack. In modern plants, increasing use is made of glass-reinforced plastic with selected resins for piping, valves, and pump linings. This is sometimes a cheaper alternative than the use of exotic metals, such as titanium, for pumps. Pumps must be group C, and stainless steel is not satisfactory. Solution strengths of up to 11 g/liter are used.

SODIUM PEROXIDE AND HYDROGEN PEROXIDE These are used for bleaching groundwood pulp. Typical solutions contain sodium silicate (5%), sodium peroxide (2%), and sulfuric acid (1.5%). The latter controls the pH of the liquor. Concentrations of bleach liquors are up to 15%. Temperatures are usually less than 90°F (32°C). Group C pumps are necessary.

WASH LIQUORS In general, the filtrate from bleach washing stages will exhibit at least some of the properties of the stage immediately before washing, owing to slight excesses of chemical present. Filtrates are collected in corrosion-resistant pipes and vessels, usually made from glass-reinforced plastic, and the pumps used will be either group B or C, depending on the stage in question. The filtrate from the chlorine dioxide stages should be pumped with a super austenitic stainless steel case and trim pump because the filtrate is not as corrosive as the bleach solution.

Spent acid from chemical preparation plants is also highly corrosive, and usually stainless steel is not satisfactory for use with it.

Effluent from the bleach plant, on the other hand, is usually a mixture of several liquors, and experience has shown that 317 stainless steel is a suitable material for pumps that handle it.

CHLORINE DIOXIDE PREPARATION; SODIUM CHLORATE Chlorine dioxide is produced by permitting sodium chlorate to react with sulfuric acid and hydrochloric acid in a vessel into

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which a reducing agent such as NaCl, SO_2, or methanol is introduced in controlled quantities. Sodium chlorate solutions are usually from 43 to 46%, at which strength the specific gravity is about 1.38. Stainless steel pumps may be used, but epoxy-resin-lined pumps are superior.

FOUL CONDENSATE This arises from the evaporation of water from black liquor at the multiple-effect evaporators, as these units flash vapor from the liquor in one stage and use this to evaporate the liquid in the next stage. The vapor when condensed contains some carry-over from the black liquor, and thus the condensate is contaminated and corrosive. When a nickel cast iron casing and stainless trim are used, group A pumps should be satisfactory. Some liquors produce very corrosive vapors, and a stainless casing pump may prove necessary. Group B pumps are recommended.

Stock Stock is the term applied to the suspension of cellulose fiber in water. It first appears either after grinding (in the case of mechanical pulp) or after the blow tank (in the case of chemical pulp). Stock production rates may be converted to pump flow rates with the following formula:

$$\text{Flow (USGPM)} = \frac{\text{Production (air dried short tons per day)} \times 1.5}{\text{Consistency (\% oven dried)}}$$

After the separation of chemicals or impurities by washing and screening, the stock is given a mechanical treatment known as either beating or refining, depending on the nature of the treatment. This enhances the sheet properties. Additives such as starch, clay fillers, alum, and size are introduced to impart special characteristics, depending on the end use of the product.

Over the range of stock in normal use, the specific gravity may be considered constant for all practical purposes, with a value equal to that of water at the appropriate temperature.

Cellulose fibers have a specific gravity slightly greater than water, and constant agitation is required to ensure that stratification does not occur in storage. Agitation, however, can also introduce air, to the detriment of the stock.

The pH of stock varies over a wide range—from as low as 1.0 during some bleaching processes to 11 with others. In the paper machine room, the pH of the stock will usually range from 4 to 8. Thus from a corrosion viewpoint washed stock does not usually present special problems except when high-grade bleached products are produced. Stains will be caused by iron sulfides or oxides, and therefore stainless steel must be used—frequently 304 for washed stock, but 316 or 317 within the bleach plant before washing or where bleach liquor is likely to be present with the stock.

Unbleached paper mills generally do not experience corrosion with washed stock, except in the case of groundwood mills, where the pH is usually lower than in chemical pulp mills.

FIBER CHARACTERISTICS Stock made from softwoods will have a predominance of fibers 2.8 to 3.5 mm long and 0.25 to 0.3 mm wide; fibers from hardwoods will be about 1.0 to 1.3 mm long and 0.1 mm wide. Straw fiber will be still shorter—0.75 mm on the average—but flax can have fibers up to 9.0 mm long. These figures are typical and are of interest because of their effect on pump performance.

CONSISTENCY This is the amount of dry fiber content in the stock, expressed as a percentage by mass. Typical values will vary from about 0.1% for the feed to the headbox of a special paper machine to 16% for stock between some bleaching stages or in high-density towers. The critical stock consistency in the selection of pumps is 6%. Up to the 6% level, pumps may be selected on the basis of their water performance.

FREENESS When stock is beaten, or refined, it acquires an affinity for water, and the longer the stock is beaten, the longer the water retention period. The retention of water by the stock increases the friction factor of the flow of stock.

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Freeness is often measured by an instrument called the Canadian standard freeness tester. The range of values covers a scale from 0 to 900, with a higher freeness value indicating a less refined stock and thus a lesser affinity for water. This instrument measures the amount of water drained from a sample of stock under a regularly decreasing head. Its use is recommended by the Technical Association of the Pulp and Paper Industry (TAPPI), and it is commonly employed in North American mills.

STOCK PUMPS In stock pumps, consistency is not a major problem until a value of about 6% is reached. The essential requirement is to get the stock to the pump impeller, and every effort should be made to keep the piping as large and straight as possible. A typical open impeller, end suction stock pump is shown in Figure 3.

Above 6% consistency, special pumps are required, and they can be of the positive displacement screw type or centrifugal type. Air entrainment in the stock will reduce pump output. Air entrainment occurs from agitation in the chests, from flow over weirs, and from flow through restricted openings. How air entrained in water and in stock affects pump performance is shown in Figure 4.

PIPING ARRANGEMENT Piping should be as straight and short as possible. This is particularly important on the suction side of the pump to prevent dewatering of the stock. The diameter of the suction piping should be at least one pipe size larger than the diameter of the pump suction and should project into the stock chest. The inlet end of the suction pipe should be cut at an angle, and the bottom of the pipe should be at least $1\frac{1}{2}$ pipe diameters from the bottom of the chest. With the long side of the pipe on top, the probability of drawing air into the suction of the pump through vortices is reduced. Some manufacturers provide a lump breaker or screw feeder at the suction side of the pump for pumping stock above from 6% to 8% consistency.

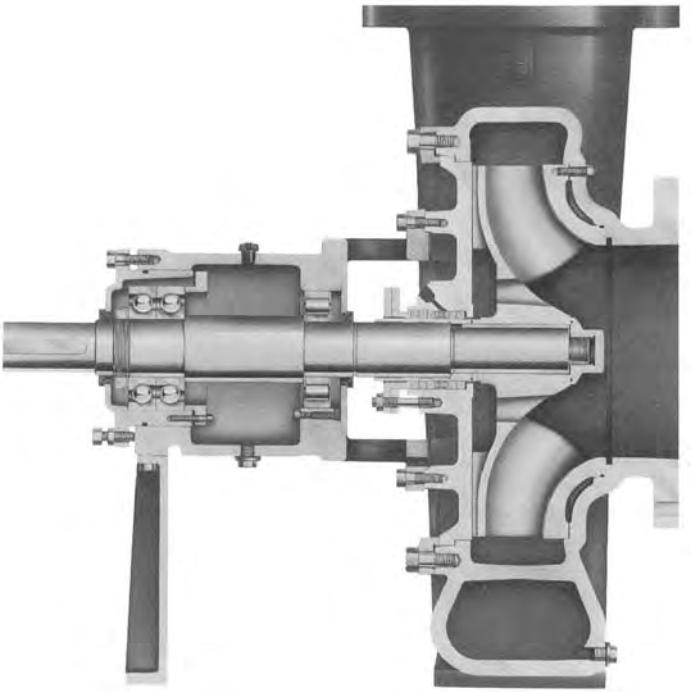


FIGURE 3 End suction stock pump (Courtesy of ITT / Goulds Pumps, Inc.)

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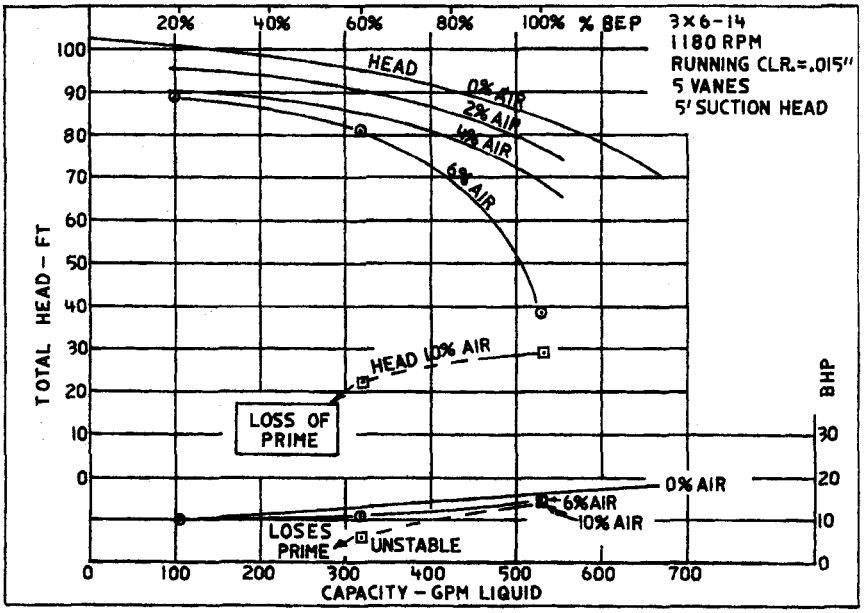


FIGURE 4 General effect of entrained gas on pump performance (Courtesy of ITT/Goulds Pumps, Inc.)

SIZE OF PUMPS It is important to estimate the performance requirements of stock pumps as accurately as possible. Oversizing of centrifugal pumps will cause an unbalanced radial thrust on the impeller resulting in excessive shaft deflection and reduced bearing and seal life. Oversizing can also result in impeller recirculation and the accompanying cavitation-like noise and damage to pump components.

PUMP TYPES AND SERVICES

The normal economic considerations of any continuous process apply equally well to pulp and paper mills, with a few points of difference. In pump installations, the improved cost figures that are possible from larger units are limited to some extent by the manufacturer's standard size units. Because the industry is capital intensive, the overriding factor in any pump installation is reliability. To achieve this, it does not necessarily follow that it is better to have two pumps installed, with one as a standby. One properly designed and serviced unit may well be better than two unknown units; this is especially true where the pump is in a portion of the process that cannot be interrupted without serious losses, either in raw materials or in quality of the finished product.

A duplication of pumps means complications in extra valves, pipework, connections for steam and viscous liquids, electric motors, cables, and starters. The result is that, in modern mills with good machinery and materials of construction, there is a strong tendency away from the duplication of pumps because of increased costs and questionable reliability. It follows that the important thing is to select the right pump and the right duty point in a particular range.

There may be several hundred pumps in a modern pulp and paper mill, but the cost of these pumps is probably less than 5% of the total equipment cost. It is unwise, therefore, to jeopardize mill reliability by compromising pump quality. Corrosion and erosion are major factors in pump life, and even with the best materials, the life of some components

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in severe service may be 12 months or less. Moreover, the power used by pumps is usually less than one-third of the mill demand. If one remembers that the cost of total power absorbed in a mill is only around 4%, even a 50% reduction in pump power will still be less than 1% net.

Efficiency The best point at which to operate a pump is, of course, its maximum efficiency, but this is not always possible, particularly in the case of stock pumps because of the wide range of process variables in most pulp and paper mills. Open impellers and excess clearances also reduce the efficiency, yet these factors are much more important in stock pumps than efficiency. Another important consideration is speed. Stock pumps should be chosen to run at as low a speed as possible to achieve stable operation, and this speed may not produce an efficient pump. The shape of the pump performance curve is much more important than the best efficiency quoted by a manufacturer. A flat or unstable head curve may produce surging or instability in the pump output. Good pump selection, therefore, must emphasize reliability as the first consideration and efficiency and costs as secondary considerations.

Pump Speeds Most of the pump duties in a mill can be accomplished by single-stage pumps and four-pole motor speeds. For liquids other than water, two-pole motor speeds should be avoided if possible. For special duties, including stock pumping, six- or eight-pole motors may be required unless some indirect or variable speed is used. Although it is true that lower speeds mean larger pumps and more expensive electric motors, lower speeds are justified because of reduced maintenance and greater reliability. Some deviation from these speeds may be necessary for pumps generating heads in excess of 150 ft (46 m), but this can often be taken care of by a larger impeller rather than a higher shaft speed.

Although not an option in every circumstance, the use of variable speed drive systems for pumps to replace control valves should also be considered. Reliable alternating current variable frequency drives are now readily available up to 2300 volts and are easily justifiable based on normal power costs at the 440-volt level. The commensurate speed reduction seen by the pump when replacing a control valve and avoiding its necessary pressure drop will also add to the reliability and life of the pump.

Multistage Pumps Except with boiler feedwater, the use of multistage pumps should be avoided. This is particularly true for stock and viscous liquors. The complicated pump design makes such units unacceptable for these services.

Pipeline Systems With black liquor, green liquor, and similar high-viscosity liquors, adequate provision must be made for steaming out and subsequent liquor drainage. The pumps must also be included in this system. Although the liquor pumps should be designed to pass some solid matter, motorized strainers should be used on the pumps for cyclone evaporators and recovery boiler-fuel pumps because both pumps discharge to spray nozzles. Dead pockets and other areas where liquor can collect should be avoided because solids from the liquor will build up in these areas and possibly break away to block pipelines or pump impellers. For protection at shutdowns, even for short periods, steaming out is essential.

Positive Displacement Pumps The principal use of the positive displacement pump is for consistency control of stocks above 5%. The normal measuring device used is quite satisfactory at low consistencies but is less reliable at the higher values. More satisfactory control may be achieved by using a screw pump, where the power is proportional to the pulp consistency at constant flow. Such pumps have been very reliable on consistency control.

Digester Circulating Pumps Digester circulating pumps are used with indirectly heated batch digesters to circulate the liquor at the digester pressure and temperature. Maintenance problems are common on these pumps because heads can be as high as 150 lb/in^2 (1034 kPa) and temperatures as high as 350°F (177°C). In addition, the circulating liquor contains some raw pulp even though screens are fitted to the digester outlets.

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Pumps for this service, therefore, should be centerline mounted, of very heavy construction, have closed impellers, and mechanical seals. In addition, there is often considerable pipework involved, for a digester may easily be 60 ft (18 m) high and pipe loads are often imposed on the pumps. Expansion joints and long-radius bends are used, but it is desirable to support the pumps on springs or slide bases.

Pumps for Heavy Black Liquor above 60% Solids A typical pump for this service is shown in Figure 11. An open impeller in a 316 stainless steel casing is recommended. A heavy sleeved shaft of 316 stainless steel with ample clearance between the rotating parts is also required for satisfactory operation. These pumps may be required to handle black liquor up to 2000 centistokes viscosity and should be provided with water cooling. Steam jacketing is not always satisfactory because the liquor may tend to bake on the walls of the casing. Pump speeds should be below 1800 rpm, if possible.

Multiple-Effect Evaporator Pumps Pumps in this service often operate in cavitation owing to problems in regulating the flow between evaporation stages. Level control valves in the suction line to the pump can alleviate this problem, but cavitation can still be expected in the pump. A self-priming pump may give longer life of the rotating elements than the condensate pump usually used on this service.

Diaphragm Pumps Diaphragm pumps are used in pumping lime mud slurries of high concentration. They consist of a rubber or neoprene diaphragm with a pulsating air supply on one side, controlled by a timer, and the slurry on the other.

Medium Consistency Stock Pumps Pumping of medium consistency stock (from about 8% to about 16% consistency) can be accomplished with positive displacement screw pumps or centrifugal pumps that have been specially adapted. Medium consistency stock is a thick viscoelastic material made up of strong networks of fibers called flocs. A high shear rate is necessary to render the stock capable of flow. Therefore all centrifugal medium consistency pumps are equipped with an inducer or feeder vane device in front of the pump suction to provide the shear necessary for the stock to flow into the impeller eye. There are also large amounts of entrained air in medium consistency stock and most centrifugal medium consistency pumps are equipped as well with an auxiliary vacuum pump to remove the air allowing for stable pump operation and high efficiencies. An example of a medium consistency centrifugal pump is shown in Figure 5.

Fan Pump Another specialized pump application in the paper mill is the pump that feeds the headbox or nozzle that spreads white paper onto the moving wire sheet that forms the paper. The *fan pump* is usually a large horizontally split double suction pump in austenitic (AISI 316) trim or all austenitic (AISI 316) stainless steel construction. To form an evenly distributed sheet, the headbox must be provided with flow that is free of pressure pulsations and flow disturbances. The fan pump rotating element (impeller, shaft, sleeves, and so on) is designed and built to special tolerances to guarantee a typical maximum pressure pulse of 0.5% of pump total dynamic head, peak-to-peak, at any frequency. A typical fan pump is shown in Figure 6.

Shower Services Shower services require pumps to operate at low flows and high pressures. Operation at low flows requires special pump designs to ensure good pump reliability. These pumps are usually of single-stage design with special, circular volute construction to minimize radial loads on the rotor and bearings. A typical low flow pump is shown in Figure 7. Some high pressure shower services require multistage or high speed (greater than 3600 rpm) pumps to reach the desired pressure. Generally, multistage and high speed pumps require more maintenance than single-stage units because of mechanical complexity.

Solid Handling Services There are many pump applications that require handling of various size solids, particularly in recycle services. Recycle services may contain tramp metal and plastics that can clog stock pumps. Recessed impeller pumps, sometimes

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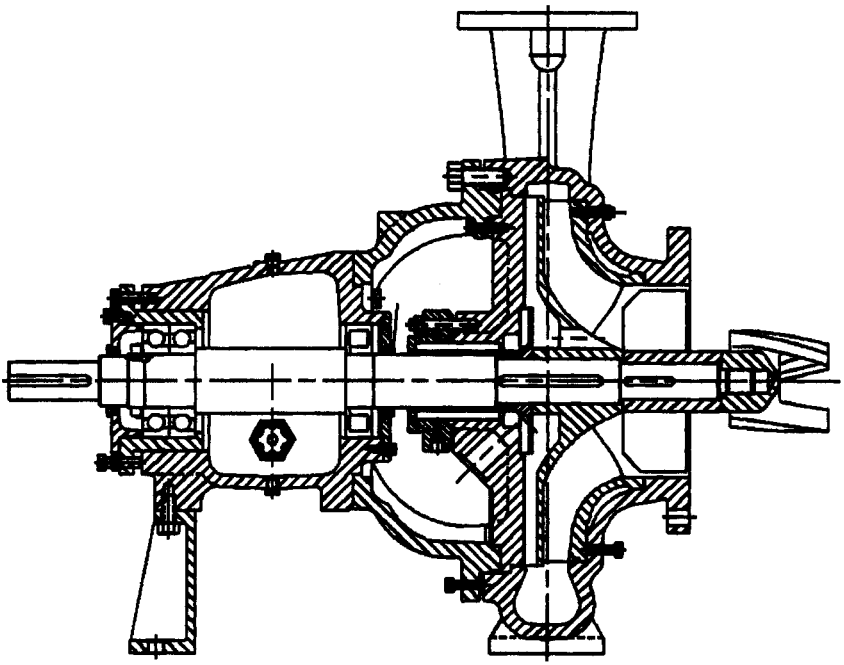


FIGURE 5 Medium consistency stock pump (Courtesy of ITT/Goulds Pumps, Inc.)

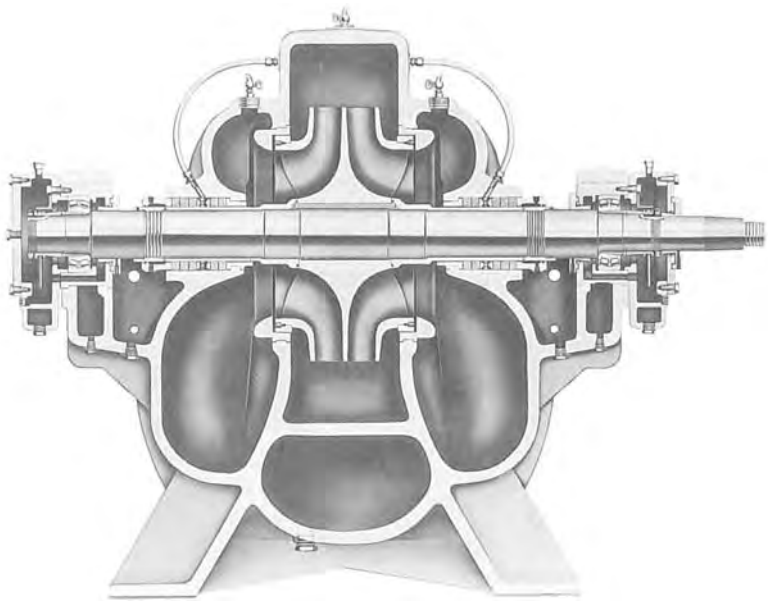


FIGURE 6 Fan pump (Courtesy of ITT/Goulds Pumps, Inc.)

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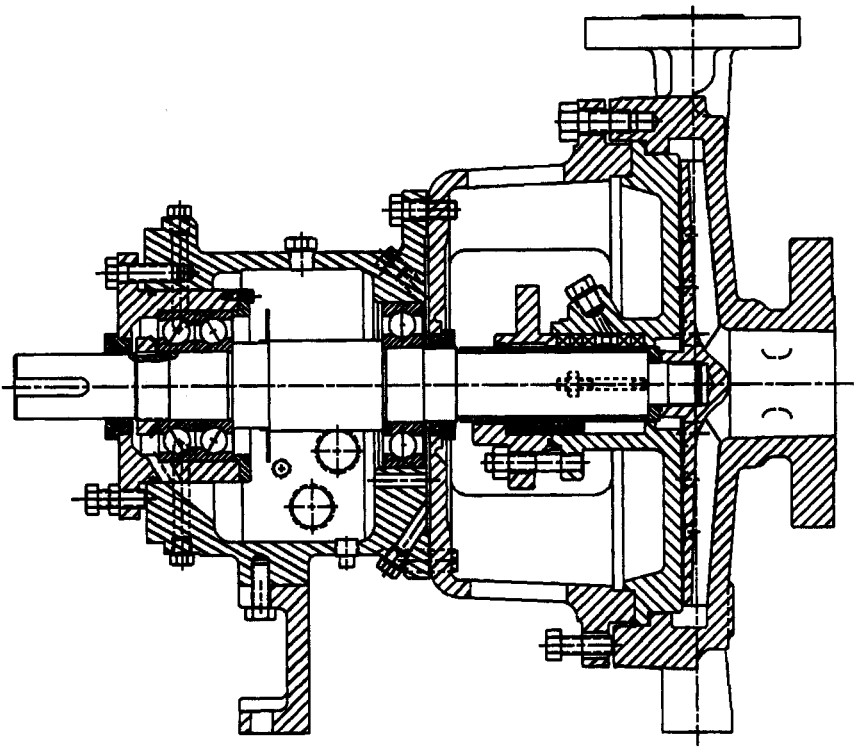


FIGURE 7 A typical low flow pump used for shower services (Courtesy of ITT/Goulds Pumps, Inc.)

referred to as vortex pumps, are particularly useful for these types of services because the impeller is recessed from the main flow allowing large solids to pass through the pump.

Sealless Pump Services In recent years, some of the severely corrosive chemical applications have been served well by sealless pumps. Magnetic drive pumps are particularly useful for tank car unloading or sodium hydroxide and other chemicals where mechanical seals have required high levels of maintenance (See Section 2.2.3).

Vacuum Pumps These are used to extract water from the sheet on the fourdrinier wire and at the suction presses by means of a vacuum up to a maximum of about 25 inHg (635 mmHg). Approximately 40,000 lb (18,100 kg) of water is extracted by this means for every ton (1000 kg) of paper produced, and this water is removed by entrainment with the air handled by the vacuum pumps. Frequently water separators are used to remove water; their use is a matter of economics, as a reduction of up to 10% in power may be achieved.

Vacuum pumps are of three basic types:

1. Water ring
2. Positive displacement
3. Centrifugal or axial-flow

Many engineers prefer the water-ring type, probably because of its simplicity. In general, however, this type uses more power, mainly because of the heating of the circulating water, which is then discharged to a drain.

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Centrifugal and axial-flow machines must be provided with water separators, but they are more efficient overall, particularly when the heat of compression is used in the machine room ventilating system. The machines run at high speed and are usually driven by a steam turbine.

The paper machine system requires vacuums at different levels, from a few inches (millimeters) of mercury to the maximum possible. Often pumps are connected to a common header, and orifice plates are used to divide the flow to ensure some measure of standby capacity. This involves throttling, however, and may create flow problems unless quantities are carefully estimated. The axial-flow machine permits extraction at any point along the rotor within fairly close limits and requires an accurate estimation of the quantities and specific vacuum required. A standby for axial-flow machines cannot usually be justified.

Stock and Liquor Pump Standardization Throughout the mill the duties of many pumps are similar, but different materials of construction may be used. If, at a slightly extra initial cost, the rotating elements of the pump can be standardized, this will reduce spare-parts inventories. This is also an advantage when purchasing pumps for a new mill. For example, if a standard arrangement consists of a complete rotating element, including bearings, only the impeller size and material need be different. Standardization is an additional reason for recommending that all pumps be stainless-fitted. Obviously, large pumps need individual evaluation. Standardization of stock pumps is less feasible, but up to about 6% stock, similar pumps can usually be specified.

Pump Selection Guidelines In many cases, an excess margin on head and capacity is specified. Where margins are excessive, mechanical troubles and cavitation often occur. The following guidelines will help to properly size a pump and motor in order to avoid these problems:

1. Carefully calculate the pump duty, using the TAPPI technical information sheet (TIS 410-14) for stock friction and the proper viscosity corrections for liquor friction. Always use a schematic of the actual piping system to be installed.
2. Properly size control valves to minimize the required pressure drop. Consider the use of variable speed drives.
3. Make the proper corrections to head and power for the presence of entrained gas in the fluid being pumped, per pump manufacturers recommendations.
4. Select a pump from the manufacturer's curve and note the impeller diameter and range for the pump. If the duty point falls near the end of the curve or if the impeller diameter is greater than 95% of the maximum impeller diameter, it is advisable to select the next size larger pump. For maximum reliability, select an impeller diameter equal to or nearly equal to: [(maximum impeller diameter—minimum impeller diameter) × 0.75] + minimum impeller diameter.
5. Allow 10% or 3 ft (whichever is larger) between NPSHA and NPSHR. Systems should be designed for a maximum available S value (suction specific speed) of 8,000.
6. Motors should be sized to be non-overloading for 105% of the impeller diameter chosen. Baseplates should allow for the installation of the next larger frame size.

SPECIAL PULP AND PAPER CONSIDERATIONS

Friction Loss of Pulp Suspensions in Pipe In any stock piping system, the pump provides flow and develops hydraulic pressure (head) to overcome the differential in head between two points. This total head differential consists of pressure head, static head, velocity head, and total friction head produced by friction between the pulp suspension and

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the pipe, bends, and fittings. The total friction head is the most difficult to determine because of the complex, nonlinear nature of the friction loss curve. This curve can be affected by many factors.

The following analytical method for determining pipe friction loss is based on the TAPPI Technical Information Sheet (TIS) 410-14 (Reference 1), and is applicable to stock consistencies (oven-dried) from 2 to 6%. Normally, stock consistencies of less than 2% (oven-dried) are considered to have the same friction loss characteristic as water.

The friction loss of pulp suspensions in pipe, as presented here, is intended to supersede the various methods previously issued.

Figure 8 and Figure 9 show typical friction loss curves for two different consistencies ($C_2 > C_1$) of chemical pulp and mechanical pulp, respectively.

The friction loss curve for chemical pulp can be conveniently divided into three regions, as illustrated by the shaded areas of Figure 10.

These regions may be described as follows:

Region 1 (Curve AB) is a linear region where friction loss for a given pulp is a function of consistency, velocity, and pipe diameter. The velocity of the upper limit of this linear region (Point B) is designed V_{max}.

Region 2 (Curve BCD) shows an initial decrease in friction loss (to Point C) after which the friction loss again increases. The intersection of the pulp friction loss curve and the water friction loss curve (Point D) is termed the onset of drag reduction. The velocity at this point is designated V_W.

Region 3 (Curve DE) shows the friction loss curve for pulp fiber suspensions below the water curve. This is due to a phenomenon called drag reduction. Reference 2 describes the mechanisms which occur in this region.

Regions 2 and 3 are separated by the friction loss curve for water, which is a straight line with a slope approximately equal to 2.

The friction loss curve for mechanical pulp, as illustrated in Figure 11, is divided into only two regions:

Regions 1 and 3. For this pulp type, the friction loss curve crosses the water curve at V_W and there is no true V_{max}.

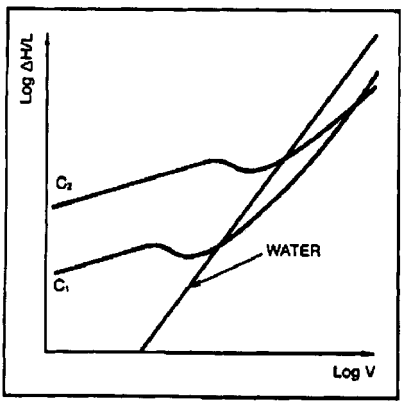


FIGURE 8 Friction loss curves for chemical pulp (Courtesy of ITT/Goulds Pumps, Inc.)

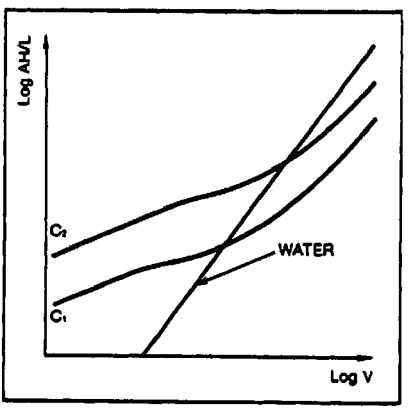


FIGURE 9 Friction loss curves for mechanical pulp (Courtesy of ITT/Goulds Pumps, Inc.)

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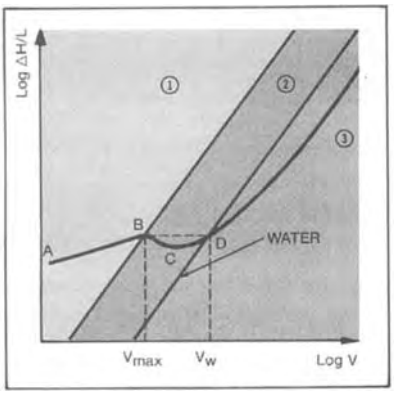


FIGURE 10 Friction loss curves for chemical pulp, shaded to show individual regions (Courtesy of ITT/Goulds Pumps, Inc.)

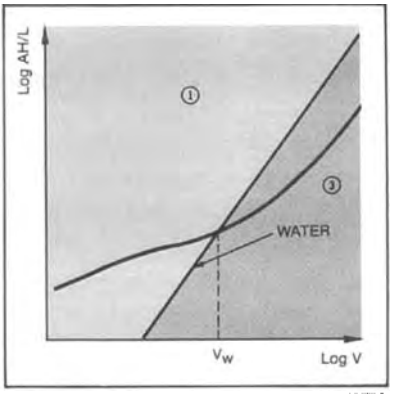


FIGURE 11 Friction loss curves for mechanical pulp, shaded to show individual regions (Courtesy of ITT/Goulds Pumps, Inc.)

To determine the pipe friction loss component for a specified design basis (usually daily mass flow rate), the following parameters must be defined:

a) **Pulp Type** Chemical or mechanical pulp, long or short fibered, never dried or dried and reslurried, and so on. This is required to choose the proper coefficients which define the pulp friction curve.

b) **Consistency, *C* (oven-dried)** Often a design constraint in an existing system. *Note*: If air-dried consistency is known, multiply by 0.9 to convert to oven-dried consistency.

c) **Internal pipe diameter, *D*** Lowering D reduces initial capital investment, but increases pump operating costs. Once the pipe diameter is selected, it fixes the velocity for a prespecified mass flow rate.

d) **Bulk velocity, *V*** Usually based on a prespecified daily mass flow rate. Note that both V and D are interdependent for a constant mass flow rate.

e) **Stock temperature, *T*** Required to adjust for the effect of changes in viscosity of water (the suspending medium) on pipe friction loss

f) **Freeness** Used to indicate the degree of refining or to define the pulp for comparison purposes

g) **Pipe material** Important to specify design correlations and compare design values

The bulk velocity (V) will depend on the daily mass flow rate and the pipe diameter (D) selected. The final value of V can be optimized to give the lowest capital investment and operating cost with due consideration of future demands or possible system expansion.

The bulk velocity will fall into one of the regions previously discussed. Once it has been determined in which region the design velocity will occur, the appropriate correlations for determining pipe friction loss value(s) may be selected. The following describes the procedure to be used for estimating pipe friction loss in each of the regions.

Region 1 The upper limit of Region 1 in Figure 10 (Point B) is designated V_{max}. The value of V_{max} is determined using Eq. 1 and data given in Table I or IA, Appendix B.

$$V_{max} = K'C^{\sigma}(\text{ft/s}), \tag{1}$$

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where K' = numerical coefficient (constant for a given pulp is attained from Table I or IA)
C = consistency (oven-dried, expressed as a percentage, not decimally)
σ = exponent (constant for a given pulp), obtained from Table I or IA

If the proposed design velocity (V) is less than V_{max}, the value of flow resistance ($\Delta H/L$) may be calculated using Eq. 2 and data given in Table II or IIA, and the appendices.

$$H/L = FKV^{\alpha}C^{\beta}D^{\gamma} \text{ (ft/100 ft)} \quad (2)$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D)
K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA
V = bulk velocity (ft/s)
C = consistency (ven-dried, expressed as a percentage, not decimally)
D = pipe inside diameter (in)
α, β, γ = exponents (constant for a given pulp), obtained from Table II or IIA

For mechanical pumps, there is no true V_{max}. The upper limit of the correlation equation (Eq. 2) is also given by Eq. 1. In this case, the upper velocity is actually V_W.

Region 2 The lower limit of Region 2 in Figure 10 (Point B) is V_{max} and the upper limit (Point D) is V_W. The velocity of the stock at the onset of drag reduction is determined using Eq. 3.

$$V_W = 4.00C^{1.40} \text{ (ft/s)} \quad (3)$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

If V is between V_{max} and V_W, Eq. 2 may be used to determine $\Delta H/L$ at the maximum point (V_{max}). Because the system must cope with the worst flow condition, $\Delta H/L$ at the maximum point (V_{max}) can be used for all design velocities between V_{max} and V_W.

Region 3 A conservative estimate of friction loss is obtained by using the water curve. $(\Delta H/L)_W$ can be obtained from a Friction Factor vs. Reynolds Number plot (for example, Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3), or approximated from the following equation (based on the Blasius equation.)

$$(\Delta H/L)_W = 0.58V^{1.75}D^{-1.25} \text{ (f/100 ft)} \quad (4)$$

where V = bulk velocity (ft/s)
D = pipe diameter (in)

Previously published methods for calculating pipe friction loss of pulp suspensions gave a very conservative estimate of head loss. The method just described gives a more accurate estimate of head loss due to friction, and has been used successfully in systems in North America and world-wide.

Please refer to Appendix A for equivalent equations for use with metric (SI) units. Tables I and IA are located in Appendix B; Tables II and IIA are located in Appendix C. Pertinent equations, in addition to those herein presented, are located in Appendix D. Example problems are located in Appendix E.

The friction head loss of pulp suspensions in bands and fittings may be determined from the basic equation for head loss, Eq. 5.

$$H = KV_1^2/2g \text{ (ft)} \quad (5)$$

where K = loss coefficient for a given fitting
V_1 = inlet velocity (ft/s)
g = acceleration due to gravity (32.2 ft/s²)

Values of K for the flow of water through various types of bends and fittings are tabulated in numerous reference sources (for example, Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3). The loss coefficient for valves may be obtained from the valve manufacturer. See Section 11.1.

The loss coefficient for pulp suspensions in a given bend or fitting generally exceeds the loss coefficient for water in the same bend or fitting. As an approximate rule, the loss coefficient (K) increases 20% for each 1% increase in oven-dried stock consistency. Please note that this is an approximation; actual values of K may differ, depending on the type of bend or fitting under consideration (4).

Appendix A When metric (S/I) units are utilized, the following replace the corresponding equations in the main text.

$$V_{max} = K'C^{\sigma} \text{ (m/s)} \tag{1M}$$

where K = numerical coefficient (constant for a given pulp), obtained from Table I or IA
C = consistency (oven-dried, expressed as a percentage, not decimally)
σ = exponent (constant for a given pulp), obtained from Table I or IA

$$\Delta H/L = FKV^{\alpha}C^{\beta}D^{\gamma} \text{ (m/100 m)} \tag{2M}$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D)
K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA
V = bulk velocity (m/s)
C = consistency (oven-dried, expressed as a percentage, not decimally)
D = pipe inside diameter (mm)
α, β, γ = exponents (constant for a given pulp), obtained from Table II or IIA

$$V_W = 1.22C^{1.40} \text{ (m/s)} \tag{3M}$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

$$(\Delta H/L)_w = 264V^{1.75}D^{-1.25} \text{ (m/100m)} \tag{4M}$$

where V = bulk velocity (m/s)
D = pipe inside diameter (mm)

$$H - KV_1{}^2/2g \text{ (m)} \tag{5M}$$

where K = loss coefficient for a given fitting
V_1 = inlet velocity (m/s)
g = acceleration due to gravity (9.81 m/s²)

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Appendix B

TABLE I Data for use with Eq. 1 or Eq. 1M to determine velocity limit, V_{max}[1]

Pulp Type	Pipe Material	K'	σ
Unbeaten aspen sulfite never dried	Stainless Steel	0.85 (0.26)	1.6
Long fibered kraft never dried CSF = 725[6]	PVC	0.98 (0.3)	1.85
	Stainless Steel	0.89 (0.27)	1.5
Long fibered kraft never dried CSF = 650[6]	PVC	0.85 (0.26)	1.9
Long fibered kraft never dried CSF = 550[6]	PVC	0.75 (0.23)	1.65
Long fibered kraft never dried CSF = 260[6]	PVC	0.75 (0.23)	1.8
Bleached kraft never dried and reslurried[6]	PVC	0.79 (0.24)	1.5
	Stainless Steel	0.59 (0.18)	1.45
Long fibered kraft never dried and reslurried[6]	PVC	0.49 (0.15)	1.8
Kraft birch dried and reslurried[6]	PVC	0.69 (0.21)	1.3
Stone groundwood CSF = 114	PVC	4.0 (1.22)	1.40
Refiner groundwood CSF = 150	PVC	4.0 (1.22)	1.40
Newsprint broke CSF = 75	PVC	4.0 (1.22)	1.40
Refiner groundwood (hardboard)	PVC	4.0 (1.22)	1.40
Refiner groundwood (insulating board)	PVC	4.0 (1.22)	1.40
Hardwood NSSC CSF = 620	PVC	0.59 (0.18)	1.8

Notes:

1. When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).
2. Original data obtained in stainless steel and PVC pipe. PVC is taken to be hydraulically smooth pipe.
3. Stainless steel may be hydraulically smooth although some manufacturing processes may destroy the surface and hydraulic smoothness is lost.
4. For cast iron and galvanized pipe, the K' values will be reduced. No systematic data are available for the effects of surface roughness.
5. If pulps are not identical to those shown, some engineering judgment is required.
6. Wood is New Zealand Kraft pulp.

TABLE IA Data (5, 6) for use with Eq. 1 or Eq. 1M to determine velocity limit, V_{max}

Pulp Type[5]	Pipe Material	K'	σ
Unbleached sulphite	Copper	0.98 (0.3)	1.2
Bleached sulphite	Copper	0.98 (0.3)	1.2
Kraft	Copper	0.98 (0.3)	1.2
Bleached straw	Copper	0.98 (0.3)	1.2
Unbleached straw	Copper	0.98 (0.3)	1.2

Estimates for othe pulps based on published literature.

Pulp Type[5, 6]	Pipe Material	K'	σ
Cooked groundwood	Copper	0.75 (0.23)	1.8
Soda	Steel	4.0 (1.22)	1.4

Note: When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).

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Appendix C

TABLE II Data for use with Eq. 2 or Eq. 2M to determine head loss, $\Delta H/L$[1]

Pulp Type	K	α	β	y
Unbeaten aspen sulfite never dried	5.30 (235)	0.36	2.14	−1.04
Long fibered kraft never dried CSF = 725[5]	11.80 (1301)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 650[5]	11.30 (1246)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 550[5]	12.10 (1334)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 260[5]	17.00 (1874)	0.31	1.81	−1.34
Bleached kraft bleached and reslurried[5]	8.80 (970)	0.31	1.81	−1.34
Long fibered kraft dried and reslurried[5]	9.40 (1036)	0.31	1.81	−1.34
Kraft birch dried and reslurred[5]	5.20 (236)	0.27	1.78	−1.08
Stone groundwood CSF = 114	3.81 (82)	0.27	2.37	−0.85
Refiner groundwood CSF = 150	3.40 (143)	0.18	2.34	−1.09
Newspaper broke CSF = 75	5.19 (113)	0.36	1.91	−0.82
Refiner groundwood CSF (hardboard)	2.30 (196)	0.23	2.21	−1.29
Refiner groundwood CSF (insulating board)	1.40 (87)	0.32	2.19	−1.16
Hardwood NSSF CSF = 620	4.56 (369)	0.43	2.31	−1.20

Notes:

1. When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).
2. Original data obtained in stainless steel and PVC pipe (7, 8, 9).
3. No safety factors are included in the above correlations.
4. The friction loss depends considerably on the condition of the inside of the pipe surface (10).
5. Wood is New Zealand Kraft pulp.

TABLE IIA Data[5,6] for use with Eq. 2 or Eq. 2M to determine head loss, $\Delta H/L$

Pulp Type[5]	K	α	β	y
Unbleached sulfite	12.69 (1438)	0.36	1.89	−1.33
Bleached sulfite	11.40 (1291)	0.36	1.89	−1.33
Kraft	11.40 (1291)	0.36	1.89	−1.33
Bleached straw	11.40 (1291)	0.36	1.89	−1.33
Unbleached straw	5.70 (646)	0.36	1.89	−1.33

Estimates for other pulps based on published literature.

Pulp Type[5,6]	K	α	β	y
Cooked groundwood	6.20 (501)	0.43	2.13	−1.20
Soda	6.50 (288)	0.36	1.85	−1.04

Note: When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).

Appendix D

The following gives supplemental information to that where T.P.D. mill capacity (metric tons per day), provided in the main text.

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1. Capacity (flow), Q—

$$Q = \frac{16.65(\text{T.P.D.})}{C} \text{ (U.S. GPM)}, \quad \text{(i)}$$

where T.P.D. = mill capacity (short tons per day)
C = consistency (oven-dried, expressed as a percentage, *not* decimally)

If SI units are used, the following would apply:

$$Q = \frac{1.157(10^{-3})(\text{T.P.D.})}{C} \text{ (m}^3\text{/s)} \quad \text{(iM)}$$

where T.P.D. = mill capacity (metric tons per day)
C = consistency (oven-dried, expressed as a percentage, *not* decimally)

2. Bulk velocity, V—

$$V = \frac{0.321Q}{A} \text{ (ft/s)} \quad \text{(ii)}$$

or

$$V = \frac{0.4085Q}{D^2} \text{ (ft/s)} \quad \text{(ii)}$$

where Q = capacity (U.S. GPM)
A = inside area of pipe (in^2)
D = inside diameter of pipe (in)

The following would apply if SI units are used:

$$V = \frac{1(10^6)Q}{A} \text{ (m/s)} \quad \text{(iiM)}$$

or

$$V = \frac{1.273(10^6)Q}{D^2} \text{ (m/s)} \quad \text{(iiM)}$$

where Q = capacity (m^3/s)
A = inside area of pipe (mm^2)
D = inside diameter of pipe (mm)

3. Multiplification Factor, F (included in Eq. 2)—

$$F = F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5 \quad \text{(iv)}$$

where F_1 = correction factor for temperature. Friction loss calculations are normally based on a reference pulp temperature of 95°F (35°C). The flow resistance may be increased or decreased by 1% for each 1.8°F (1°C) below or above 95°F (35°C), respectively. This may be expressed as follows:

$$F_1 = 1.528 - 0.00556T_1 \quad \text{(v)}$$

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where T = pulp temperature (°F), or

$$F_1 = 1.35 - 0.01T_1 \qquad \text{(vM)}$$

where T = pulp temperature (°C)

F_2 = correction factor for pipe roughness. This factor may vary due to manufacturing processes of the piping, surface roughness, age, and so on. Typical values for PVC and stainless steel piping are listed below:

$F_2 = 1.0$ for PVC piping

$F_2 = 1.25$ for stainless steel piping

Please note that the previous values are typical values; experience and/or additional data may modify the factors.

F_3 = correction factor for pulp type. Typical values are listed below:

$F_3 = 1.0$ for pulps that have never been dried and reslurried

$F_3 = 0.8$ for pulps that have been dried and reslurried

Note: This factor has been incorporated in the numerical coefficient, K, for the pulps listed in Table II. When using Table II, F_3 should *not* be used.

F_4 = correction factor for beating. Data have shown that progressive beating causes, initially, a small decrease in friction loss, followed by a substantial increase. For a kraft pine pulp initially at 725 CSF and $F_4 = 1.0$, beating caused the freeness to decrease to 636 CSF and F_4 to decrease to 0.96. Progressive beating decreased the freeness to 300 CSF and increased F_4 to 1.37 (see K values in Table II). Some engineering jugment may be required.

F_5 = design safety factor. This is usually specified by company policy with consideration given to future requirements.

Appendix E

The following are three examples which illustrate the method for determination of pipe friction loss in each of the three regions shown in Figure 10.

EXAMPLE 1 Determine the friction loss (per 100 ft of pipe) for 1000 U.S. GPM of 4.5% oven-dried unbeaten aspen sulfite stock, never dried, in 8 in schedule 40 stainless steel pipe (pipe inside diameter = 7.981 in). Assume the pulp temperature to be 95°F.

Solution:

a. The bulk velocity, V, is

$$V = \frac{0.4085\,Q}{D^2} \qquad \text{(ii)}$$

and Q = flow = 1000 U.S. GPM

D = pipe inside diameter = 7.981 in

$$V = \frac{0.4085\,(1000)}{7.981^2} = 6.41 \text{ ft/s} \qquad ()$$

b. It must be determined in which region (1, 2, or 3) this velocity falls. Therefore, the next step is to determine the velocity at the upper limit of the linear region, V_{max}.

$$V_{max} = K'C^{\sigma} \qquad (1)$$

and K' = numerical coefficient = 0.85 (from Appendix B, Table I)

C = consistency = 45%

σ = exponent = 1.6 (from Appendix B, Table I)

$V_{max} = 0.85(4.5^{1.6}) = 9.43$ ft/s

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c. Since V_{max} exceeds V, the friction loss, $\Delta H/L$, falls within the linear region, Region 1. The friction loss is given by the correlation:

$$\Delta H/L = FKV^{n}C^{\beta}D^{y} \tag{2}$$

and F = correction factor = $F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5$

F_1 = correction factor for pulp temperature. Since the pulp temperature is 95°F

$F_1 = 1.0$

F_2 = correction factor for pipe roughness. For stainless steel pipe,

$F_2 = 1.25$ (from Appendix D)

F_3 = correction factor for pulp type. Numerical coefficients for this pulp are contained in Appendix C, Table II, and have already incorporated this factor.

F_4 = correction factor for beating. No additional beating has taken place, therefore

$F_4 = 1.0$ (from Appendix D)

F_5 = design safety factor. This has been assumed to be unity.

$F_5 = 1.0$.

$F = (1.0)(1.25)(1.0)(1.0)(1.0) = 1.25$

K = numerical coefficient = 5.30 (from Appendix C, Table II)

σ, β, y = exponents = 0.36, 2.14, and −1.04, respectively (from Appendix C, Table II)

V, C, D have been evaluated previously.

$$\Delta H/L = (1.25)(5.30)(6.41^{0.36})(4.5^{2.14})(7.981^{-1.04})$$
$$= (1.25)(5.30)(1.952)(25.0)(0.1153)$$
$$= 37.28 \text{ ft head loss/100 ft of pipe}$$

This is a rather substantial head loss, but may be acceptable for short piping runs. In a large system, the economics of initial piping costs versus power costs should be weighed, however, before using piping which gives a friction loss of this magnitude.

EXAMPLE 2 Determine the friction loss (per 100 ft of pipe) of 2500 U.S. GPM of 3% oven-dried bleached kraft pine, dried and reslurried, in 12 in schedule 10 stainless steel pipe (pipe inside diameter = 12.39 in). Stock temperature is 1250°F.

Solution:

a. V, the bulk velocity, is

$$V = \frac{0.4085\,Q}{D^2} \tag{ii}$$
$$= \frac{0.4085(2500)}{12.39^2} = 6.65 \text{ ft/s}$$

b. The velocity at the upper limit of the linear region, V_{max}, is

$$V_{max} = K'C^{\sigma} \tag{1}$$

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and $K' = 0.59$ (from Appendix B, Table I)
$C = 3.0$ (from Appendix B, Table I)
$\sigma = 1.45$ (from Appendix B, Table 1)
$V_{max} = 0.59\,(3.0^{1.45}) = 2.90$ ft/s

c. Region 1 (the linear region) has been eliminated, since the bulk velocity, V, exceeds V_{max}. The next step requires calculation of V_W.

$$V_W = 4.00\,C^{1.40} \tag{3}$$
$$= 4.00(3.0^{1.40}) = 18.62 \text{ ft/s}$$

d. V exceeds V_{max}, but is less than V_W, indicating that it falls in Region 2. The friction loss in this region is calculated by substituting V_{max} into the equation for head loss, Eq. 2.

$$\Delta H/L = F\,K\,(V_{max})^{\varkappa}\,C^{\beta}\,D^{y}$$

and
$$F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5 \tag{iv}$$
$$F_1 = 1.528 - 0.00556T \tag{v}$$

and T = stock temperature = 125°F
$F_1 = 1.58 - 0.00556(125) = 0.833$
$F_2 = 1.25$ (from Appendix D)
$F_3 = F_4 = F_5 = 1.0$
$F = 0.833(1.25)(1.0) = 1.041$
$K = 8.80$ (from Appendix C, Table II)
$\alpha, \beta, y = 0.31$, 1.81, and -1.34, respectively (from Appendix C, Table II)

V_{max}, C, and D have been defined previously.

$$\Delta H/L = 1.041(8.80)(2.90^{0.31})(3.0^{1.81})(12.39^{-1.34})$$
$$= 1.041(8.80)(1.391)(7.304)(0.03430)$$
$$= 3.19 \text{ ft head loss/100 ft of pipe}$$

EXAMPLE 3 Determine the friction loss (per 100 ft of pipe) for 2% oven-dried bleached kraft pine, dried and reslurried, through 6 in schedule 40 stainless steel pipe (inside diameter = 6.065 in). The pulp temperature is 90°F; the flow rate 1100 U.S. GPM.

Solution:

a. The bulk velocity is

$$V = \frac{0.4085\,Q}{D^2}$$
$$= \frac{0.4085(1100)}{6.065^2} = 12.22 \text{ ft/s} \tag{ii}$$

b. It must be determined in which region (1, 2, or 3) this velocity falls. To obtain an initial indication, determine V_{max}.

$$V_{max} = K'C^{\sigma} \tag{1}$$

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and $K = 0.59$ (from Appendix B, Table I)

$\sigma = 1.45$ (from Appendix B, Table I

$V_{max} = 0.59(2.0^{1.40}) = 1.61$ ft/s

c. Since V exceeds V_{max}, Region 1 (the linear region) is eliminated. To determine whether V lies in Region 2 or 3, the velocity at the onset of drag reduction, V_W, must be calculated.

$$V_W = 4.00\, C^{1.40}$$
$$= 4.00(2.0^{1.40}) = 10.56 \text{ ft/s}$$

d. V exceeds V_W, indicating that it falls in Region 3. The friction loss is calculated as that of water flowing at the same velocity.

$$(\Delta H/L)_w = 0.579\, V^{1.75}\, D^{1.25}$$
$$= 0.579(12.22^{1.75})(6.065^{-1.25})$$
$$= 4.85 \text{ ft head loss/100 ft of pipe} \quad (4)$$

This will be a conservative estimate, as the actual friction loss curve for pulp suspensions under these conditions will be below the water curve.

Digester Pump Applications Digester pump services encounter severe operating conditions and, therefore, require more consideration than some other pump services in the pulp mill. Two types of digesters are encountered in the pulping operation, continuous and batch. As both systems have similar pressure and temperature controls to produce pulp, the pump applications have similar characteristics.

Typical pressures encountered are the digester pressure that will be in the 250 to 300 psig range. The temperature range will be in the 200 to 350°F (93 to 177°C) range. The pumps that see these temperature and pressure ranges should be centerline-mounted design (Figure 12). The product pumped will include white liquor, black liquor, and pulp fiber depending on the location in the process.

The normal pump material is duplex stainless steel (such as ASTM A890 Grade 3A) due to the operating pressures encountered and the need for resistance to abrasion on many of the applications.

The operating conditions may result in severe pipe loading from thermal expansion. Some of the services may also encounter cyclical operations with the pump running throughout its curve thus creating high radial loads on the power end. These services may

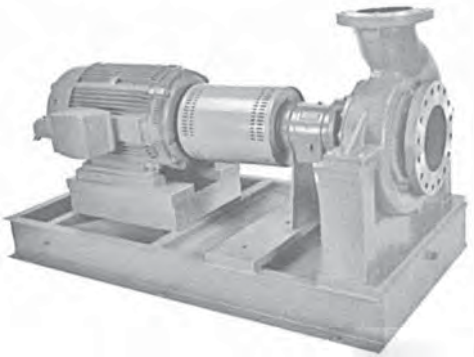


FIGURE 12 Digester pumps with centerline mounting (Courtesy of ITT/Goulds Pumps, Inc.)

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also encounter some abrasives from sand carry over or the presence of pin chips and fiber from incomplete pulping operations. Another consideration on some of the digester pump applications is the NPSH available. Elevated operating temperatures and high flow rates involved in some circulation pump services coupled with limited suction head may create cavitation problems.

These operating characteristics may create high maintenance needs so pump selection should focus on operating at the slowest speed possible, properly sizing the pump for the duty conditions experienced and allow for a higher pressure and temperature operation with centerline mounting and enclosed impellers (Figure 13). Finally, considering variable speed drives to address large pump flow rate swings will enhance long-term pump life.

Medium Consistency Pump Services Paper stock pulp in the consistency range of 8 to 16% is referred to as medium consistency pulp. Pulp fibers become a very strong interlocked network and create high friction between the fiber and pipe wall. The fluid characteristics of pulp are clearly non-Newtonian. Traditional methods to pump pulp in the 8 to 16% consistency ranges began with positive displacement pumps and were extended to centrifugal pumps over the past 25 years.

The non-Newtonian fluid characteristics, both within the pump and with respect to friction loss characteristics in the pipe system, along with air content of the pulp, present specific challenges for either positive displacement or centrifugal pumps.

Perhaps the first question to address is—why a medium consistency centrifugal pump versus a positive displacement unit? Some of the advantages offered by centrifugal pumps over positive displacement pumps are:

- Lower installation cost
- Lower maintenance cost and significant ease of maintenance when required

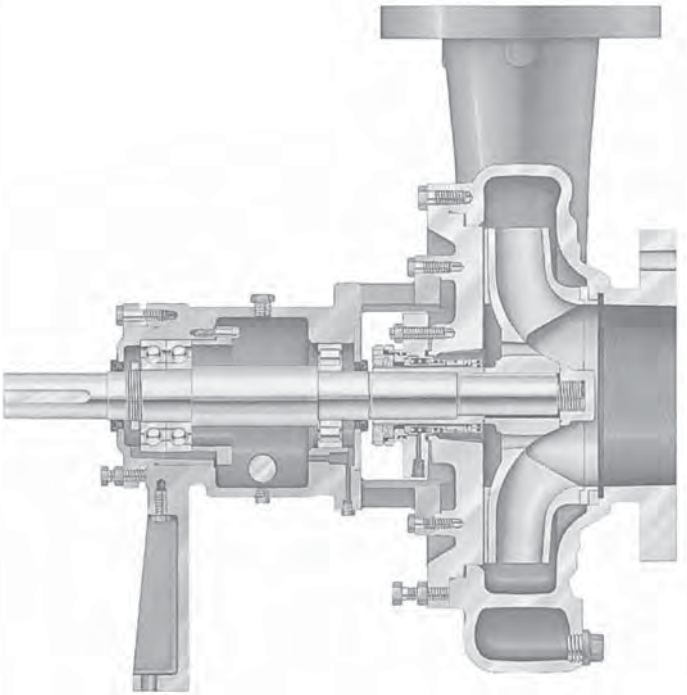


FIGURE 13 Digester pump with enclosed impeller (Courtesy of ITT/Goulds Pumps, Inc.)

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- Removal of air from pulp which can be detrimental in further unit operations such as chemical mixing in bleach plants
- Hydraulic capability to develop higher pressures often required in more modern pulping and bleaching systems

For effective centrifugal pumping of pulp suspensions in the 8 to 16% range there are three key areas to success. They are:

- Delivery system to the pump
- Air removal
- Simple control scheme

Delivery System to the Pump Figure 14 represents the most common configuration of medium consistency pump installation. The standpipe arrangement receives pulp from the stock source and provides a self-contained entity to accomplish delivery of pulp to the pump and the means to control the unit. Common sources of stock to the medium consistency pump can be a vacuum or pressure washer or disc filter.

The medium consistency pump is sized from a flow standpoint based on the required production rate of the process. Production rate changes translate into flow rate changes for the medium consistency pump. The table below illustrates the pump flow rate as a function of production rate with pulp consistency held constant.

The relationship of flow rate to production rate is:

$$Q = P \times 15/C$$



FIGURE 14 Medium consistency pump standpipe arrangement (Courtesy of ITT/Goulds Pumps, Inc.)

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where

Q = flow rate (GPM)
P = production rate (air dry short tons/day, adstpd)
C = pulp consistency (bone dry %)

Flow Rate at Different Production Rates

Production Rate, P (adstpd)	% C	Flow Rate, (GPM)
1000	12	1250
750	12	938
500	12	625

A last important aspect of the pump suction system relates to the pulp temperature. Typically, operation up to 170°F (76°C) can be accomplished in a normal 14 to 15 in tall standpipe. This will allow for 4′ to 6′ of suction head of pulp above the pump centerline. As pulp temperature increases, then the suction level above the pump must be increased to prevent flashing to vapor at the pump inducer or in the degas system. The increase in pulp operating level must be accompanied by a corresponding increase in standpipe height. On occasion in existing systems, there will be limitations which can present problems. For long-term, reliable operating performance this aspect should not be compromised.

Degas or Air Removal System This aspect of the pump system design is vital to successful operation of the pump. As many have experienced, the presence of even small amounts of air can reduce the effective performance of a conventional centrifugal pump. With medium consistency pulp suspensions between 8 and 16%, the air content encountered will be at least equal to the pulp consistency and often higher. Without air removal the pump operating control design will falter and flow rate reduction—or production rate tonnage from the operation viewpoint—will occur.

The air removal must be effective over a wide range of flow rates (production rates) and consistencies. For this to properly occur requires the functional integration of the design of the casing volute configuration, air removal holes, and air removal mechanism.

Control Scheme Figure 15 illustrates the control scheme normally used for a standpipe configured medium consistency pump. The importance aspect of the control scheme is the effective response to system changes. The system should be automatic and not require manual control or monitoring to make adjustments. The level control scheme is most common for

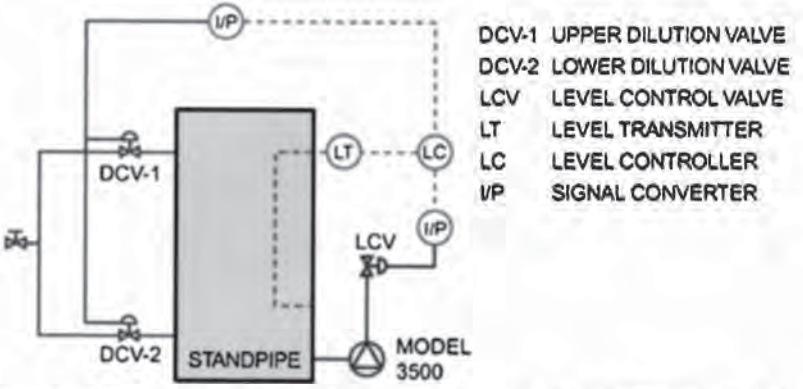


FIGURE 15 Medium consistency pump standpipe level/dilution system (Courtesy of ITT/Goulds Pumps, Inc.)

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standpipe design units. The unit operates to a set point level. Should the level deviate beyond the deadband range of the set point, control valve modulation will attempt to control to the set point. If the level change continues, then dilution valves open and the consistency is reduced. This allows the pump to regain control and return to the level set point. As the level returns to the set point, the dilution is reduced until it once again is off and the unit is controlled to the set point. Variable speed operation will differ somewhat for this method.

An example using a common situation that can arise would involve an increase in pulp consistency from the washer. Assume a unit is designed to handle 12% consistency pulp. If the consistency increases quickly to 14%, the system TDH would begin to increase dramatically. This is because consistency has, by far, the greatest impact on friction in the system. As the level moves rapidly outside the deadband (usually +/−10%), the pump moves back on its curve to develop sufficient TDH to deliver pulp through the system. Modulation of the level control valve when the production rate is near its design may not allow the unit to regain control. Sequencing of valves in the standpipe begins reduction of the pulp consistency to reduce system friction. This allows the pump to increase the flow rate. As the flow rate through the pump increases, the standpipe level begins to reduce and the standpipe level returns to its set point. As the consistency of the pulp being delivered to the pump returns to its design range, the system returns to the normal control mode and dilution is stopped.

Fan Pump Applications The importance of minimizing pressure pulsations in a headbox supply system has been emphasized with the inception of the hydraulic headbox to achieve higher paper machine speeds. This type of headbox can be used on twin wire or conventional fourdrinier paper machines, on paper grades ranging from corrugated mediums to tissue paper. Experience has shown that rotating equipment can be a source of pressure pulsations at the headbox, resulting in excessive machine direction basis weight variation (density variation) and thus poor quality paper as well as paper machine operation problems.

Pressure variations at the headbox will be either periodic or random. Operation with conventional airpadded headboxes was designed to control the periodic pressure variation. There was less emphasis on the random variation which often masked many periodic variations. Fan pump impeller specifications requiring impellers to have split, staggered, and skewed vanes evolved to increase the frequency, and reduce the magnitude of periodic pulsations at the vane pass frequency.

From a pulsation viewpoint, the fan pump is one component of the stock delivery system which deserves concern. However, other factors closely related to the installation of the fan pump also deserve careful consideration.

LOW PULSATION IMPELLERS Recent emphasis on minimizing pressure pulsations in fan pumps is placed on those factors which create pressure variations at low frequencies (i.e., 1 × rpm) rather than vane pass frequencies since these are above the range of concern for most hydraulic headboxes. One pressure pulse associated with fan pumps is the vane pass frequency pulse.

To reduce this pulsation level, pump manufacturers produce double suction impellers with split, staggered, and skewed vanes, thus reducing the magnitude of the pulse and increasing the frequency by having more vanes passing each cutwater per revolution. The vane pass frequency pulse is generally the largest magnitude pulse produced by the pump.

However, due to friction resistance within the system and the damping characteristics of the headbox, the vane pass frequency pulse rarely creates pressure pulsation problems. Due to its relatively long wave length, the low frequency (1 × rpm) pulse will not be dampened and can pass much more readily through the hydraulic headbox. This results in pressure variations at the headbox of a magnitude which results in unacceptable machine direction basis weight variations on the sheet. Control of the 1 × rpm pulse requires special attention to the design criteria and manufacture of the impeller. The types of nonuniformities which can create a 1 × rpm pulse include balance, singular surface irregularities, surface eccentricities, and surface finish.

Pressure pulsation requirements focus on the frequency range from 0 to approximately 25 Hz. The (1 × rpm) frequency of a fan pump is obtained by dividing the pump shaft rpm by 60 to obtain the frequency in hertz. A fan pump will only cause pressure pulsations at integer multiples of its shaft speed. Maximum pressure pulsation levels will depend on the headbox manufacturer, but they typically range from 0.15 to 0.25 psi (peak to peak), or greater.

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Other Design Considerations While the prime concern of the pump manufacturer is to minimize sources of the 1 × rpm pulse within the pump, external factors can influence the pump output or result in undesirable system effects when acting in conjunction with the headbox supply pump. To minimize potential pulsation problems, careful attention must be paid to the total system including the pump suction piping.

A common fan pump installation will utilize a suction transition piece between the wire pit silo and the pump suction. Two or more recirculation lines may enter this transition piece. These flows entering into the main body of the transition piece may be unequal. If there is a wide disparity between the flows from the recirculation lines, one side of the double suction impeller may be starved and the other side overloaded. In addition to increasing pressure pulsations at the pump discharge, such unequal loading of the impeller could cause pump operational problems.

Recirculation lines can be utilized for headbox overflow, primary and secondary cleaner accepts or rejects, rotary screen rejects, or thick stock addition for basis weight control. Generally, pressure pulsations can be transmitted along these lines to the pump suction. The fan pump may dampen these variations but will not completely eliminate them. It is also possible for pressure pulsations, caused by throttling valves in these lines, to pass directly back through the recirculation line and appear at the headbox since, as fluid-borne noise, pressure pulsations travel in all directions.

To minimize the pressure pulsations within a system, there should ideally be no recirculation lines entering the pump suction transition piece. All return lines should recycle directly into the wire pit silo which provides a large mass with a free surface to dampen out any pressure variations transmitted in the recirculation lines. A less desirable alternative would have the recirculation lines enter on the vertical centerline of the transition piece at a 45° angle. If thick stock is added, this line should be located closest to the pump. Recirculation line velocities should be a maximum of 1.1 to 1.5 times the velocity in the transition piece at the point of entry. The branch line nearest the pump should be no closer than a distance equal to twice the pump suction diameter.

DISCHARGE PIPING The factors which should be considered when designing the discharge piping of a fan pump include, but are not limited to, elbows located at or near the pump discharge, throttling valves, and piping support in the vicinity of the pump. Bends or 90° elbows close to the pump discharge can contribute to pressure pulsations and should be avoided.

A throttled valve can create pressure variations at frequencies similar to the fan pump frequency. In this situation, amplification of existing pressure pulsations can occur. Poorly supported discharge piping, which is susceptible to vibration, may amplify existing pressure pulsations in a stock delivery system.

PUMP DRIVE SYSTEMS Experience has shown that resonant system conditions often occur in stock delivery systems at, or near, normal fan pump operating speeds. It is, therefore, desirable to operate a pump at a different speed in order to move the pump frequency away from a system resonant frequency. Additionally, the existence of a system resonant condition can be ascertained by varying the pump speed and determining if the frequency of the pressure pulsations varies directly with pump speed (pump induced), or remains at a constant frequency regardless of the pump speed (system resonant condition). When constant speed drivers are used, the pump frequency cannot be varied; this makes it difficult to accurately determine sources of pressure pulsations as well as to avoid problems should they occur. For this reason, serious consideration should be given to a variable speed drive arrangement. An additional benefit of a variable speed drive system is the ability to closely match the system head with the fan pump performance curve. This allows high pump efficiency to be maintained over a wide range of operating conditions. Also, the discharge throttling valve can operate in a more desirable range and may even be eliminated.

OTHER SYSTEM COMPONENTS Pressure screens will often be used in a stock delivery system to remove foreign matter and provide uniformity to the stock. This is done by passing the stock stream through a screen at a low pressure (3 to 7 psi) differential. Materials

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which do not pass through the screen are removed by rotating foils or scraper blades. When these foils pass the screen discharge, a pressure pulse may be generated.

Consideration should be given to the relative operating frequencies of the pump and screens. It is possible that, if the screen frequency is close to other rotating equipment frequencies, reinforcement of existing pressure pulsations can occur. It is suggested that all pump operating frequencies be separated from the screen foil frequency by at least 20%.

Pressure Pulsation Testing To evaluate the pressure pulsation characteristics of a fan pump, pressure transducers and a spectrum analyzer are required. The pressure transducers are designed to measure variations in system pressure at the point of installation within the system. Transducers will normally be installed after the process equipment but may also be located near the inlet. The key to meaningful analysis of pressure pulsation data is the spectrum analyzer. This device performs a frequency analysis of the pulsation data, separating the signal from the transducer (amplitude vs. time) into its various components (amplitude vs. frequency). The vibration analyzer breaks down components of vibrations by individual frequencies or frequency bands.

This analysis can be conducted in both a factory and a mill environment. It can be expected that in a mill environment, additional pressure variations would be seen. Examples would be screen foil frequencies, valve cavitation, reflections from piping elbows, mechanical vibrations, and system resonances. A complete analysis using pressure transducers located throughout a headbox supply system would enable overall effects of different components to be determined, as well as demonstrate interaction of system components.

Recycle Service Applications Pumping recycled fiber presents several unique challenges that other paper services do not. This is the result of the shorter fibers that tend to dewater easier and the inherent contaminants contained in the bundles of recycled products used to start the process. The number one pump problem in the recycle paper process is plugging at the impeller eye. This plugging is due to large contaminants (strips of tape, wire, cans, and so on). The solids start to build up at the eye of the impeller in two significant ways, first, a general clog/plug between the vanes where the passageways are the narrowest. This is the result of simply too many solids trying to get through too small of a passage. Second, plastic straps or banding can be looped over or caught on the leading edge of a vane. If this banding doesn't get cut by the impeller or manage to slide off the vane's leading edge enabling the pump to clear itself, these contaminants will eventually rope together and create a plug in the suction of the pump.

The worst fibers are old corrugated containers (OCC). In addition to having a high content of contaminants, a high filler content (clays) is also present. The fillers sometime bind the contaminants together, making them larger. In addition, the course OCC fibers and the clays used as fillers are quite abrasive. As a result, OCC fibers and their inherent contaminants cause higher wear than most recycle fibers such as newsprint. The easiest fibers to pump are de-inked newsprint. There is no impeller plugging problems usually associated with these fibers.

Listed below are the names of some of the most difficult services:

	Pulper Pumps	Screen Pumps	Rejects Pumps
	Hydropulper	Primary Secondary Tertiary Slotted Hole Course Fine	Primary Secondary Tertiary
Comments:	Initial pump in system Sees all fibers/ contaminants from recycle source	Initial group of 6 to 10 pumps in system	Contaminants and fillers tend to concentrate in the rejects system

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When selecting pumps for a recycle fiber project it is always important to know:

1. What type and size of the contaminants the pump may see.
2. The service names of the pumps. They can give you an idea of the type of fluid and solids might be encountered in the application.

Note: Just because the data sheet may say 2%, there may be contaminants as well. Depending on the type and size of these contaminants, use of a recessed impeller pump or a modified paper stock pump may be appropriate. Modifications may include an opened impeller eye, increased casing/sideplate-to-impeller clearance, or a slotted sideplate.

RECESSED IMPELLER (VORTEX) PUMPS These pumps should be used on long stringy solids like tape, plastic, and wire (see Figure 16). They should also be used on the first pump in the system (usually called the hydropulper pump).

MODIFIED STOCK PUMPS A modified stock pump can be used on some of these applications. Typically, a three or 2/4 vane impeller is used in conjunction with a slotted sideplate (see Figure 17). The minimal number of impeller vanes allows a larger solid size to pass and the slotted sideplate will act as a shearing device, cutting the straps or banding should they get caught between the impeller and sideplate.

A two-vane impeller could also be used, but depending on the installation and application, but don't be surprised if you see vibration levels in excess of 0.3 in/s.

Another reliable alternative is the use of an impeller specially designed with long stringy solids in mind. These impellers have a wide open eye, a heavily raked vane leading edge, and a large impeller-to-casing/sideplate clearance (see Figure 18). These impellers are excellent at passing these long plastic scraps of banding commonly associated with recycled fiber. Any stringy solids caught on the leading edge of the vanes will slide off the impeller and into the process with being permanently caught on the impeller causing a roping plug.

Another modification that can be helpful is slotting the backplate of a dynamic seal. Recycled fiber has a tendency to clump up and dewater, especially in the dynamic seal area. The addition of the slots in the backplate has been proven through experience to create additional turbulence behind the repeller (see Figure 19). This additional turbulence

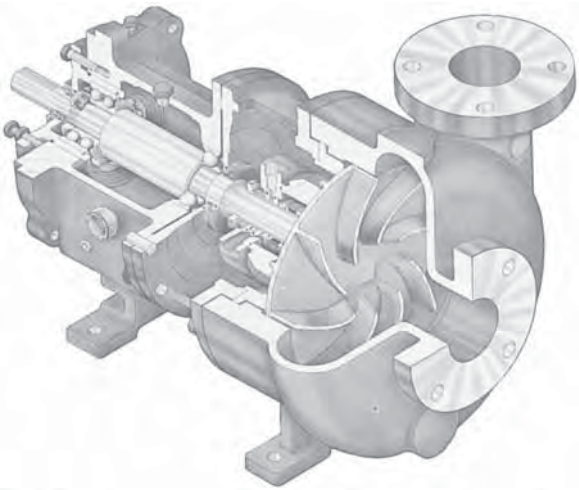


FIGURE 16 Recessed impeller (vortex) pump (Courtesy of ITT/Goulds Pumps, Inc.)

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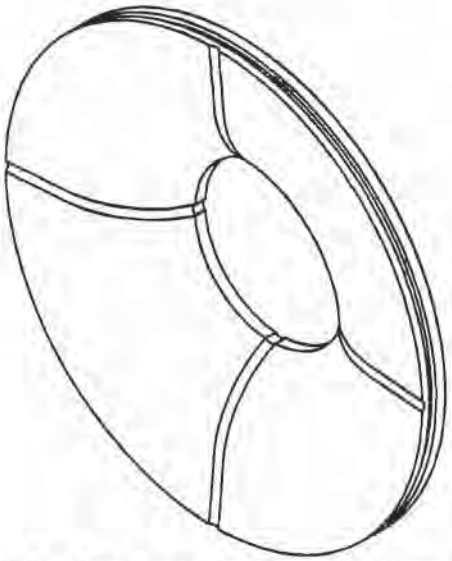


FIGURE 17 Slotted sideplate (Courtesy of ITT/Goulds Pumps, Inc.)

keeps the stock moving and prevents it from getting stuck between the repeller and backplate and quickly wearing the backplate.

Because of the high content of contaminants and fillers found in some recycle fibers the wear rate will be higher than on normal virgin fibers. To combat the wear and extend pump life, it is recommended that either 316 ss as a minimum, or preferably a form of duplex stainless steel be used for its increased hardness and greater resistance to wear.

Air Handling Applications Pump applications in many industrial processes involve handling liquid and gas mixtures. The entrained gas may be an essential part of an industrial process, or it may be unwanted. The pulp and paper industry, for example, injects

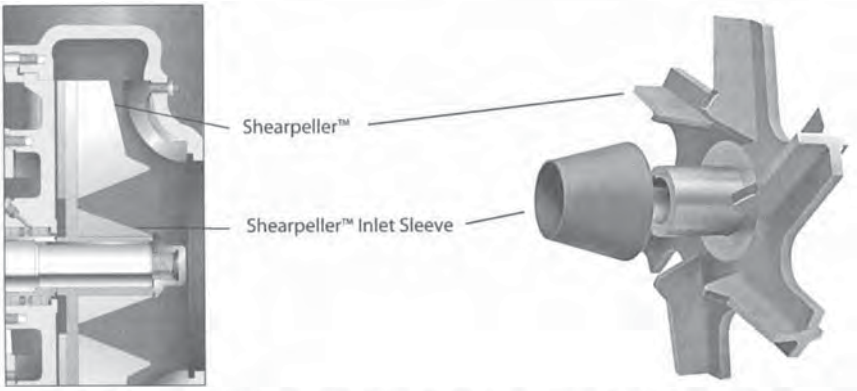


FIGURE 18 Impeller—sideplate design for long, stringy solids (Courtesy of ITT/Goulds Pumps, Inc.)

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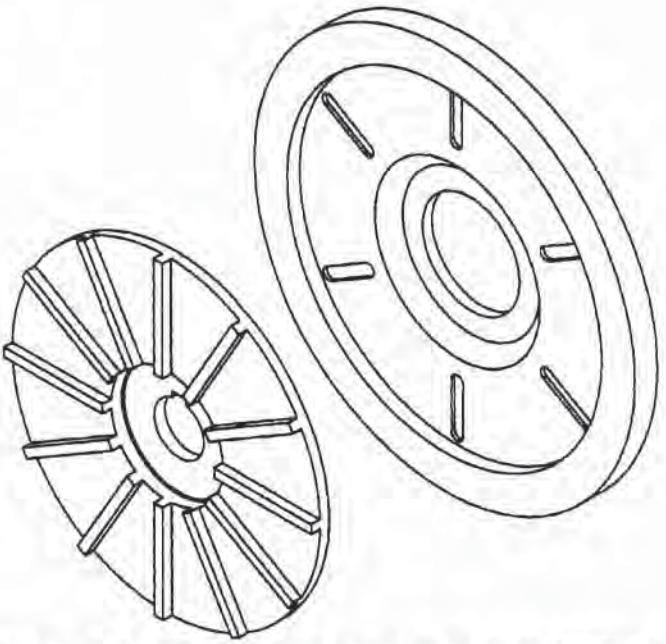


FIGURE 19 Slotted backplate provides turbulence to keep stock moving (Courtesy of ITT/Goulds Pumps, Inc.)

4 to 10% air into a dilute recycled pulp slurry as part of the ink removal process in a flotation cell. Many chemical and petrochemical processes also involve pumping a two-phase flow. Unwanted entrained gas can result from excess agitation or vortexing due to inadequate submergence on the suction of a pump.

The proper selection of a centrifugal pump for liquid and gas (two phase) mixtures is highly dependent on the amount of gas and the characteristics of the liquid. The presence of entrained gases will reduce the output of centrifugal pumps and can potentially cause loss of prime. Depending on the severity of the conditions, there are several possible solutions to this pumping challenge. This section will discuss these possible solutions, their limitations, and applications.

Ranges of Air Entrainment The first range is 0 to 4% entrained air. In this case, the size of the impeller must be increased to account for the effects of the air. This case would apply when designing a new system and the air content is known to be 0 to 4% or when trouble shooting an existing system.

The second range is for 5 to 10% entrained air. To account for the decrease in performance of the pump due to the air, the size of the impeller must be increased and may also require a larger size pump all together. This range also requires increased clearances between the impeller and casing/suction sideplate. Again, this is used in new systems or when solving existing field problems.

The third range is for 10 to 20% air. The best way to handle this case is to use a specially designed pump intended to handle the entrained gas. This would typically be a recessed impeller (or vortex) pump. An inducer may also be used on a modified conventional pump to boost its air handling capability above the normal 10% limit.

The final case is the use of an inducer. Typically, the inducer is thought of more as a problem solver for existing systems as opposed to part of a designed system. However, as inducers become more popular, it is becoming more common to see them used as part of an original system design.

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RANGE 0 TO 4% AIR Most centrifugal pumps have the ability to pump a small amount of entrained gas without major changes. However, impeller diameter corrections must be made to account for the loss of performance caused by the gas bubbles. The gas bubbles occupy space that would otherwise be occupied by the fluid. Consequently, the head output of the pump at a given flow is reduced. Increasing the impeller diameter will account for this loss of performance.

When air moves into the pump, the gas bubbles expand as they enter the eye of the impeller. This is because the fluid picks up a great deal of speed from the impeller. As the fluid speed increases, the pressure in that area of the pump decreases. With the decreasing pressure, the air bubbles are given a chance to expand. When the air concentration exceeds 4%, the air bubbles will accumulate in the low pressure, suction area of the pump; eventually the air will choke off the pump, and the pump will lose its prime.

Checking Figure 20, you can see that the head loss due to entrained gas concentrations of 2 to 4% can be as little as 5% of the pump performance for 2% below 60% of best efficiency point (BEP) up to 20% for 4% at BEP.

RANGE 5 TO 10% AIR As the gas concentration increases over 4%, the bubbles start to gather at the suction eye. Increased sideplate/impeller clearances will help break up the bubbles, moving them through the pump. The increased clearances create recirculation from the discharge side of the impeller to the suction eye, which retards the formation of large gas pockets. However, as a result of the increased clearances, the pump's efficiency is dramatically reduced. This compounds the effect of decreased performance from the entrained gas. Consequently, the pump selection will most likely increase in size with a considerable impeller diameter correction. The pump selection must be made such that the rated impeller diameter after correction is greater than or equal to 85% of the maximum diameter.

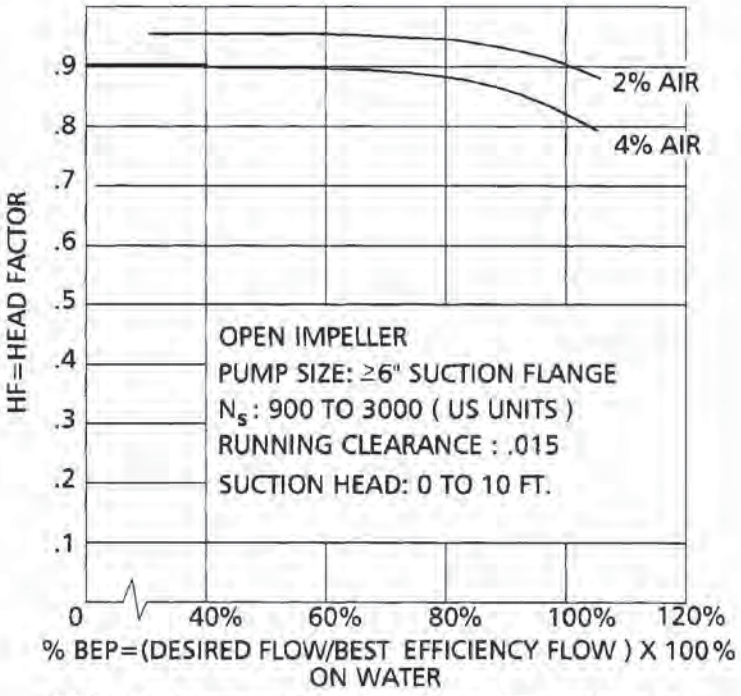


FIGURE 20 Head correction factors, 0–4% air (Courtesy of ITT/Goulds Pumps, Inc.)

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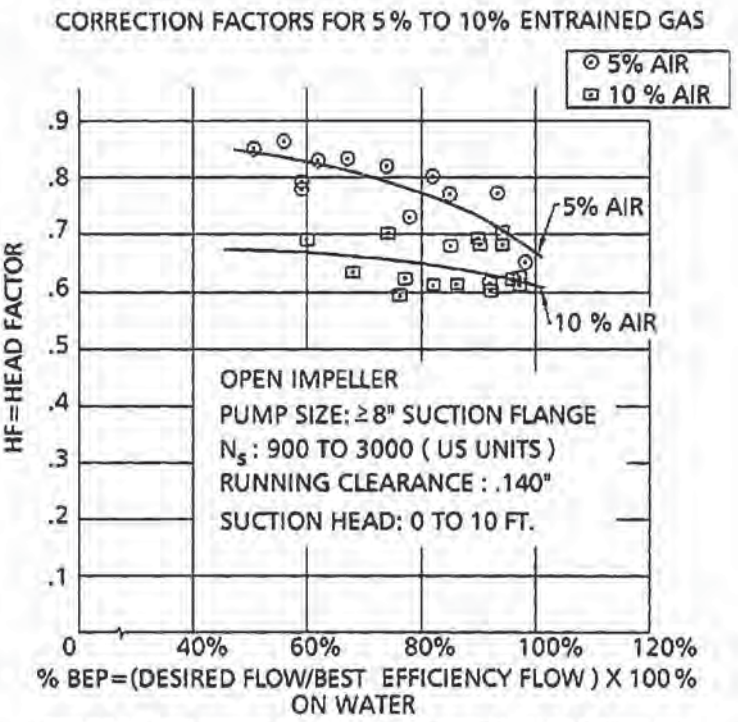


FIGURE 21 Head correction factors, 5–10% air (Courtesy of ITT/Goulds Pumps, Inc.)

Checking Figure 21, we see that the correction factors are in the 0.85 to 0.6 range. Meaning, with 5% gas at 50% BEP, you would experience a 15% loss in pump head and with 10% gas at BEP, you would see a 40% loss in pump head performance.

It is important to note that when using a conventional end-suction pump, whether with or without modification for air/gas handling, the rating/operating point should always be to the left of the BEP. The pump will not operate at or to the right of BEP because even with the increased impeller-to-casing/sideplate clearance, there will not be enough recirculation over the vanes to break up the collecting gas at the eye of the impeller.

RANGE 10 TO 15% AIR When gas concentrations exceed 10%, many times a recessed impeller pump can be used effectively. However, at these concentrations, the volume of air being pumped is rather large. Therefore, there is a very large head correction that needs to be applied for satisfactory operation.

Figure 22 shows the effect of entrained air/gas on a recessed impeller pump. As with the previous charts, the higher the air/gas concentration, and the higher the losses of the pump head performance. However, unlike the previous correction charts, you'll see here that pump performance does not worsen as the operating point approaches the BEP. To the contrary, performance actually improves. This is due to the pump design.

Inducers Inducers have several different uses and can be applied across all the ranges listed above. The primary function in an entrained air application is to agitate the air, keeping it in suspension and to increase the suction pressure into the pump. By agitating the air, it reentrains the air, increasing its pumpability and preventing the loss of prime due to the air.

A by-product of this action is the slight generation of head by the inducer. This head generation effectively increases the suction pressure into the pump. It has been shown in

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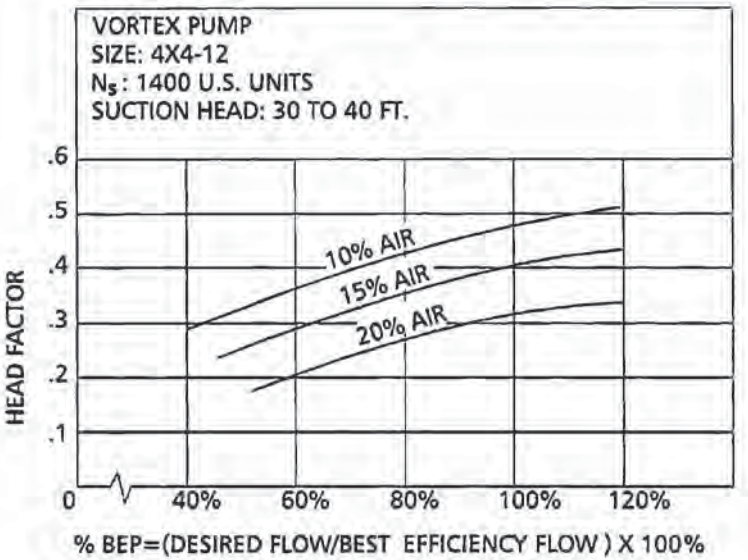


FIGURE 22 Head correction factors, 10–20% air (Courtesy of ITT/Goulds Pumps, Inc.)

testing that increased suction pressure greatly increases the pumpability of an air-entrained fluid. Because the inducer is generating head, albeit a small amount, it is doing work and, consequently, requires power input. All inducers will require some amount of horsepower input similar to a dynamic seal.

In all cases (0 to 4% air, 5 to 10% air and greater than 10% air), an inducer will reduce the correction factors used in sizing the pumps. With the decreased correction factors, at times smaller pumps can be used. The trade off for the benefit of the reduced corrections is increased power requirements and a somewhat reduced operating range. Instead of having the ability to run the pump from minimum flow to runout, it is now limited to a range of minimum flow to BEP.

There are two different designs that pump manufacturers commonly use. The first is the traditional style, Figure 23. The traditional inducer resembles a corkscrew. This design is used on clear liquids and is commonly referred to a flight inducer.

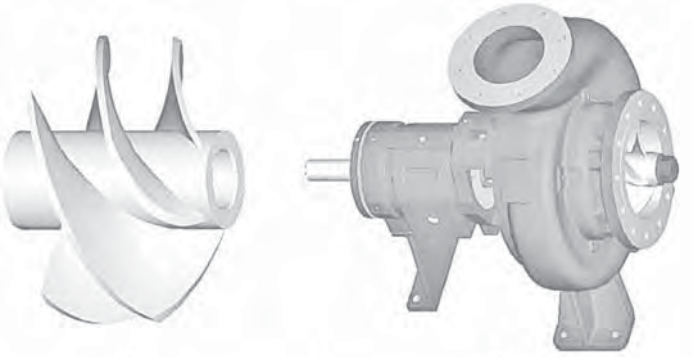


FIGURE 23 Traditional inducer design (Courtesy of ITT/Goulds Pumps, Inc.)

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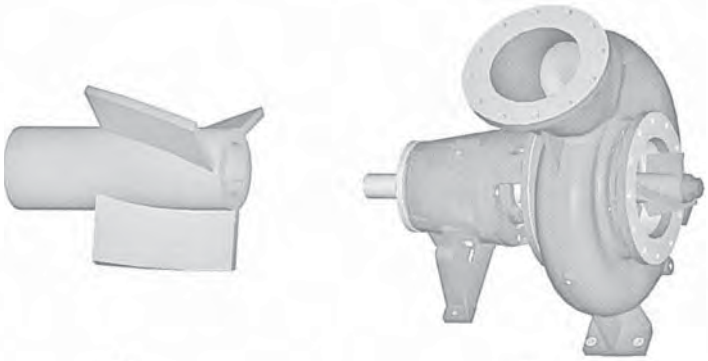


FIGURE 24 Feeder vane inducer style (Courtesy of ITT/Goulds Pumps, Inc.)

The second type is a feeder vane style, Figure 24. It has three blades or vanes that may appear as a mixer. The feeder vane inducer is used on pulp and nonhomogeneous liquids. The feeder vane inducer keeps the solids mixed and moving. It helps to retard plugging. Both styles require special design work for satisfactory performance.

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SECTION 12.10
FOOD AND BEVERAGE PUMPING

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JOHN DRANE

INDUSTRY STANDARDS

Pumps for use in food, beverage, and pharmaceutical industries must meet various hygiene standards. These standards define the materials of construction and the "cleanability" of the pump. Unlike other industrial applications, pumps for the food and beverage industries must meet rigid sanitation codes, know in the industry as the "3-A" standards*. These standards were originally established for the dairy industry by the following organizations:

1. The International Association of Milk, Food, and Environment Sanitarians.
2. The U.S. Public Health Service.
3. The Dairy Industry Committee, composed of representatives from the following:
   - American Butter Institute
   - American Dry Milk Institute
   - Dairy and Food Industries Supply Association
   - Evaporated Milk Association
   - International Association of Ice Cream Manufacturers
   - Milk Industry Foundation
   - National Cheese Institute
   - National Creameries Association

*Available from the International Association of Milk, Food and Environmental Sanitariums, Box 437, Shelbyville, IN 45176.

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In the U.S., the 3A standards are widely enforced. They are recognized internationally, but are not enforced to the same degree. The 3A standards are a self-certifying standard, and they consider design and construction features only. They do not require microbiological testing for cleanability.

In Europe, the "Supply of Machinery (Safety) Regulations" (1992) defines the safety and hygienic requirements for equipment to be used on agri-foodstuffs. This is legislative and is not a voluntary standard. Compliance with these regulations enables the CE Mark to be attached to the machine. Noncompliance means machines cannot be sold within the European community (EC). The European Committee for Standardization (CEN) will produce a standard (TC 197) for pumps for food use, in support of the "Machinery Regulations" that will eventually form the controlling standard for all pumps designed for food use in the EC.

An independent group, formed mainly from users of sanitary equipment (European Hygienic Equipment Design Group—EHEDG), has produced guidelines for all types of equipment, including pumps. This group acts in an advisory capacity to CEN and other standardization bodies. These guidelines are generally very onerous, but they are often specified by end users of hygienic equipment.

The principle requirements of hygienic pumps are that the wetted parts should be compatible with the products being pumped, and that the pump can be easily cleaned, either by dismantling or by clean-in-place (C.I.P.) processes. Externally, the pumps should be smooth and free from crevices where dirt could lodge and bacteria or insects could flourish.

Emerging standards issued by the American Food and Drug Authority (AFDA) and European Directives based on these standards specify suitable materials for contact with foodstuffs. They also define any restrictions that may apply to them and test methods to prove compliance. These standards are recognized internationally and are widely used.

PUMP DUTIES

Pumps are used for filling, emptying, transferring, dosing, and mixing. They are also used to convey the process fluid through plant items with a high resistance to flow, such as membrane filters and heat exchangers. In general, pumping should not harm any solids in the liquid, and delicate shear-sensitive products must be handled gently. The choice of a particular pump depends on consideration of a number of factors. These include capacity, delivery pressure, and suction conditions. The calculation of system pressure losses is discussed in Section 11.1.

It is important to consider the viscosity of the liquid at the actual shear rate and temperature appropriate for the system. Newtonian liquids have a constant viscosity with shear rate (for example, glucose). Nonnewtonian liquids such as tomato ketchup have a viscosity that is dependent on shear rate. Some liquids become less viscous with increasing shear rate; these liquids are called *thixotropic*. Examples of a thixotropic liquid are starch and molasses. Others, less commonly, become more viscous with increasing shear rate; these liquids are called *dilatant*. An example of a dilatant liquid might be some candy compounds. Most liquids also become less viscous with increasing temperatures.

The fact that viscosity changes with shear rate and temperature means that a careful assessment must be made of all fluids that we intend to pump. This assessment would normally produce three values of viscosity that must be considered in the pump and system design.

1. Viscosity in the storage tank (low shear)
2. Viscosity in the pipe work (medium shear)
3. Viscosity in the pump (high shear)

A shear-sensitive material, which will degrade with work, would normally need a positive displacement pump with low shear characteristics, pump speed being selected accordingly. If solids in the product are not to be damaged, a pump with suitable-sized spaces and

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cavities must be selected. In some cases, high shear rates will be advantageous to the process (for example, mixing of emulsions) and the pump and system can be selected accordingly.

The chemistry of the liquid will determine the compatibility of the pump materials. The viscosity of the process fluid and its rheological properties affects both the speed at which the pump can run and also the friction losses to be expected within the system. The friction losses will affect the power required to run the pump. Generally, the more viscous a product, the slower the pump will need to run. If the product is very viscous, larger suction piping may be required, or in some cases, special enlarged inlets are used with auger assistance.

If the product is abrasive or carries hard solids in suspension, again pump speed will often need to be reduced to give economic life to pump components. Often, pumps with resilient components or those that do not have fixed clearances are better suited for this service. If, however, the solids in suspension are liable to settle at low speeds, speed must be increased to keep the solids in suspension to protect component life.

The product temperature may affect the suction performance capability of the pump. Also, for high fluid pumping temperatures, the pump and pumping elements must be selected to take account of differential expansion or contraction resulting from temperature variations. In some cases, pump performance is limited because of liquid properties.

PUMP TYPES

Food and beverage pumps are divided into two generic types: rotodynamic and positive displacement. Both types are widely used in the food and beverage industries.

Rotodynamic Pumps Rotodynamic (centrifugal) pumps are widely used in hygienic applications (Figure 1) because they are simple and inexpensive, they give a smooth delivery, and they are relatively easy to clean. They are, however, limited by viscosity that will greatly reduce pump efficiency above about 500 centipoise, generally regarded as the limiting viscosity. As the pumps rely on speed of rotation to generate head, there are fairly high shear rates within the pumping elements that affect shear-sensitive products. Also, because it is not self-priming, it has a limited ability to handle entrained gases and vapors. A positive feature of centrifugal pump designs is that the performance varies according to the system operating conditions to which it is subjected. Unlike a positive displacement pump, the discharge of a centrifugal pump can be partially or fully closed without over-pressurizing the pump.

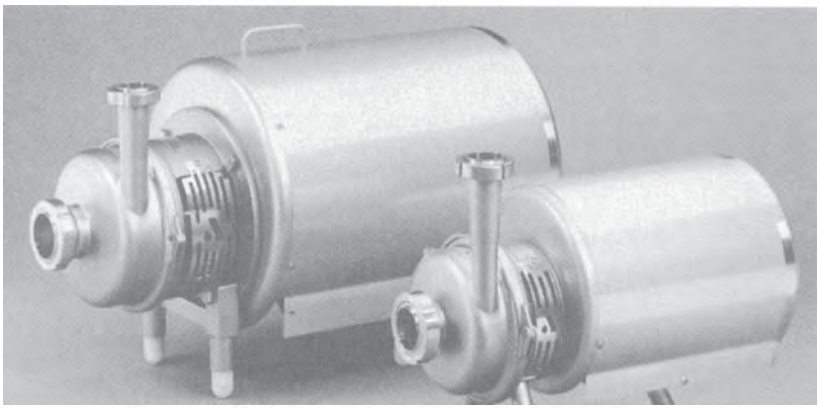


FIGURE 1 Hygienic centrifugal pump (APV Fluid Handling)

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ADVANTAGES These pumps are simple; they are of generally robust construction; pumps are easy to clean in place; pumps cannot be over-pressurized against a closed valve; they can be connected in series for increased pressure; and they can be connected in parallel for increased flow.

DISADVANTAGES These pumps typically are not self-priming; they are generally limited to about 500 centipoise viscosity; they are limited ability to handle fluids that are shear-sensitive; and they have a limited ability to handle high percentages of entrained gases.

Positive Displacement Pumps Positive displacement pumps are used in processes where viscosity limits the capabilities of a centrifugal pump or where the process needs to be versatile and the batches vary. They are normally self-priming and can handle gaseous products. However, all positive displacement pumps require pressure relief systems to prevent damage to the pump or system if a valve is closed or the outlet blocked.

Many types of positive displacement pumps are available and they are normally split into two groups—rotary and reciprocating, the reciprocating type needing a valving system to be able to operate. Valves increase shear rates and are susceptible to wear and blockage.

Gear Pumps

ADVANTAGES These pumps are self-priming; there is uniform discharge with little pulsing; the pumps are reversible; due to small clearances, little flow variation with change of viscosity or pressure.

DISADVANTAGES They can only pump clean fluids due to fine clearances and rubbing gear teeth; they cannot run dry; they need close tolerances to operate, so fits and alignment are critical.

Lobe Rotor Pumps (Figure 2)

ADVANTAGES These pumps can run dry for short periods as rotors have clearance, but run-dry time is limited by the seals; pumps can self-prime on low lifts; pumps can handle solids in suspension; there is little change of product velocity as it passes through the pump, so the pump has good net positive suction head (NPSH) performance, once primed; pumps can generally handle viscous materials; pumps are usually reversible.

DISADVANTAGES Fixed clearances can mean rapid performance degradation because of abrasive wear; pump has two shafts and associated shaft seals; suction lift limited at low speed/viscosity due to clearance of rotors.

Flexible Impeller Pumps (Figure 3)

ADVANTAGES Pumps are generally self-priming; pumps create relatively low pressure pulsations while pumping; pumps are reversible; pumps can handle gaseous fluids and fluids with some solids in suspension; pumps are easily cleaned.

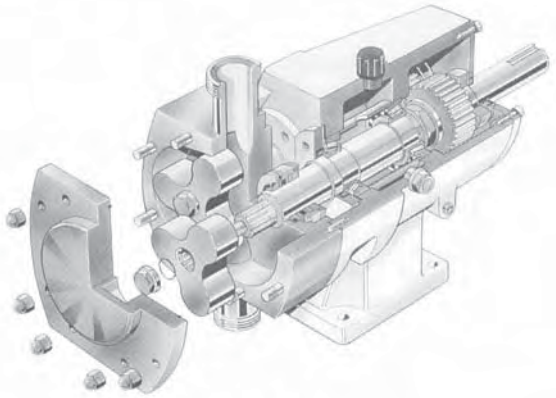
DISADVANTAGES Pumps cannot run dry; pumps are only suitable for low pressure applications; pumps are not suitable for abrasive services.

Progressive Cavity Pumps (Figures 4 and 7)

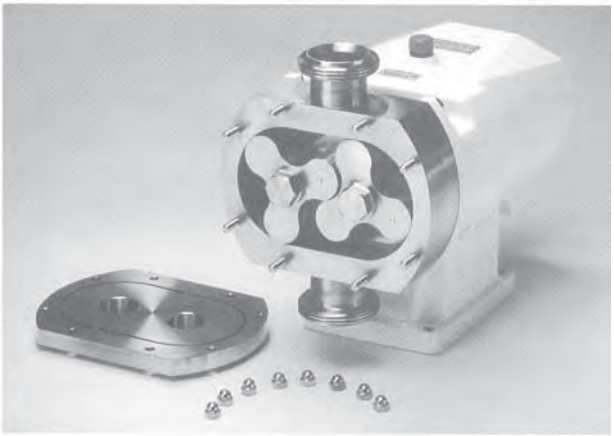
ADVANTAGES These pumps are self-priming; they exhibit uniform discharge with little pulsation; they are reversible and can handle gaseous fluids; they can handle solids in suspension and abrasives and can handle shear-sensitive fluids with little damage.

DISADVANTAGES The pumps cannot run dry.

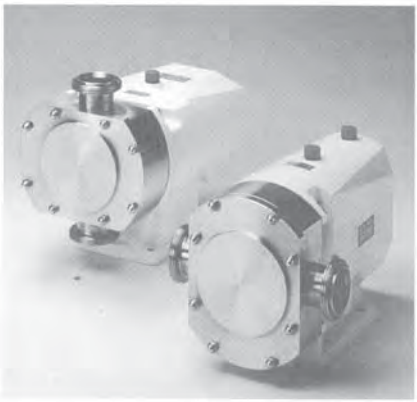
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A



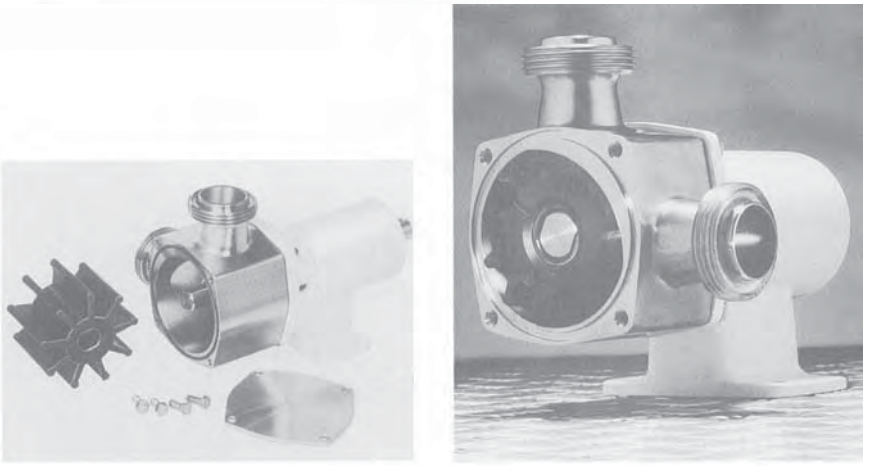
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C

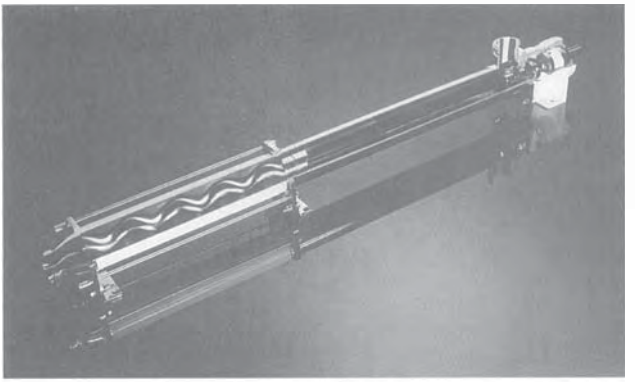
FIGURE 2A through C Rotary lobe pumps (Alfa Laval Pumps, Ltd.)

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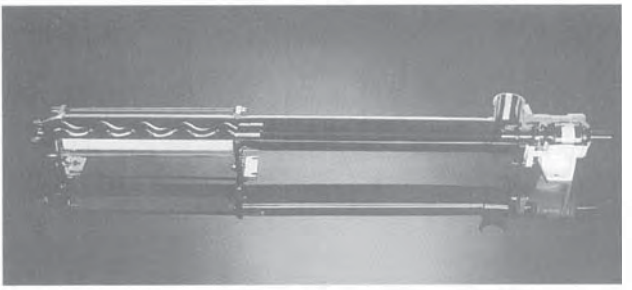


A B

FIGURE 3A and B Flexible impeller pumps (ITT/Jabsco).



A



B

FIGURE 4A and B Hygienic progressive cavity pumps (Mono Pumps, Ltd.)

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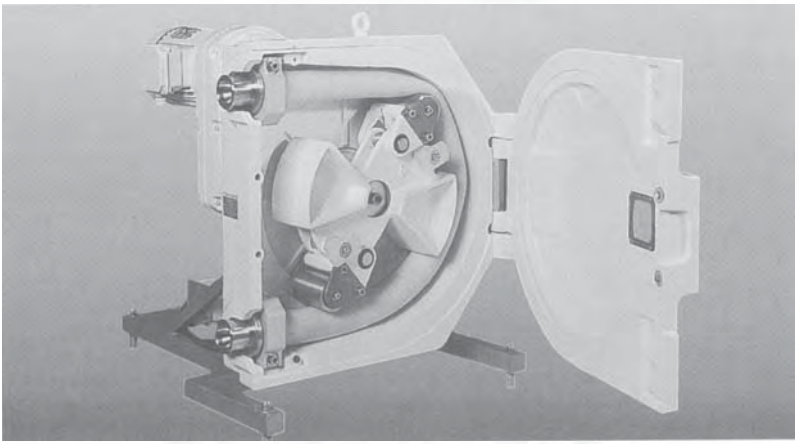


FIGURE 5 Hygienic peristaltic pump (Watson-Marlow, Ltd.)

Peristaltic Pumps (Figure 5)

ADVANTAGES Pumps are self-priming; they can run dry and are reversible; they can handle gaseous fluids and can pump abrasive material and solids in suspension; they have only one contact part with no seals.

DISADVANTAGES Tube failure means loss of product; at low rotational speeds, pulsation can be a problem.

Piston Pumps

ADVANTAGES Pumps may be used for very precise dosing; pumps can handle high pressures.

DISADVANTAGES Pumps are not reversible; they need valves to operate; and they generally have pulsing flow.

Diaphragm Pumps

ADVANTAGES Pumps can run dry for short periods; they are fairly easily cleaned; and they have no glands.

DISADVANTAGES Pumps are not reversible; they need valves to operate; and they have pulsing flow.

Mechanical Seals Pumps in the food and beverage industries are almost totally fitted with mechanical seals (see Figure 6). These seals must be simple in construction and designed in such a way as to facilitate cleaning. Cleaning in place (C.I.P.) is a frequent requirement, and mechanical seals for these pumps are specifically designed for this purpose.

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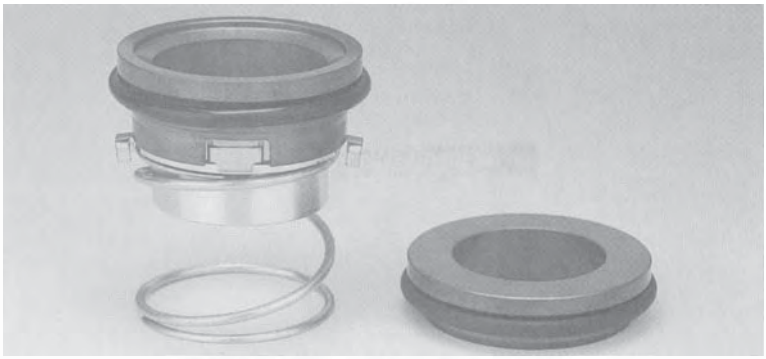
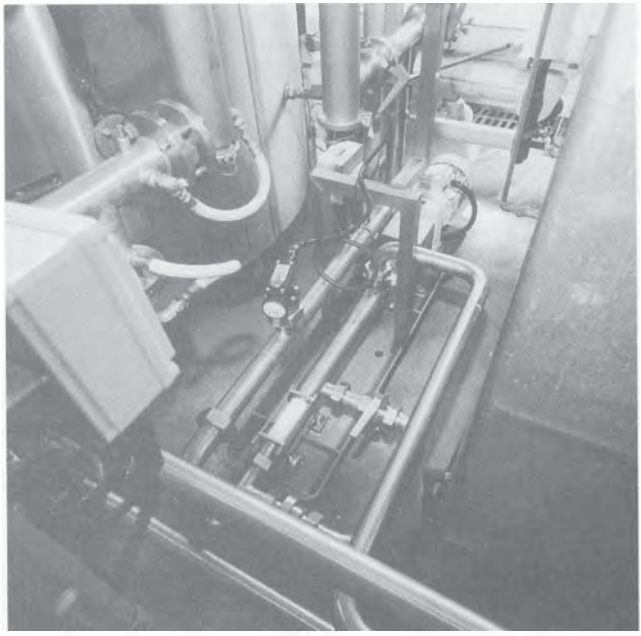


FIGURE 6 A hygienic mechanical seal (Roplan, Ltd.)

Pump Drives Standard electric motors, usually totally enclosed and fan-cooled, are used throughout the industry. A stainless steel casing is sometimes used to enclose the motor and simplify external cleanability. Pump units must be either grouted to the floor or mounted at least 2 in (50mm) above the floor to enable cleaning and eliminate any build-up of debris.

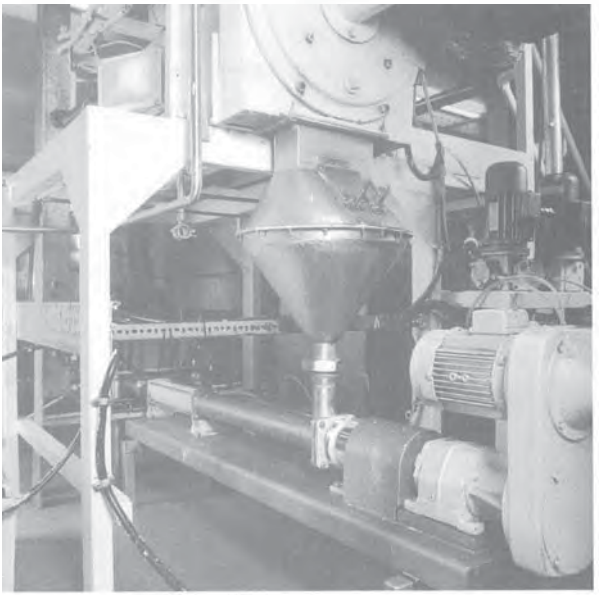
Figures 7A, B and C show typical installations of progressive cavity hygienic pumps.



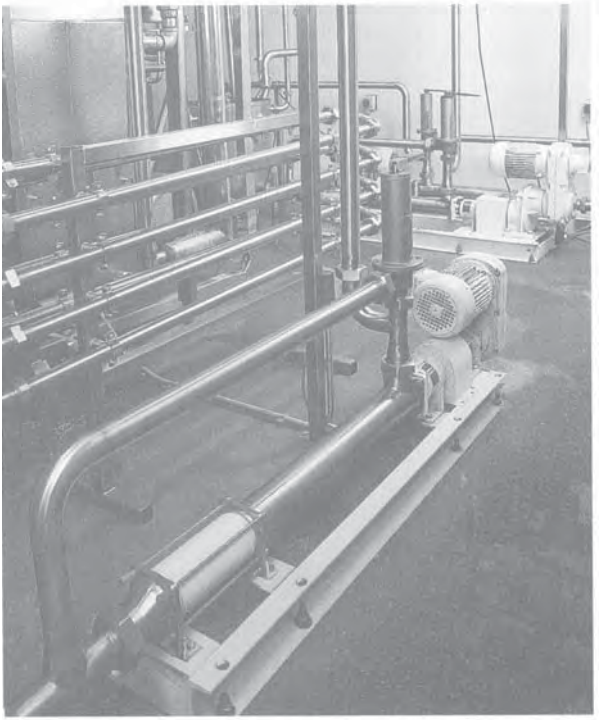
A

FIGURE 7A through C Typical installations of a progressive cavity hygienic pumps (Mono Pumps, Ltd.)

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B



C

SECTION 12.11
MINING

W. D. HAENTJENS

It might be asked why pumps used in mining services receive a separate classification when the types involved do not differ in principle or in general appearance from those used in fresh-water service. The answer is the need for utmost reliability and an ability to withstand both corrosive and abrasive waters. Not all mine waters are corrosive, but almost all mine waters from active mines are abrasive because of the suspended solids from the mining operations. Much can be done to limit the amount of solids handled, but the solution generally requires a compromise based on economics. Removal of most coarse solids is usually justified but removal of fine solids is generally impractical. Thus, a heavy-duty pump has evolved that warrants the classification of mine pump.

PUMPING CONDITIONS AND PUMP TYPES

Pumping conditions in mining services can be determined from a consideration of the types of mines and the material being mined. For example, the broadest category would be a division between open-pit and underground or deep mines. Open pit mines seldom exceed 600 ft (183 m) in depth so the pumping heads generally are not much greater than this unless very long discharge lines are required. For many open-pit mines, the greatest pumping load does not result from groundwater but from rainfall. Unless a certain increase in water level can be tolerated at the bottom of the pit, the maximum rainfall rate and the drainage area involved will determine the required pumping capacity.

As mining progresses, the pumps generally must move with the operator. This suggests the use of barge-mounted vertical pumps. A practical cost-effective way of achieving dewatering under this scenario is the use of a pond system with barge-mounted vertical cantilevered shaft pumps. Barge mounting adds a great deal of flexibility. Simple barges, as in Figure 1, illustrate the simplicity of this mounting. Except for occasional greasing of the bearings, there is little maintenance, as these pumps do not require a stuffing box or

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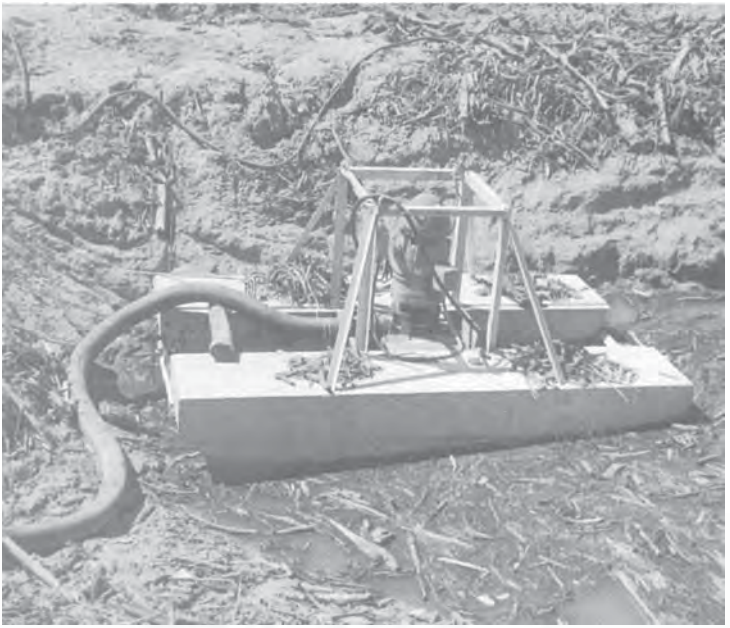


FIGURE 1 A simple barge (Hazleton Pumps, Inc.)

mechanical seal. Almost any size can be barge mounted. Figure 2 shows one of three 1500 HP (1119 kW) vertical cantilevered shaft pumps that are installed on one barge. Each pump is rated at 14,000 gpm (3180 m^3/h).

Where there is significant rainfall, the flow rates are generally large and the combination of large capacities with moderate heads may suggest a double suction pump. The selection of a double suction pump may permit the use of a higher speed pump; however, the selection may not be based on hydraulic considerations alone. An examination of hydraulic design for a double suction pump may show it is ideal for a barge-mounted vertical pump, providing there is ample water level in the pit. On the other hand, minimum water levels may be required to facilitate mining operations. If these conditions exist, which commonly occurs, selection of a top inlet single-stage pump may be better. Top inlet barge mounted vertical pumps can pump the water level down without drawing the mud from the bottom. This is the conflict, as the desirable features of the double inlet pump are lost.

Comparative features can be readily seen from an examination of the net positive suction head (NPSH) requirements. High-capacity moderate head pumps have a specific speed (N_S) which may require a greater NPSH; that is, a greater submergence. This may alter the pump length and the ability to utilize a cantilevered shaft pump. Such pumps are ideal for barge mounting if the NPSH (required submergence) conditions exist.

Figures 3 and 4 are reproductions of charts published by the Hydraulic Institute. An example will highlight the available design choices. Calculations may show that some combinations of flow, head, and pump speed are not feasible. This is an important consideration before proceeding with design. Assuming we have a possible floating pump station with a pumping requirement of 7500 gpm (1700 m^3/h) and a calculated discharge head of 300 ft (91m), Table 1 shows the calculation of the required absolute suction conditions. Converting from absolute pressure to submergence from the water level to the impeller centerline, under standard atmospheric conditions, the table shows how much submergence is required for a given operating speed. The table shows that a single inlet pump at 1800 rpm is not practical, and a pump designed for 1200 rpm is barely acceptable. A double

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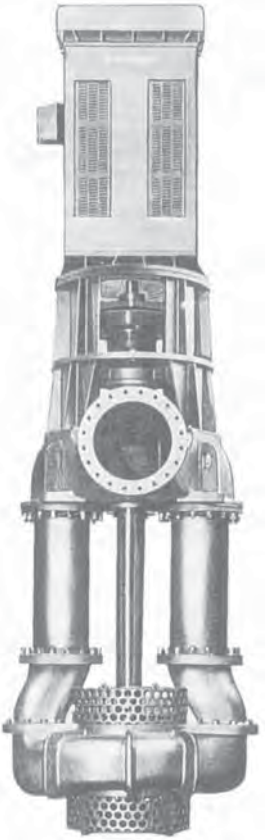


FIGURE 2 Vertical cantilevered shaft pumps used for barge mounting (Hazleton Pumps, Inc.)

suction pump at 1800 rpm is also barely acceptable if there is to be a cushion for low barometer days. A double suction pump at 1200 prm is fully acceptable. We are not locked into this selection, as a lower flow rate with more pumps may be fully acceptable at a higher speed.

The calculation for NPSH shows that the 1800 rpm double suction pump and the 1200 rpm single suction pump have about the same NPSH requirement. Although both are for the same hydraulic conditions, they are substantially different pumps. If the NPSH requirements can be met with a simple single inlet higher-speed barge mounted pump installation, a substantial reduction in cost and weight is possible.

Technically, other solutions are possible when the NPSH is marginal. If the NPSH requirement is only a few additional feet, it may be possible to lengthen the pump to increase the submerged depth. In comparing available NPSH and required NPSH, the altitude of the installation and the temperature of the liquid should be considered. Altitude correction can be simply applied as 1 ft (0.3 m) for each 1000 ft (305 m) of elevation in the calculation of NPSH available.

The development of large overhung vertical shaft pumps has been a boon to mining operations. Figure 5 shows the installation of four floats, each with a 400 HP (300 kW) top inlet cantilevered shaft pump. The type of mining operation frequently dictates the pump design and particularly the installation. Figure 6 shows float mounted pumps designed to

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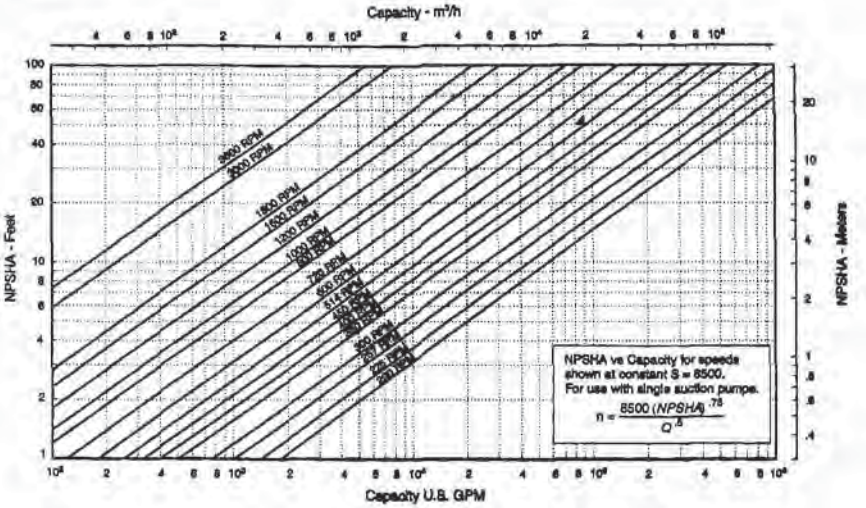


FIGURE 3 Recommended maximum operating speeds for single suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 1)

skim the clean liquid in a settling pond. The float construction acts as a weir so only the cleanest liquid is returned to the plant. The float mounts 250 HP (185 kW) top inlet vertical cantilevered shaft pumps. An interesting feature is the hydraulic thrust balancing by the use of opposed flexible hose discharge lines. Note that these forces also exist in steel pipeline and must be adequately restrained.

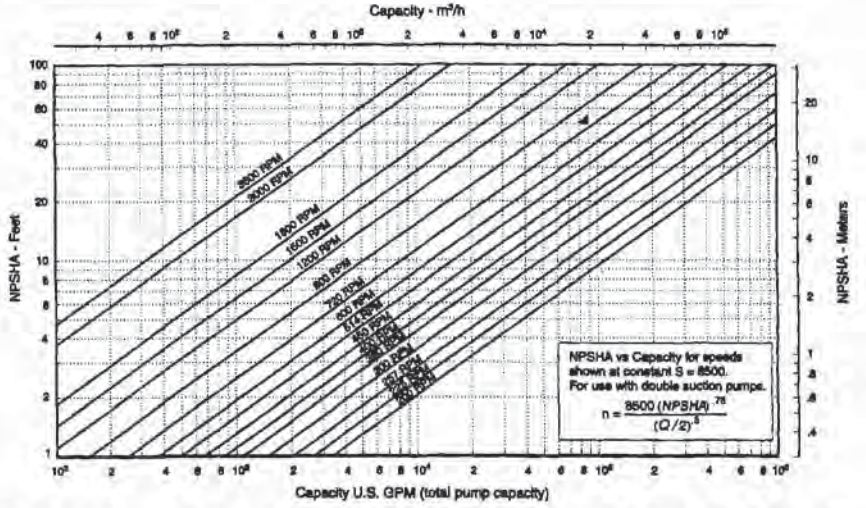


FIGURE 4 Recommended maximum operating speeds for double suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 1)

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TABLE 1 Calculation of required absolute suction conditions ($\Omega_s = N_s/2733$)

Pump Type	Operating Speed—rpm	Specific Speed, N_s	NPSH Required ft (m)	Submergence ft (m)
Single Suction*	1800	2163	48 (14.6)	18 (5.5)
Single Suction*	1200	1529	30 (9.1)	0**
Double Suction	1800	1442	31 (9.5)	1 (0.3)***
Double Suction	1200	1019	18 (5.5)	0***

*Can be top or bottom inlet
**Barely acceptable with submergence sufficient to prevent vortexing at inlet
***Very safe

Figure 7 illustrates an interesting installation of a float-mounted pump of unique design. The pump is of cantilevered shaft design (no submerged bearings) with the pump utilizing the motor bearings. Pump construction is shown in Figure 8. This is ideal for barge mounting. It involves a special motor, but the lower center of gravity permits a smaller barge and readily compensates for the extra cost of the motor. No submerged shaft bearing is required, so dirty water can be handled with little maintenance. A double discharge design also provides balanced hydraulic side thrust and eliminates the need for a shaft seal.

If floats are to be used for pump mounting, considerations in addition to hydraulics must be examined. Floats must be designed for safe operation in all circumstances. Floats for individual pumps are simple and readily moved about a mining operation; however, there are other factors beside the weight of the pump and motor. A portion of the discharge line (usually a hose on floats) must be supported by the pump float. Consideration must be given to the possibility that a number of workmen might gather at one side of the float and overturn it. Stability calculations must be made by considering

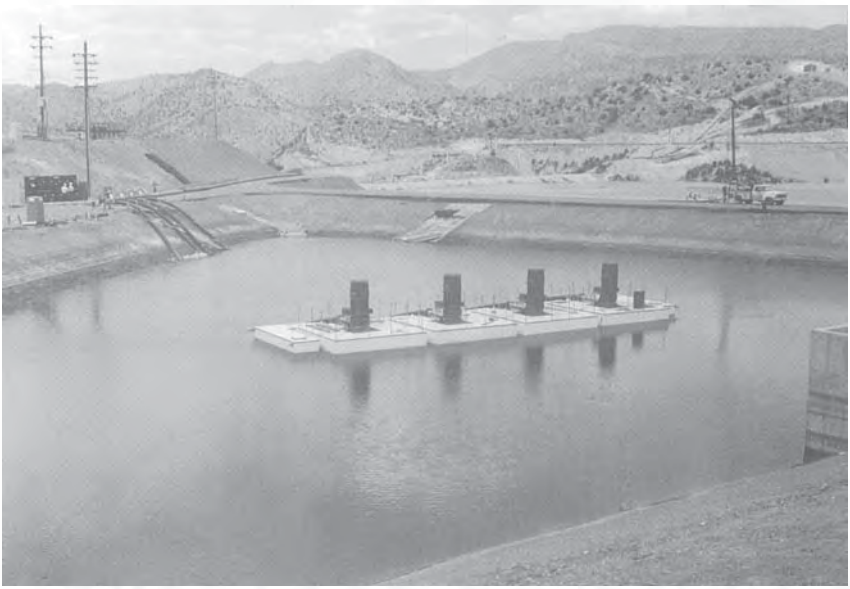


FIGURE 5 An installation of four floats with top inlet cantilevered shaft pumps (Hazleton Pumps, Inc.)

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FIGURE 6 Float-mounted pumps designed to skim the clean liquid in a settling pond (Hazleton Pumps, Inc.)

lateral rotation of the float and the uprighting forces due to the additional buoyancy on one side (or end). The center of gravity must be within the center of buoyancy. If the center of gravity moves beyond the center of buoyancy, the float will overturn. It must also be remembered that, after a pontoon is submerged, that is the limit of buoyancy. In addition to these factors, wind effects must be considered. If the climate is not too severe,

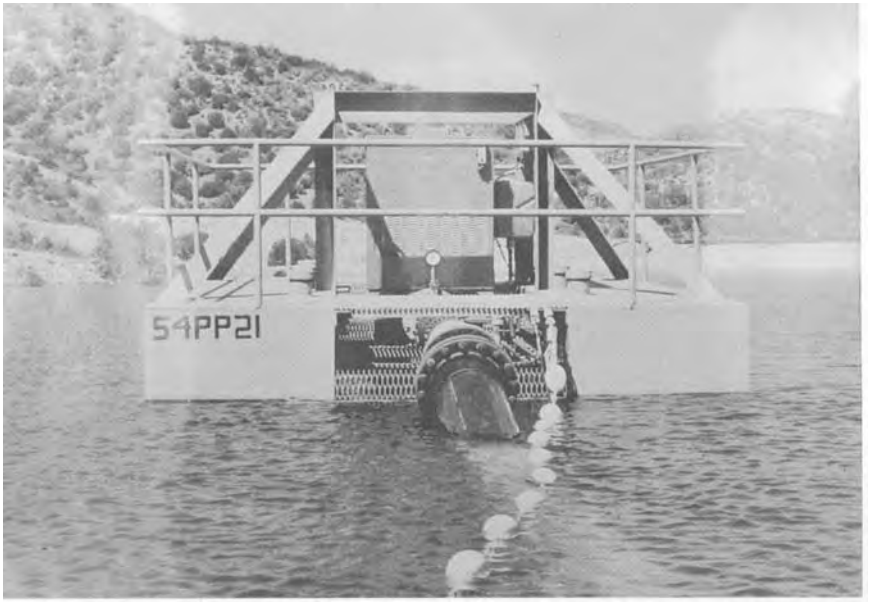


FIGURE 7 Unique design float-mounted pump (Hazleton Pumps, Inc.)

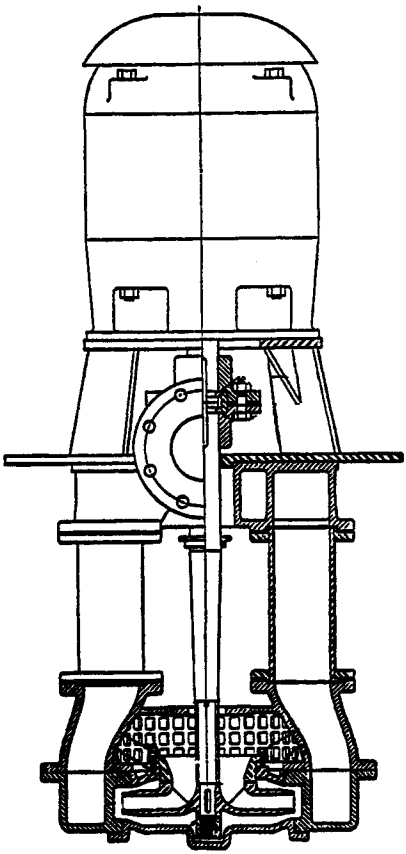


FIGURE 8 Cantilevered shaft design utilizing motor bearings (Hazleton Pumps, Inc.)

pumps on simple floats can be protected from ice damage if the float is surrounded with a simple air bubble system.

As the pump flow rates increase, fixed installations become more manageable. Figure 9 shows four 24-in (610 mm) discharge, 1250 HP (750 kW) vertical pumps. The pumps are of stainless steel in a mining operation. This is an ideal installation as a structure is in place for pump maintenance.

The selection of the discharge line will have a significant effect on the total pumping output. If selected for normal pumping conditions, the pipeline may not add significant capacity to the system by the use of an additional pump. As an example, examine an installation utilizing a 14-in (355 mm) Schedule 40 steel pipe discharge line for a normal flow of about 5000 gpm (1135 m^3/h). Assume a friction head of 70 ft (21.3 m) and a static head of 180 ft (54.9 m), for a total pumping head of 250 ft (76.2 m). Figure 10 shows that adding a second pump to the system will not double the pumping capacity as the two pump characteristics are added to the system head curve. In this exaggerated example, adding a second pump will increase the total flow by less than 1000 gpm (227 m^3/h). Note that each pump now will discharge less than 3000 gpm (680 m^3/h). The obvious answer is to either use a larger discharge line and a different pump selection, or the use of a separate discharge line for each pump. In that case, the total capacity would be 10,000 gpm

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FIGURE 9 24 in (610 mm) discharge, 1250 HP (750 kW) vertical pumps (Hazleton Pumps, Inc.)

(2270 m^3/h). This becomes a study in economics for each installation. In the above example, this installation is obviously poorly designed. As pipeline velocities increase, discharge line transients may present problems on shutdown. The author is aware of one long pipeline that failed in 17 places on the first shutdown. Other factors must also be considered, such as the projected life of the project and the cost if the project is flooded.

Much has been said about pumping from mines. Obviously, the least costly method is to keep the water from entering the mining operation. It is not always applicable, but a ring of "deep-well" turbine pumps, on the periphery of the operation, can lower the water table sufficiently to keep the mining pit relatively dry. A typical large turbine pump with submersible motor is shown in Figure 11.

Pumps for operation in deep mines have different considerations. If a mine is not more than approximately 1200 ft (365 m) deep, Schedule 40 pipe is generally satisfactory for the pressure involved. For example, 12-in (305 mm) seamless Schedule 40 steel pipe is listed as 1200 lb/in^2 (82.75 bar) hydrotest pressure. However, if the water is corrosive, wall thickness allowance must be made, or stainless steel pipe used. 1200 lb/in^2 converts to approximately 2700 ft of water. Allowance must also be made for water hammer if the line velocity is high. Generally, average line velocities of 10 ft/s (3.1 m/s) will produce high

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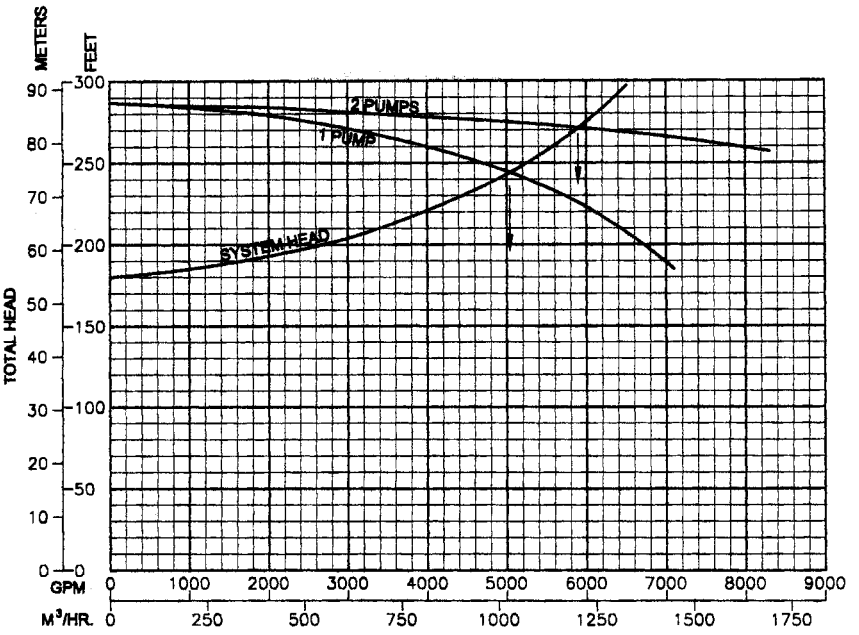


FIGURE 10 One and two pump operation on a system curve

transient pressures upon pump shutdown, so more conservative velocities should be selected. As these installations are generally long term, considerable savings can be obtained by reduced power requirements if the pipeline velocities are near 5 ft/s (1.5 m/s).

In designing a mine pumping system, an analysis must be made where the water is coming from. It is generally not practical to put the main pump station at the bottom of the mine if the bulk of the water is occurring at an upper level. If this is the case, a decision must be made for the pumping condition of the mine bottom pumps. Should they simply pump to the upper level or directly to the surface? Both financial and safety considerations are essential. Failure of upper level pumps may jeopardize the mine safety. These are decisions that involve much more than basic equipment selection. One solution to the problem of a possible power failure is the use of submersible pumps at the shaft bottom. Single-stage submersible pumps, as shown in Figure 12, are available to 600 HP (448 kW) and can readily pump to upper levels in case of flooding at the bottom level.

If the mine is deeper than 1200 ft (365 m), other approaches must be considered. Consideration of Class 250 or 300 valves and pipefittings may show that the cost warrants pumping only to an upper level to stay within the limits of the Class 150 fittings. This does not necessarily mean a duplicate pumping station as series pumps can safely be installed at an upper level. This is practical only if the bulk of the water occurs at or near the bottom level. Because there are mines deeper than 2000 ft (365 m), the problems become more acute. In any event, the main consideration is to conduct a study to determine at what level the greatest flow of water occurs.

A very old but interesting photo (see Figure 13) shows a sealed pump-room housing three 1000 HP (746 kW) bronze constructed pumps in a coal mine. This pump room had access only from an upper level so water could rise in the mine by approximately 200 ft (36.5 m) before flooding the pump room.

Not all mines suffer from ingress of a great deal of water, but many of those that do have closed because the cost of pumping was too great. An example is shown in Figure 14. There were three 2000 HP (1490 kW) pumps in one pump room in a zinc mine. Note in this

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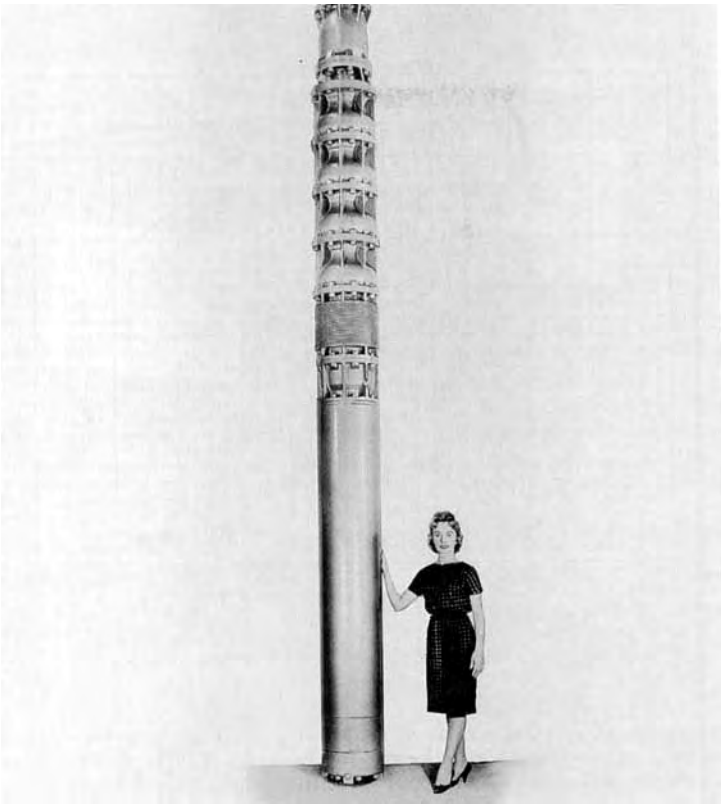


FIGURE 11 A typical large turbine pump with submersible motor (Hazleton Pumps, Inc.)

photograph that these large pumps are in segments so they can be lowered into the mine and reassembled.

Pump discharge line forces must be calculated for all installations. The reactive forces from the piping must be isolated from the pump. Underground pumps generally operate at high pressure and the forces generated are high. Failure to adequately support and restrain the piping can cause pipe failure and result in severe pump misalignment.

MATERIALS OF CONSTRUCTION

Most water pumps perform quite satisfactorily with bronze impellers, wearing rings, and shaft sleeves, but the dirty or corrosive waters in mine service require superior materials. Because mine waters range from neutral (pH of approximately 7) to severely acidic (as low as pH 1.5 in some coal mines) and to very basic (as occurs in limestone mines and so on), there is no universally best material. The choice obviously is the lowest-priced material that gives satisfactory service life.

Many times, the decision must be made not on the basis of the best available materials, but on the basis of which material will best withstand the abrasive conditions during the intended life of the project. If there is a five-year anticipated life of the mine, there is little advantage in selecting materials that will last twice as long. Conversely, a mine with

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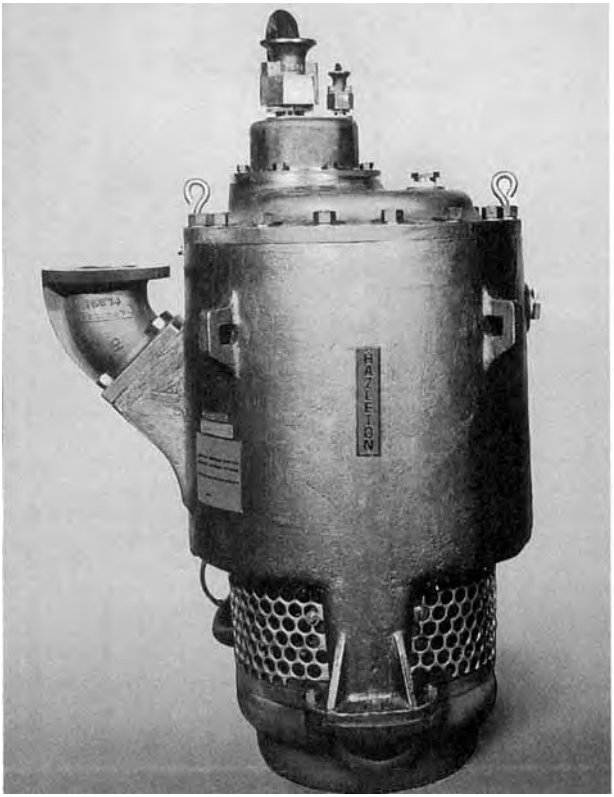


FIGURE 12 Single-stage submersible pump (Hazleton Pumps, Inc.)

a projected life of 25 years would require the best commercially available materials. There are also the exotic alloys, but their use is seldom justified.

Assuming that there will always be a slight amount of solids present, the metallurgy given in Table 2 should be considered.

Care should be exercised when selecting alloys because many stainless steels do not have as great a strength as carbon steel. The same applies to bolting. Thus, a pump rated for high-pressure service with a carbon steel or alloy steel casing and bolting may not be suitable when made of bronze or stainless steel.

Although ceramic-coated shaft sleeves are excellent in many applications, remember that ceramic coatings are porous and that the base metal must be able to withstand the environment. Also, some ceramics will not be suitable in strongly basic water. Others, however, are suitable, and so a general specification for ceramic coating should never be made. A plasma-applied ceramic is generally denser and more serviceable than one applied by a simple flame spray.

HYDRAULIC CONDITIONS

Low-Head Pumps For low-head pumps, up to approximately 150 ft (46 m), the installation must be carefully checked to prevent cavitation during periods of low-head operation. This is particularly important during one-pump operation on a parallel system. For

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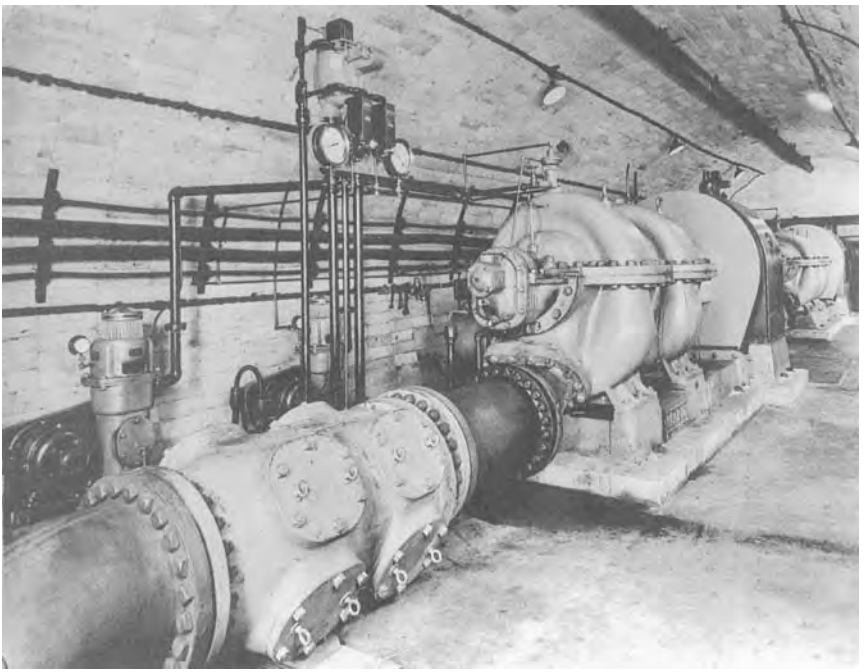


FIGURE 13 A sealed room in a coal mine housing three 1000 HP (746 kW) bronze constructed pumps (Hazleton Pumps, Inc.)

example, if two 5000-gpm (1136-m^3/h) pumps are discharging into a common discharge line against a static head of 80 ft (24 m) and a frictional head of 60 ft (18 m), the frictional head will be only 15 ft (4.6 m) when one pump operates alone at 5000 gpm (1136 m^3/h). The total head will now be only 95 ft (69 m), and the single pump will carry out to a much higher capacity. Unless sufficient NPSH is available for the single-pump runout point, the pump will cavitate. Although the two pumps should be selected for 5000 gpm (1136 m^3/h) at 140 ft (43 m) total head, the required NPSH should be determined not at 5000 gpm (1186 m^3/h) but at the capacity corresponding to the intersection of the pump and system curves.

High-Head Pumps For high-head pumps, 1000-ft (305 m) head or more, the risk of cavitation for single-pump operation on a parallel system is less than for low-head pumps. For example, if the static head is 1000 ft (305 m) and the frictional head is 60 ft (18 m) when two 5000-gpm (1136-m^3/h) pumps are operating, the total head will decrease from 1060 ft (323 m) to only 1015 ft (309 m) when one pump operates at 5000 gpm (1136 m^3/h). This means that, for either one- or two-pump operation, the capacity of each pump will be approximately the same and the risk of run out cavitation is minimal.

Waterhammer and Pressure Pulsations A waterhammer analysis should be made of both high- and low-pressure pumping systems. Although the transient pressure pulsations are related to the rate of change of velocity rather than the magnitude of the steady-state condition, mine experience indicates that waterhammer problems can be anticipated when pipe velocities exceed 10 ft/s (3 m/s). In high-pressure pumping systems, it is not unusual for transient pressure pulsations to be as high as 300 lb/in^2 (2068 kPa) above or below the steady-state pressure.

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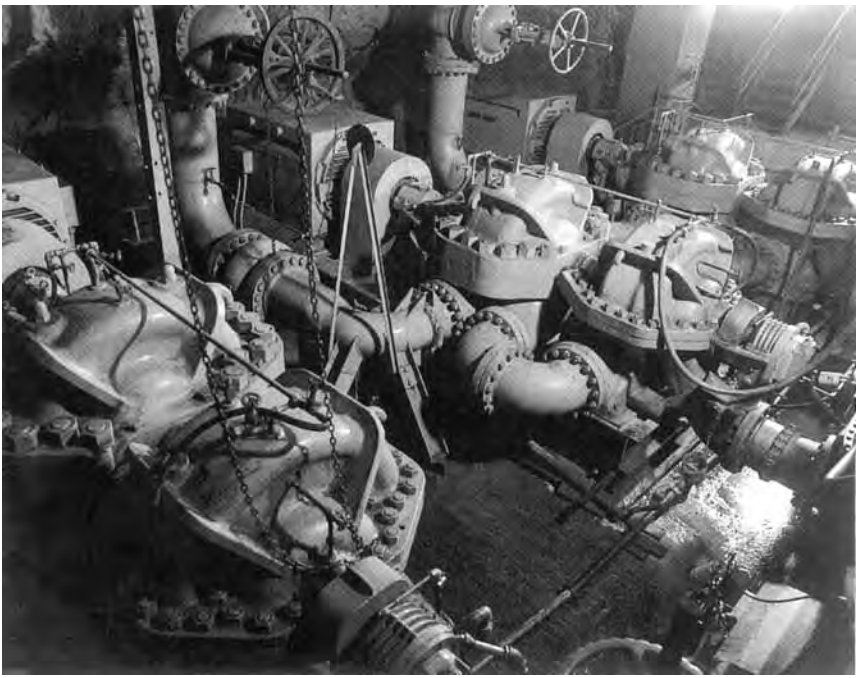


FIGURE 14 Three 2000 HP (1490 kW) pumps in one room in a zinc mine (Hazleton Pumps, Inc.)

Transient pressure pulsations have been experienced in low-pressure pumping systems. The danger here is that the low-pressure portion of the cycle will fall below atmospheric pressure and the pipe will collapse.

Although any pumping system for mine service should be analyzed in detail for transient pressure pulsations, experience has shown that adequate air bottles have proved to be one of the most effective and least expensive means of surge suppression. Slow-closing valves and flywheels have been used, but they must be sized correctly. This is especially true with high-speed pumps because these units possess little rotational inertia and will decelerate very rapidly on shutdown, with accompanying high-pressure surge.

SUMPS

Permanent sumps are seldom used in open-pit mines because the sump area generally moves as the mining operation progresses. This means that the pump station must be portable, and the installation of the pumps on a barge provides the most convenient arrangement. Either horizontal or vertical pumps may be used, but vertical pumps eliminate the need for priming equipment. If the vertical pumps are of the overhung-shaft design, the stuffing box may be eliminated. This is important if the water is dirty. Although small cyclones can be used to clarify the water for pumps that have stuffing boxes, care must be taken to prevent leaves and other trash from blocking the gland water line. In freezing climates, special provision must be made to prevent the suction line, pump, and even the barge itself from freezing in place.

Underground mine sumps present special problems because they function not only as sumps but also as clarifiers. A well-designed sump is a good clarifier, but all too frequently

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TABLE 2 Materials of construction for mine pumps

	Neutral waters		Acidic waters		Basic waters	
	Moderate heads	High heads	Moderate heads	High heads	Moderate heads	High heads
Casing	Cast iron	Ductile iron/ cast steel	316 S.S./ alloy 20	17-4 PH	Cast iron	Ductile iron/ cast steel
Impeller	28% Cr	28% Cr	PH55A/ 17-4PH CD-4MCu	PH55A/ 17-4PH CD4-MCu	28% Cr	28% Cr
Wear rings	28% Cr	28% Cr	Same as impeller	Same as impeller	28% Cr	28% Cr
Shaft sleeve	28% Cr or 303 S.S. ceramic-coated	28% Cr or 303 S.S. ceramic-coated	316 or alloy 20 ceramic-coated PH55A, etc.	316 or alloy 20 ceramic-coated PH55A, etc.	28% Cr or 303 S.S. ceramic-coated	28% Cr or 303 S.S. ceramic-coated
Shaft	Carbon steel	High-tensile alloy steel	316 S.S./ alloy 20	17-4 PH	Carbon steel	High-tensile alloy steel
	Alloy 20	21% Cr	29% Ni	2.5% Mo		
	CD4-MCu	26% Cr	5% Ni	2.0% Mo		
	28% Cr	28% Cr	—	—		
	304 S.S.	19% Cr	10% Ni	—		
	316 S.S.	19% Cr	10% Ni	2.5% Mo		
	17-4 PH	17% Cr	4% Ni	—		Hardenable
	PH55A	20% Cr	10% Ni	3.5% Mo		Hardenable

inadequate provisions are made for cleaning the sump. If it is not cleaned at regular intervals, the loss of storage capacity may be critical in the event of a power failure. Furthermore, a sump partially filled with solids does not give the proper retention time for clarification, and the solids are directed into the pump. Although few sumps can economically be made large enough for complete clarification, it is important that a large portion of the solids be removed. This is particularly true for 3600-rpm pumps because the high-speed generally produces a high head per stage and the high differential pressure between stages causes severe wear if abrasive solids are present. Some mines use conventional thickeners and flocculating agents in an attempt to keep a high concentration of solids from reaching the pump.

Some general rules should be considered in designing sumps for underground pump rooms:

1. Attempt to get a complete analysis of the water (from another portion of the mine or from an adjacent mine if necessary).
2. Analyze the sample for corrosive properties to determine the proper materials of construction for the pump.
3. Analyze the sample for possible scale buildup in the pipeline and pumps. Check the velocity effect, if any, on the buildup rate.
4. Determine the percentage of suspended solids in the sample, its screen analysis, and the settling rate for various fractions. Determine the sump dimensions necessary for removal of all solids and then for progressively larger solids in order to select the most economical size.

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5. Compare the sump size as determined in rule 4 with the size required for physical storage capacity for *(a)* continuous pumping, *(b)* off-peak power pumping, *(c)* programmed pumping, and *(d)* storage during estimated maximum length of power interruption.
6. Calculate practical sump dimensions, considering the geologic conditions.
7. Install grit traps ahead of the sump to remove large, heavy solids. Consider methods for cleaning the grit traps.
8. Install trash screens to prevent wooden wedges, and so on from entering the sump.
9. Review sump cleaning methods and program. The best-designed sump is of no value if it is not cleaned. Compare mechanical cleaning methods with cost of parallel sumps.
10. Review the suction requirements of the pumps to be used. Because of altitude, temperature, distance from low water level to pump centerline, and suction line loss, the available NPSH may be inadequate for even an 1800-rpm pump. If a decision as to pump size, type, and speed has been made and an NPSH problem does exist, a decision must be made either to use low-speed booster pumps or to lower the pump room level to below the sump level. From a safety standpoint, the use of a booster pump is preferable, although it does add another piece of equipment.
11. Where the storage capacity is inadequate to meet possible power failures, consider either vertical pumps—possibly up to 100 ft (30 m)—for the shaft bottom pumping up to the main pump station level, or sealed pump rooms that can operate over wide variations in the sump level from a 15-ft (4.6-m) suction lift to a positive head of several hundred feet.
12. Determine the final design based on a compromise between the mine engineer (who wants maximum output), the electrical engineer (who wants small starting load), the geologist (who wants small sump dimensions), and the mechanical engineer (who wants the most reliable and easily maintained equipment).

AUTOMATIC PUMP CONTROL

With proper instrumentation, almost all pump stations can be operated automatically. Remote monitoring is simple, relatively inexpensive, and can provide safe operation and signaling of nearly all operating conditions. Equipment is presently available to measure, record, and transmit the operating conditions to remote locations. The proper equipment can thus relieve worry about the operation of the facility even if it is many miles from the operation's headquarters. Automatic control can be a simple float switch or a pressure switch, or it can be sufficiently complex to provide reliable operation under the most critical or adverse conditions. Automatic control can provide greater reliability, and its cost can depreciate over only a few years. Furthermore, the automatic recording of flow rate, flow totalizing, and periods of operation provides valuable data for analyzing the performance of the pumping installation as well as the possible cost savings in pumping during off-peak power periods.

Where the safety of a mine is dependent on the reliable operation of the dewatering pumps and controls, the following minimum requirements should be considered:

1. There should be a sump level alarm for high water, both local and remote (at the surface).
2. Sump level control should be dependable. For example, electrodes are generally unreliable in waters that leave a conducting film.
3. The control should be programmed where more than one pump is installed. However, the use of an alternator is not always desirable because all pumps are exposed to the same degree of wear. It is preferable to have one standby pump programmed through a sequence selection switch to operate at least once per week.

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4. Pump priming should be positive. Hydraulic devices should be combined with electric controls so complete dependence is not on the electric control. The presence of water in the pump should be detected to prevent the starting of a dry pump.
5. A delay circuit should be provided to ensure complete priming.
6. The control should provide for at least three starting attempts (unless an overload has occurred).
7. The priming time should be limited (if under a suction lift system).
8. The control should provide for a restart in the event of a false loss of prime on start-up (suction lift system).
9. Pump and motor bearings should have thermostats to stop pumps in the event of bearing failure.
10. Vibration monitoring may be important. This is particularly true for vertical pumps.
11. Pressure controls should indicate normal pressure and fail-safe in the event of a loss of pressure (broken column line, and so on).
12. Flow indication (check valve flow switch) is needed to signal a shaft failure.
13. Remote indication (generally at the mine office) should provide at least an indication of operation and signal pump failure or high water. More detailed information may be transmitted.
14. For long distances, investigate the use of carrier-current indication schemes, together with signal multiplexing, and so on.
15. Provide a method to test the control and alarm system.

DRIVERS

Open-pit mines use electrically driven mining equipment, such as drag lines and shovels, and the availability of power has permitted the use of electrically driven pumps. There are still many gasoline-driven or diesel-driven units, but the convenience of electric power, particularly for automatically controlled units, has increased the trend to electric drive. The availability of reliable high-voltage cable has made portable high-voltage equipment safe and economical. Pumps in open-pit service are seldom provided with sophisticated control or drive mechanisms. The primary requirements are reliability, portability, and wear resistance.

In locations where rainfall may be heavy and there is danger of power failure, a combination of electric drive and engine drive is used. The engine can be direct-coupled to the pump through a motor with a double-extended shaft or with a clutch between the engine and the motor. Automatic control is simple and reliable.

Although some steam-driven pumps still exist in underground service, their number is rapidly decreasing. Electric motor drive is the simplest for automatic control. Variable-speed units, however, are seldom used in underground service because the ratio of static head to total dynamic head is quite high. Thus the frictional loss is not a large percentage of the total head loss and not much advantage is gained by variable speed. The solution is usually a multiple-pump installation. This must be designed with care because it is possible to raise the frictional head to a point where an additional pump produces little additional capacity. Multiple discharge lines are the answer and are frequently used for safety reasons. In normal service, all discharge lines are used in parallel, although conservative design allows each line to handle the required capacity.

As with all pumping installations, a complete set of system-head curves must be prepared to analyze the power requirements under all conditions.

Motor enclosures are important in underground service. Because of the high humidity, special insulation (epoxy encapsulated, and so on) should be specified. Dripproof enclosures are the minimum requirement, with weather-protected Type I the preferred construction. Heaters should also be provided. Screens should be installed to prevent the

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entrance of rats. Winding temperature detectors, bearing thermostats, and ground-fault detectors are recommended in mine service and should be incorporated in the pump-control and alarm circuits.

Although the starting torque of a centrifugal pump is low, the available torque may have to be checked in some cases. A normal-torque motor should be suitable for pumps in the range of 500 to 3000 (10 to 60) specific speed. High-specific-speed pumps, however, have the highest power at shutoff, and it will be necessary to examine the starting arrangements for such pumps.

Starting a pump against a long empty pipeline may present overload problems, and repeated starts may be necessary. In such cases, the number of permissible starts per hour should be checked. Winding temperature detectors are important in such applications.

Although reduced-voltage starters may be required in some instances, most modern mines have electrical facilities designed for across-the-line starting. This is preferable because the starting equipment is cheaper. Before deciding on a reduced-voltage starter, the effect of the starting load on the transformer and line impedance should be checked. It may be that the voltage drop will eliminate the requirement for reduced-voltage starters. On the other hand, the effect on the primary side should be checked so the voltage drop is not so large as to drop out other equipment.

Synchronous-motor drives are seldom used unless they are large (generally at least 1000 HP) (750 kW) and then only if they are in relatively continuous service. Under such conditions, they can be operated under "leading current" conditions for power factor correction. Smaller installations frequently provide capacitors at the pump installation to provide the necessary correction for a particular installation.

Surge protection from lightning should not be overlooked. Some locations are particularly susceptible to lightning damage, especially to long surface lines. Lightning arresters should be provided at the surface, and surge arresters should be mounted at the motor location.

REFERENCES

1. American National Standard for Centrifugal Pumps for Design and Application, ANSI/HI 1.3-2000, Hydraulic Institute, Parsippany, NJ, www.pumps.org.

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SECTION 12.12
MARINE PUMPS

WILLIAM J. SEMBLER

Pumps used in marine shipboard applications, both commercial marine and Navy, can typically be divided into several groups. These groups include pumps associated with a vessel's propulsion, pumps used with the generators that produce electricity, pumps used in ship's service systems, pumps used to provide hotel services for crew and passengers, and pumps that are used in cargo or other specialized systems.

Because marine pumps must operate on a moving platform, they should be designed to withstand dynamic loads resulting from vessel motion (for example, pitch, roll, and so on). In addition, they must often operate in a hot, humid, and potentially corrosive environment. In addition, marine pumps must frequently be suitable to operate with a range of flow rates to accommodate anything from operation of the vessel at full speed to operation in port with the propulsion equipment secured. Furthermore, the minimization of size (especially the required deck space) and weight is always important when designing marine equipment. For this reason, many shipboard pumps are mounted vertically (Figures 1 and 2), and smaller units are frequently furnished in a close-coupled configuration (Figure 3) with the pump's rotating parts mounted directly on the driver's shaft. To enable them to stand freely under pitch and roll conditions, vertically mounted shipboard pumps often have larger bases than comparable shore-side units. Typical materials used in the construction of marine centrifugal pumps are listed in Table 1.

A description of the features typically incorporated into the designs of pumps used in selected shipboard applications follows. This information is general in nature, however, and may not apply in all cases based on the requirements for specific installations or the preferences of vessel owners and designers.

LIBERTY EXHIBIT 1013, Page 1424

1 Casing
2 Impeller
3 Casing wearing ring
4 Impeller wearing ring
5 Shaft
6 Shaft sleeve
7 Packing
8 Gland
9 Thrust bearing
10 Line bearing
11 Bearing housing
12 Pump base
13 Motor bracket
14 Coupling

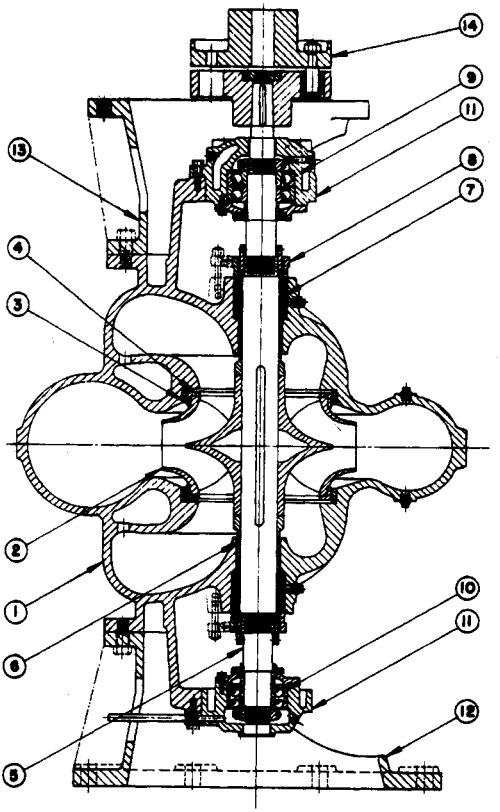


FIGURE 1 Vertical between-bearings centrifugal pump (Flowserve Corporation)

PROPULSION APPLICATIONS

Steam-Turbine-Propelled Vessels

FEED PUMPS A main feed pump is used to return water to a steam-powered vessel's boilers or steam generators. On a typical steam ship with fossil-fueled boilers, a main feed pump takes suction from a deaerating feed tank (DFT) and discharges feedwater to the steam drum in each of the vessel's boilers. In many cases, a main feed pump's discharge is connected to two separate lines that both lead to the boilers: a main feed line and an auxiliary feed line. In addition, the feedwater usually passes through one or more heaters before entering the steam drums. Although a single feed pump is frequently sized to handle a vessel's full-load requirements, some ships have multiple partial-capacity feed pumps that operate in parallel. Additional pumps are ordinarily provided for standby duty.

Typical feed pump configurations include single- and two-stage centrifugal pumps that are close-coupled to steam turbines (Figure 4) and multistage flexibly coupled pumps that are driven by steam turbines or electric motors. Although flexibly coupled feed pumps often have cast axially split volute-type casings (Figure 5), barrel pumps with diffusers and forged cylindrical casings are sometimes used. Turbine-driven feed pumps are usually mounted horizontally. Motor-driven feed pumps, however, have been used in both horizontal and vertical configurations.

LIBERTY EXHIBIT 1013, Page 1425

7450	Coupling guard
3281	Bearing cover, driver side
3712	Bearing lock nut
4595	Lock washer
3132	Adapter
4542	Bearing-cover gasket
3282	Bearing cover, pump side
4300/2	Lip seal
6573	Gland stud
4120	Gland
4610/2	Inner O-ring
4113	Cooling cap
1222	Stuffing-box cover
4521	Casing gasket
6710	Impeller key
1111	Casing
4510	Impeller-lock-washer gasket
4552	Impeller-nut gasket
3170	Pump base
2912	Impeller nut
6543	Impeller lock washer
1500	Casing wearing ring
2200	Impeller
1521	Stuffing-box-cover wearing ring
2450	Shaft sleeve
4133	Throat bushing
4134	Lantern ring
4610/1	Outer O-ring
4130	Packing
6731	Shaft-sleeve key
2540	Slinger
3042	Line bearing
3853	Grease nipple
2100	Pump shaft
3133	Bearing housing
3041	Thrust bearing
3853	Grease nipple
6742/2	Pump coupling key
7112	Coupling
3160	Mounting frame
6742/1	Motor coupling key
8010	Motor

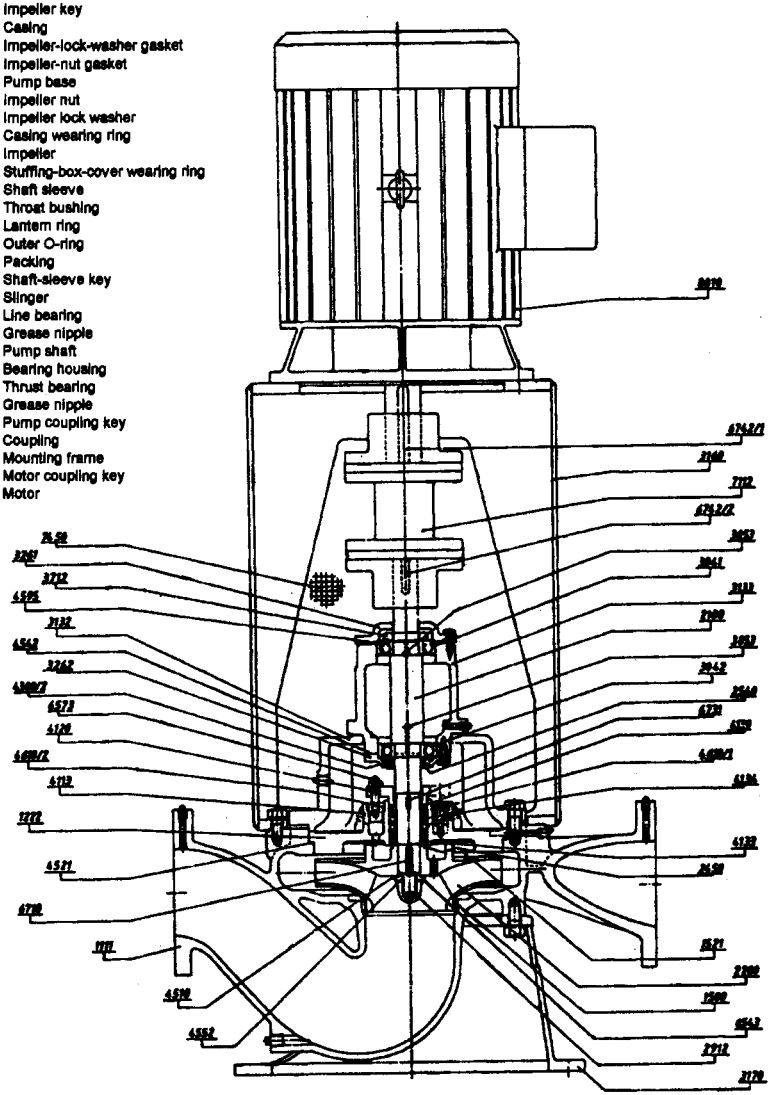


FIGURE 2 Vertical back-pull-out centrifugal pump (Flowserve Corporation)

LIBERTY EXHIBIT 1013, Page 1426

1 Casing and suction head
2 Impeller
3 Casing wearing ring
4 Impeller nut
5 Shaft sleeve
6 Gland
7 Seal cage
8 Shaft
9 Slinger
10 Shaft-sleeve seal ring
11 Packing

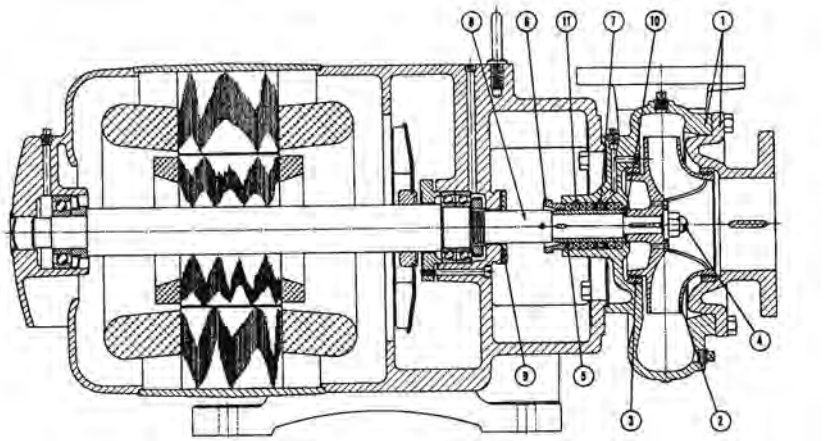


FIGURE 3 Horizontal close-coupled centrifugal pump (Flowserve Corporation)

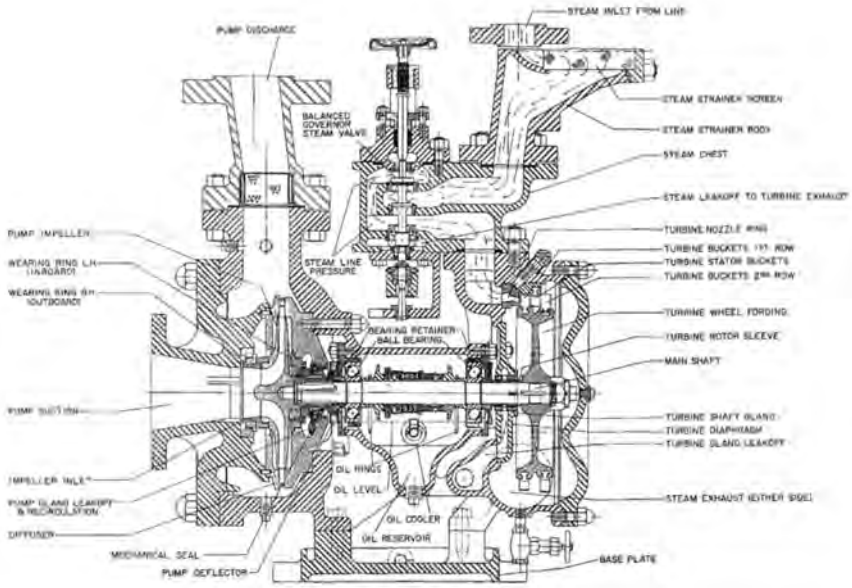


FIGURE 4 Close-coupled single-stage centrifugal main feed pump (Coffin Turbopump)

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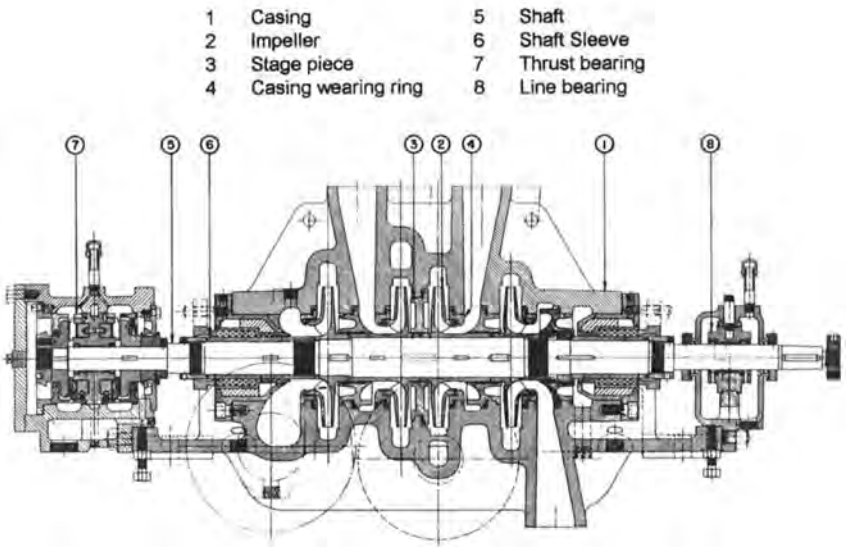


FIGURE 5 Flex-coupled, multistage centrifugal main feed pump (Flowserve Corporation)

Motor-driven feed pumps may have grease-lubricated external bearings. Bearings in a turbine-driven feed pump, however, are usually lubricated with oil that is circulated by a shaft-driven rotary pump. Oil discharged by this pump also ordinarily lubricates the turbine's bearings and may be used in the pump's governor. A strainer, filter, and seawater-cooled heat exchanger are often installed in the lubricating-oil system. The lubricating oil used with the turbine-driven feed pumps on many steam-turbine-propelled vessels is the same grade of oil used in the main propulsion lubricating-oil system.

Although packed stuffing boxes are used to seal shaft penetrations in many marine feed pump casings, to reduce stuffing-box leakage, some feed pumps are fitted with mechanical seals. In addition, a condensate-injection shaft seal consisting either of a nonrotating labyrinth-type fixed breakdown bushing or a series of spring-loaded floating rings that are stacked axially is sometimes used. When a feed pump that has a condensate-injection shaft seal is operating, cool condensate is injected into the seal and fills the close radial clearance between the nonrotating seal parts and a rotating shaft sleeve. A small portion of this condensate may flow into the pump. The remainder, however, flows outward and enters a collection chamber that is usually piped back to the vessel's gland-exhaust condenser.

During constant-speed operation, the capacity of water delivered to a boiler by a main feed pump is typically controlled by the throttling action of an automatic feedwater-regulating valve. However, when steam is used to drive a feed pump, the amount that the regulating valve must be throttled is often reduced by controlling the amount of steam supplied to the pump's driver and, therefore, the pump's operating speed with either a constant-pressure or a constant-differential-pressure governor.

A constant-pressure governor automatically regulates a feed pump's operating speed to maintain a constant pressure at the pump discharge. If there is a reduction in the boiler load and the feedwater-regulating valve begins to close, the feed pump's discharge pressure will rise and the capacity of feedwater that the pump delivers to the boiler will be reduced. The constant-pressure governor, however, will sense the rise in discharge pressure and will reduce the pump's speed until this pressure returns to the desired value. As a result of the speed reduction, the amount that the regulating valve must close to limit the feedwater flow rate is reduced. An increase in boiler load has the opposite effect.

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A constant-differential-pressure governor (sometimes referred to as an excess-pressure governor) regulates a steam-driven feed pump's operating speed to maintain a set difference between the pump discharge pressure and the pressure on the boiler-side of the feedwater-regulating valve. Because changes in the feed flow entering the boiler result, primarily, from variations in pump speed, the throttling action of the feedwater regulating valve is greatly reduced.

A relief valve is often installed on the discharge side of a main feed pump to protect the feed system from overpressurization. In addition, to prevent a feed pump from operating with too low a capacity, which could occur when the boiler load is low, a recirculation line is typically connected from the pump discharge to the DFT. An orifice is typically installed in the recirculation line to limit flow and to reduce the pressure of the water being recirculated to match the pressure in the DFT. A valve that can be closed during high-load operation is also frequently mounted in a feed-pump recirculation line.

Steam turbines that are used to drive feed pumps are generally protected with low-lubricating-oil-pressure, overspeed, and high-turbine-exhaust-pressure trips. In addition, a low-suction-pressure trip is sometimes provided to prevent a feed pump from operating with too low a suction pressure, which can result in cavitation, overheating, and a loss of load on the pump's driver.

In addition to the main feed pumps, a smaller capacity feed pump is also installed on some steam-powered vessels for use in port or during emergencies. Steam-driven, direct-acting piston-type (Figure 6) and motor-driven plunger-type reciprocating pumps, either vertically or horizontally mounted, are often used for in-port feed service.

Because the feedwater removed from a DFT is normally at its vapor pressure, the net positive suction head (NPSH) available to a main feed pump is essentially equal to the elevation of the water level within the DFT above the feed pump, less losses in the feed-pump suction line. On a vessel where the elevation of the DFT is not sufficient to provide an adequate NPSH to the main feed pumps, separate electric-motor-driven centrifugal-type booster pumps are typically installed between the DFT and the main feed pumps. The booster pumps, which operate in series with but at a much lower speed than the main feed pumps, raise the pressure of the feed water entering the main pumps and, therefore, reduce the potential for cavitation.

MAIN CONDENSATE PUMPS A typical main condensate pump takes suction from the hotwell in a main condenser and discharges condensate, through various heat exchangers, to a deaerating feed tank (DFT). Vertically mounted centrifugal pumps with two or more stages are frequently used in this application. Although many of these pumps are driven by electric motors, some main condensate pumps are driven through reduction gears by steam turbines. Two condensate pumps are generally provided for each main condenser, with each pump sized to handle full-load requirements.

A typical two-stage condensate pump (Figure 7) is fitted with grease-lubricated ball bearings at the upper end of its shaft to absorb both axial and radial loads. In addition, an internal water-lubricated radial sleeve bearing is often installed between the two impellers. The first-stage impeller is usually mounted near the lower end of the shaft, which increases its submergence. In addition, its suction eye is directed upward, which enables the impeller to be self-venting. To help facilitate the removal of any air that may enter the pump, a vent line is ordinarily connected from the suction side of a condensate pump's casing to the upper portion of the condenser. The second-stage impeller is ordinarily mounted near the upper end of the condensate-pump shaft with its suction eye directed downward. With this orientation, the hydraulic axial thrust applied the second stage impeller opposes the axial thrust acting on the first-stage impeller, and the net axial load that must be absorbed by the pump's thrust bearing is reduced. In addition, condensate at the base of the shaft seal has already passed through both impellers and is, therefore, at an elevated pressure. This helps to prevent air from being drawn into the pump through the shaft seal. The effectiveness of the shaft seal, which can consist of a packed stuffing box or a mechanical seal, is also frequently increased by injecting pressurized condensate recirculated from the pump's discharge into the seal area.

The condensate removed from a condenser's hotwell is normally at or close to its vapor pressure. Consequently, the NPSH available to a condensate pump is essentially equal to

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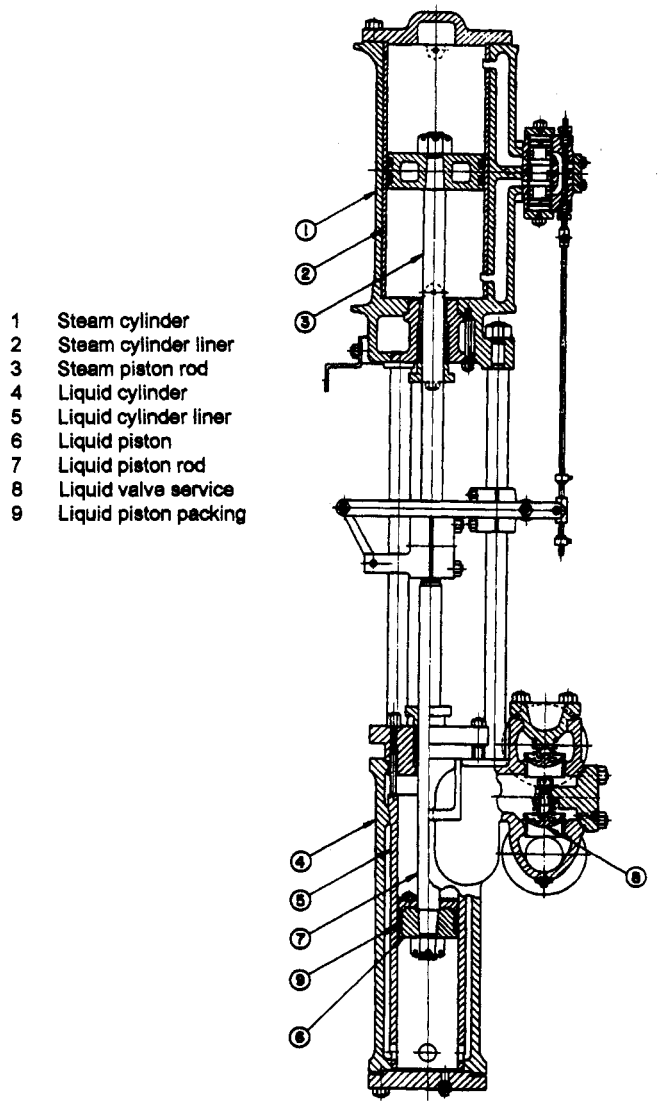


FIGURE 6 Simplex direct-acting reciprocating in-port feed pump (Flowserve Corporation)

the height of the water level in the hotwell above the pump's first-stage impeller, which is seldom more than a few feet, less frictional losses in the suction piping. To help increase the NPSH available to main condensate pumps, they are ordinarily installed as low in a vessel as is practicable. They also frequently have special low-NPSH first-stage impellers.

If a condensate pump is driven by a steam turbine or by a variable-speed electric motor, its operating speed can be adjusted with plant load. This allows the capacity at which condensate is removed from the hotwell to match the rate at which steam is exhausted into

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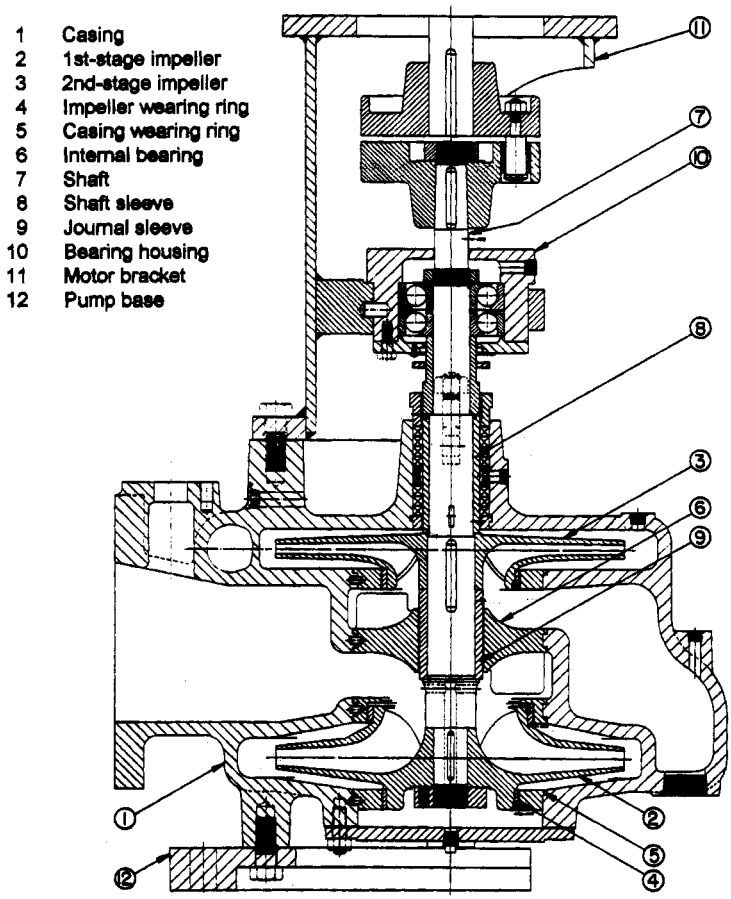


FIGURE 7 Two-stage centrifugal main condensate pump (Flowserve Corporation)

the condenser. This is usually accomplished by means of water level control. Alternatively, when a constant-speed driver is used, the capacity delivered by a main condensate pump is sometimes regulated by cavitation, which is referred to as submergence control. When submergence control is utilized, a reduction in plant load will typically result in a reduction in the main-condenser hotwell level and, therefore, in the NPSH available to the condensate pump. After the NPSH available drops below the condensate pump's NPSH requirement, the capacity being removed from the hotwell will be reduced by cavitation within the pump. As the hotwell level continues to drop, the amount of cavitation occurring will increase, and the pumped capacity will continue to be reduced until it eventually matches the rate at which steam is entering the condenser. When this occurs, the water level in the hotwell will stabilize at an elevation where the NPSH available to the condensate pump is equal to the pump's NPSH requirement at the new reduced capacity being pumped. Pumps designed to operate with submergence control need to be of robust construction and low-energy level (per stage) to prevent damage from cavitation and cavitation-induced vibration.

To avoid operation with excessive cavitation, the capacity delivered by a constant-speed condensate pump can be regulated by throttling the pump's discharge valve as

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needed to maintain a constant condenser-hotwell level. However, when steam-jet air ejectors are used to deaerate a condenser, condensate-pump operation at too low a capacity can result in insufficient cooling in the ejector inter- and after-condensers. In addition, low-capacity operation can lead to surging caused by suction and discharge recirculation within a condensate pump. To prevent these problems from occurring and to reduce the need to throttle a condensate pump's discharge valve, a recirculation line back to the condenser is frequently used. This line is connected to a tee in the condensate-pump discharge piping, downstream from any air-ejector and gland-exhaust condensers. With this arrangement, if the hotwell level drops, a valve in the recirculation line can be opened, either manually or automatically, to permit a portion of the water discharged by the condensate pump to be returned to the condenser. The hotwell level can be maintained, therefore, at a value that is sufficient to suppress cavitation. In addition, because the condensate pump can always operate at or near its rated capacity, low-flow operation is avoided. Furthermore, an adequate flow of condensate through air-ejector condensers, when used, and a vessel's gland-exhaust condenser can be maintained during plant start-up and low-load operation. In some cases, the condensate-recirculation line may also be fitted with a thermostatically-controlled valve that opens and permits the condensate flow rate through the air-ejector condensers to increase when the temperature of the condensate leaving these heat exchangers exceeds a pre-set value.

CONDENSER-EXHAUSTING PUMPS Electric-motor-driven liquid-ring vacuum pumps are sometimes used instead of steam-jet air ejectors to deaerate a main condenser. The vacuum created by a liquid-ring vacuum pump is limited by the vapor pressure of the liquid compressant (water) within the pump's casing, which increases with temperature. Therefore, water separated from the gases discharged by a liquid-ring condenser-exhausting pump is generally cooled in a heat exchanger before being returned to the pump. Two pumps that are each capable of maintaining the required vacuum during full-load plant operation are frequently provided for each main condenser.

FRESHWATER-DRAIN-COLLECTING-TANK PUMPS A freshwater-drain-collecting-tank (FWDCT) pump (also sometimes referred to as an atmospheric-drain-tank pump) typically transfers condensate (collected from uncontaminated drains that are above atmospheric pressure) from a freshwater-drain-collecting tank to a deaerating feed tank (DFT). Electric-motor-driven single-stage centrifugal pumps are often used in this application. Two full capacity pumps are usually provided. With this arrangement, the on-line pump is often started and stopped automatically by a float control in the drain tank. The condensate removed from a freshwater-drain-collecting tank, which is ordinarily at a temperature of approximately 200 to 210°F (93 to 99°C), is close to its boiling point. To increase the NPSH available to a vessel's FWDCT pumps, they are frequently installed as far below the drain tank as practicable.

MAIN CIRCULATING PUMPS On a steam-powered vessel, a main circulating pump delivers seawater to a main condenser that receives steam exhausted from a propulsion turbine. In addition, a portion of the seawater discharged by a main circulating pump may be diverted to other seawater-cooled heat exchangers, such as the main lubricating-oil coolers. After passing through the main condenser or another cooler, the seawater is directed overboard. In addition to the main suction flange, which is connected to a sea chest, some main circulating pumps also have an auxiliary side-suction connection that can be used for emergency dewatering of the machinery space bilge.

Typically, one or two main circulating pumps are provided for each of a vessel's main condensers. Single-stage axial-flow propeller pumps (Figure 8) that deliver high flow rates at relatively low discharge pressures are often used in this application. In many cases, the pump is not furnished with a thrust bearing. Instead, the pump shaft is rigidly coupled to the driver's shaft, and axial loads are absorbed by the driver's thrust bearing. Radial loads applied to the pump shaft, however, are generally absorbed by a journal bearing that is located above the overhung propeller. During operation in harbors and inland waterways, the water delivered by a main circulating pump will often contain silt, sand, and other abrasives. Consequently, internal journal bearings that are lubricated by water discharged

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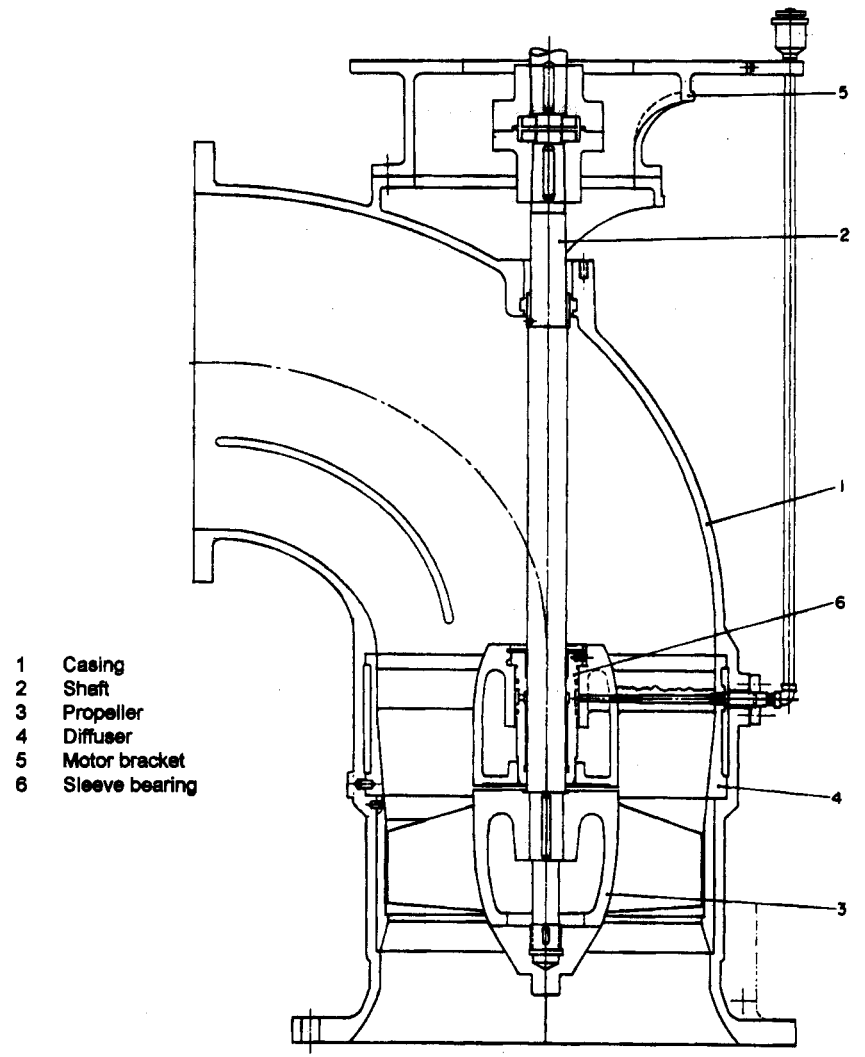


FIGURE 8 Axial-flow main circulating pump (Flowserve Corporation)

from the propeller are sometimes furnished in abrasion-resistant grades of rubber or composite materials. Alternatively, some main-circulating pump internal bearings are lubricated by grease or clean water supplied through an external connection in the pump's casing.

When higher discharge pressures are required, single-stage mixed-flow pumps with overhung end-suction impellers or single-stage radial-flow pumps with double-suction impellers mounted between bearings are sometimes used as main circulating pumps.

Main circulating pumps are generally driven either by steam turbines with reduction gears or by two-speed electric motors. This permits the capacity of seawater discharged through the condenser to be reduced during system startup and shutdown and other

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periods of low-plant-load operation by reducing the speed of the circulating pump's driver. Many steam ships are fitted with a scoop-injection system that permits seawater to be forced through the main condensers by the forward momentum of the vessel. With this arrangement, the main circulating pumps are usually stopped whenever the vessel is operating above a certain speed.

FUEL-OIL SERVICE PUMPS A fuel-oil (FO) service pump typically takes suction through either high- or low-suction ports in the FO service or settling tanks and supplies fuel oil to the burners in a steam ship's oil-fired boilers. At least two service pumps each capable of delivering the vessel's full-power fuel-oil requirement are generally provided. FO service pump suction and discharge lines are generally fitted with duplex strainers. In addition, one or more flow meters and heaters are frequently installed in a FO service system.

Horizontally and vertically mounted multiple-screw rotary pumps that are driven by either steam turbines or two-speed electric motors are often used in this application. In addition, some older ships have steam-driven reciprocating FO service pumps. When a vessel normally burns heavy (residual) fuel oil, which must be heated before being delivered to a boiler's burners, a separate electric-motor-driven rotary pump is also usually provided to supply unheated distillate fuel oil to the boilers during plant start-up.

The steam flow to a turbine-driven rotary or a direct-acting reciprocating FO service pump is often regulated with a constant-pressure governor that acts to maintain a constant FO-service-pump discharge pressure. With this arrangement, changes in the flow rate passing through the FO service pump result from variations in the pump's operating speed. When a two-speed electric motor is used to drive a FO service pump, the motor is ordinarily operated at low speed during low-plant-load periods (for example, in port) and at high speed during high-plant-load periods. However, during operation at either speed, a motor-driven FO service pump typically discharges more oil than the amount required by the burners. The excess oil is usually returned to the suction side of the FO service system through a recirculation line. The flow rate through this recirculation line is often regulated by an automatic valve that maintains a constant FO-service-pump discharge pressure. An additional recirculation line with a hand-operated valve is frequently provided to permit fuel oil to be circulated through a FO-service-system heater and a boiler's burner manifold prior to lighting-off the boiler. To prevent FO-service-system overpressurization, a relief valve should always be installed on the discharge side of each FO service pump.

Remote controls are typically provided that enable a vessel's FO service pumps to be stopped in an emergency from outside of the machinery space. In addition, flanged FO-service-pump discharge connections are ordinarily fitted with wrap-around shields to deflect spray in case of a leak. In addition, a drip pan or similar device is usually installed under a FO service pump to prevent oil that may leak out of the pump from draining into a vessel's bilge.

LUBRICATING-OIL SERVICE PUMPS A lubricating-oil service (LOS) pump removes lubricating oil from the propulsion-reduction-gear sump and discharges it to the propulsion-turbine bearings, the reduction gears and their bearings, and the main thrust bearing for each propulsion shaft. This oil is also usually directed to the inlet side of a speed-limiting governor pump mounted on the forward end of each propulsion turbine's shaft and to the propulsion-turbine throttle- or nozzle-valve controls. In addition, a portion of the pumped oil may be sent to an overhead gravity tank that holds enough oil to lubricate the propulsion machinery for several minutes following a loss of LOS-pump discharge pressure.

Most steam-powered vessels have two or three LOS pumps. Horizontally and vertically mounted multiple-screw pumps are frequently used. When mounted vertically, the pumps may be submerged directly within the lubricating-oil (LO) sump. In most cases, at least one rotary LOS pump is electric-motor-driven. Remaining units may also be motor driven, or they may be driven by steam turbines or off the propulsion shafting. In addition, steam-driven reciprocating-piston pumps are sometimes used as standby LOS pumps. In addition, some vessels have a battery-operated emergency LOS pump.

LOS pumps are typically installed low in a vessel and close to the LO sump. Suction and discharge lines are often fitted with duplex strainers. In addition, a relief valve and

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coolers are generally installed in pump discharge lines. Also, one or more orifices are typically installed in a LOS system to reduce the pump's full discharge pressure, which is usually based on the requirements of the propulsion-turbine governors and controls, down to the pressure required by the bearings and gears. A pressure-controlled switch or valve that automatically starts a standby pump if the oil pressure at the discharge of the operating LOS pump drops below a preset value is often provided. Most steam-turbine-propelled vessels also have a device that stops the supply of steam to the ahead propulsion turbines in case of a LOS-system failure.

Diesel-Propelled Vessels

A propulsion diesel engine that operates below approximately 300 rpm and is directly connected to a propulsion shaft is usually classified as a slow-speed engine. Diesel engines that are connected to propulsion shafts through reduction gears and have maximum operating speeds below 900 to 1200 rpm are typically classified as medium-speed engines, and engines that operate above 900 to 1200 rpm and are used with reduction gears are generally classified as high-speed engines.

ENGINE FRESHWATER COOLING PUMPS A pump is ordinarily used to circulate freshwater through jackets in a propulsion diesel engine's cylinders and cylinder heads, the engine's turbocharger, and, on some vessels, an evaporator where the jacket water provides heat to produce fresh water. The jacket water may also pass through the engine's lubricating-oil and charge-air coolers. In addition, a heat exchanger in which the jacket water is cooled by either freshwater or seawater is frequently included in the system. Alternatively, this heat exchanger is sometimes eliminated when a vessel has a central freshwater cooling system. Instead, some freshwater from a separate low-temperature loop is admitted into the high-temperature jacket-water loop where it mixes with and cools the hot jacket water. An additional heat exchanger that can be used to preheat the jacket water prior to engine start-up is often provided as part of a main engine's jacket-water system.

The pumps that circulate cooling water through a propulsion diesel engine are frequently called jacket-water circulating pumps. Alternatively, however, they may be referred to as high-temperature freshwater cooling pumps on a vessel that has a central freshwater cooling system. Two vertically or horizontally mounted single-stage centrifugal pumps that are each sized to meet normal full-load requirements are often provided for each propulsion engine. Although each pump may be driven by an electric motor, when used with a high- or medium-speed propulsion engine, one of the pumps is often mounted on and driven off the engine. An elevated tank is ordinarily included in the system to allow for any thermal expansion of the water and to maintain a positive pressure at the pump suction.

Separate electric-motor-driven pumps may be used to circulate fresh water through a propulsion engine's pistons when they are water-cooled. Although single-stage centrifugal pumps are often used in this application, vertical turbine pumps that are submerged in the piston-cooling-water drain tank are used on some vessels. When required by the engine design, an additional pair of electric-motor-driven centrifugal pumps is provided to circulate fresh water through an independent loop that cools the main-engine fuel valves or injectors.

ENGINE SEAWATER COOLING PUMPS On some vessels, single-stage centrifugal pumps are used to supply seawater to diesel-engine charge-air, lubricating-oil, and jacket-water coolers. Two pumps that are each sized to meet normal full-load requirements are often provided for each propulsion engine. Although both seawater-cooling pumps may be driven by electric motors, when a medium- and high-speed diesel engine is used, one pump is sometimes attached to and driven by the engine.

FUEL-OIL SUPPLY AND BOOSTER PUMPS Fuel-oil (FO) pumps used for diesel engines should be suitable to handle any of the various grades of fuel that may be burned in a vessel's

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engines, which can often include both light distillates that are used during maneuvering and heavy residual oils that are used while underway at sea. When heavy fuel oil is supplied to an engine, it must generally be heated to reduce its viscosity.

A FO booster pump typically takes suction either directly from a diesel-propelled vessel's daily service tanks or from a separate mixing tank and supplies fuel oil to the main-engine injector pumps. Two multiple-screw- or gear-type fuel-oil booster pumps that are each capable of meeting full-power requirements are often provided for each propulsion engine. Both FO booster pumps may be electric motor driven. Alternatively, however, when a medium- and high-speed diesel engine is used, one FO booster pump is sometimes attached to and driven by the engine. Also, in some high-temperature heavy-fuel-oil systems, a pair of electric-motor-driven, screw- or gear-type FO supply pumps that operate upstream of and in series with the FO booster pumps are located between the service tanks and the mixing tank. More oil is typically delivered to an engine than the amount required for combustion, and the excess oil is ordinarily returned to the service or mixing tank through a recirculation line. Strainers, filters, flow meters, and heaters are also frequently installed in the FO service system.

LUBRICATING-OIL PUMPS The main-engine lubricating-oil (LO) pump typically removes lubricating oil from a sump located below a propulsion engine and discharges the oil to the engine's bearings, governor controls, turbochargers, and, when they are cooled by oil, the engine's pistons. Strainers, filters, and a cooler are also usually included in a LO system. Two main LO pumps that are each capable of delivering the oil required during normal full-load engine operation are usually provided for a propulsion engine. Vertically or horizontally mounted multiple-screw and gear pumps are frequently used in this application. Although both pumps may be driven by electric motors, one pump is sometimes attached to and driven off a medium- or high-speed engine. Alternatively, some vessels have electric-motor-driven, vertical turbine pumps that are submerged within the LO sump installed. A device that sounds an alarm following a failure of the LO system is normally provided.

When a vessel is propelled by a crosshead-type diesel engine, some of the oil discharged by the main LO pump is often directed to the suction-side of a lower-capacity rotary-type booster pump that supplies oil to lubricate the engine's crosshead bearings. In addition, a separate low-capacity rotary pump is frequently required to fill a head tank that supplies a different grade of oil to lubricators that inject the oil into each of a crosshead engine's cylinders. Also, with some designs, a separate pair of rotary pumps is required to supply lubricating oil to the engine's camshaft bearings.

In addition to the pumps that supply lubricating oil to propulsion engines, separate rotary pumps are generally used to supply lubricating oil to reduction gears and their bearings on vessels propelled by medium- or high-speed engines.

Gas-Turbine-Propelled Vessels

FUEL-OIL SERVICE PUMPS Typically, fuel oil is delivered to a propulsion gas turbine's combustion-chamber nozzles by a gear pump that is mounted on and driven by the gas turbine. In addition, a pair of two-speed electric-motor-driven rotary pumps that operate upstream of and in series with the attached pump are usually provided. Duplex strainers, filters, and heaters are also ordinarily included in a gas turbine's fuel-oil service system.

LUBRICATING-OIL PUMPS A typical gas turbine is furnished with an attached gear- or centrifugal-type lubricating-oil (LO) pump that is driven off the gas turbine. This pump takes suction from a LO reservoir and discharges synthetic oil to the bearings for the gas turbine and to control devices. Filters are also included in the LO system. In addition, electric-motor-driven gear, vane, or centrifugal pumps are ordinarily used to circulate lubricating oil through the system during start-up and cool-down periods and to serve as a backup to the attached pump. Excess oil delivered by a gas turbine's LO pump is frequently returned through a pressure regulating valve to the LO reservoir. A device is normally provided

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that sounds an alarm and, in some cases, automatically stops the flow of fuel to a gas turbine following a failure of the LO system.

Separate rotary scavenge pumps may be used to return oil that drains from a gas turbine's bearings to the LO reservoir. In addition, multiple-screw pumps are frequently used to circulate mineral oil through an independent lubrication system for a gas-turbine-propelled vessel's reduction gears and their bearings. Although one of these screw pumps may be driven off the reduction gears, the remaining reduction-gear LO pumps are usually electric motor driven. Some of the mineral oil in the reduction-gear LO system frequently passes through a heat exchanger in which it absorbs heat from the synthetic oil that lubricates the gas turbine's bearings. The reduction-gear lubricating oil also usually passes through a second heat exchanger in which it is cooled either by freshwater or seawater.

GENERATOR APPLICATIONS

Steam-Turbine-Driven Turbogenerators

AUXILIARY CONDENSATE PUMPS When the steam leaving a steam ship's turbogenerator exhausts into an auxiliary condenser, an auxiliary condensate pump is ordinarily used to remove condensate from the auxiliary condenser's hotwell and return it to the DFT. A typical auxiliary condensate pump is a vertical two-stage centrifugal pump that is similar in configuration to a main condensate pump but is smaller in size. One auxiliary condensate pump is typically furnished for each auxiliary condenser. Crossover lines are, however, often provided so each auxiliary condensate pump on a vessel can remove condensate from any of the vessel's auxiliary condensers.

AUXILIARY CIRCULATING PUMPS A single-stage centrifugal pump is often used to circulate seawater through the tubes of an auxiliary condenser that receives steam exhausted from a vessel's turbogenerator. Seawater discharged by this pump may also be directed to the generator's lubricating-oil and air coolers. One auxiliary circulating pump is typically furnished for each auxiliary condenser. Crossover lines, however, are often provided so one generator's auxiliary circulating pump can supply seawater to any of the vessel's auxiliary condensers. In addition, a line is often installed that permits seawater discharged from a vessel's auxiliary circulating pumps to be directed, in case of a main circulating pump failure, to a main condenser.

LUBRICATING-OIL PUMPS A typical turbogenerator is fitted with an internal or external gear-type lubricating-oil (LO) pump that is geared to and driven by the turbine. The pump ordinarily takes its suction from a sump located below the turbine's reduction gears and discharges lubricating oil to bearings, the turbine's reduction gears, and the governor that controls the flow of steam to the turbine. A separate hand-operated or electric-motor-driven gear pump may also be provided to enable lubricating oil to be circulated through the LO system during generator start-up and shutdown periods.

A duplex strainer, a cooler, and one or more relief valves are often included in a turbogenerator's LO system. In addition to protecting the system from overpressurization, the relief valves act as backpressure valves that maintain the desired LO pressure in various parts of the system. This arrangement permits lubricating oil sent to the governor to be at a higher pressure than the oil directed to bearings and reduction gears. A turbogenerator is ordinarily fitted with a trip that stops the flow of steam to the turbine if the LO pressure drops below a preset value.

Diesel-Driven Generators

JACKET-WATER CIRCULATING PUMPS A diesel engine that drives a generator is often furnished with an attached centrifugal-type jacket-water circulating pump that circulates freshwater through passages in the engine's cylinders and cylinder heads, the engine's

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turbocharger, and the engine's lubricating-oil cooler. In addition, the jacket-water system may include a preheater that is used to warm-up the water prior to engine start-up. When the jacket water is air cooled, it will also pass through a radiator. Alternatively, the jacket water may be cooled by seawater or freshwater in a heat exchanger. When seawater is used as the cooling medium in a jacket-water cooler, the seawater may be circulated by a centrifugal pump that is attached to and driven by the diesel engine or by a separate electric-motor-driven centrifugal pump.

FUEL-OIL PUMPS A diesel engine that drives a generator frequently has an attached rotary gear-type fuel-oil (FO) pump that is mounted on and driven by the engine. The pump typically takes suction from a daily service tank and delivers fuel oil to the engine's injector pumps. Excess oil delivered to the engine is ordinarily returned through a recirculation line to the service tank. A filter is typically included in the system. A separate hand-operated FO pump that can be used to prime the engine-driven FO pump (for example, following maintenance performed on the FO system, such as filter replacement) may also be provided. A diesel engine's FO supply line is often fitted with a valve that can be closed from outside the machinery space to stop the engine in case of a fire or other emergency.

LUBRICATING-OIL PUMPS A diesel engine that drives a generator is usually fitted with an attached rotary gear-type lubricating-oil (LO) pump that is mounted on and driven by the engine. The pump typically takes suction from a sump located below the engine and discharges oil to the engine's bearings. A filter and cooler are also ordinarily included in an engine's LO system. In addition, a hand-operated or air-motor-driven rotary-type LO pump is often provided to permit the engine's bearings to be prelubricated prior to startup. A diesel engine that drives a generator is often protected by a trip that stops the engine if the pressure in the LO system drops below a preset value.

AUXILIARY APPLICATIONS

Ship's Service Pumps

FUEL-OIL TRANSFER PUMPS A fuel-oil (FO) transfer pump is used to remove fuel oil from one or more of a vessel's FO storage tanks and to transfer the oil to settling tanks, to other storage tanks (i.e., to adjust the list, trim, or stability of the vessel), or to an above-deck connection through which the oil can be directed ashore or to another vessel. Most vessels have at least two pumps that can be used to transfer fuel oil. Although horizontally and vertically mounted multiple-screw rotary pumps are frequently used in this application, steam-driven reciprocating-piston pumps are sometimes used to transfer fuel oil. FO transfer pumps are often installed low in a vessel to improve suction conditions.

Some rotary FO transfer pumps are driven by two-speed electric motors or steam turbines. When this is the case, the capacity of fuel transferred can be changed by altering the speed of the transfer pump's driver. The capacity delivered by a FO transfer pump during constant-speed operation can often be reduced by opening a valve in a bypass line through which a portion of the oil discharged by the pump is recirculated back to the suction line. A FO transfer pump is usually protected with a suction strainer and a discharge relief valve and is often fitted with remote controls that enable the pump to be stopped from outside of a vessel's machinery spaces.

On a steam-propelled vessel with oil-fired boilers, a FO service pump typically takes suction directly from the fuel-oil settlers and discharges oil to burners in the vessel's boilers. On a diesel-propelled vessel, however, fuel oil in the settlers is ordinarily passed through one or more purifiers as it is transferred to separate daily service tanks from which the engine FO supply or booster pumps take suction. The rotary gear-type pump that frequently delivers fuel oil to a purifier is often mounted on and driven by the purifier. In some cases, a second attached rotary pump is used to boost the pressure of the clean

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oil leaving a purifier. Alternatively, independent electric-motor-driven pumps are sometimes used with FO purifiers.

FRESHWATER COOLING PUMPS To reduce corrosion from seawater, some vessels are fitted with a central freshwater cooling system. When this is the case, single-stage centrifugal pumps are generally provided to circulate freshwater through two separate cooling loops: a low-temperature loop and a high-temperature loop. Two centrifugal pumps that are each capable of delivering the full-load capacity are frequently installed in each freshwater loop. Smaller pumps may also be provided for startup and in-port use.

A low-temperature freshwater cooling pump ordinarily circulates freshwater through a vessel's various condensers, oil coolers, and air coolers. Jacket-water coolers for a vessel's auxiliary diesel engines may also be included in the low-temperature loop. In addition, freshwater in the low-temperature loop typically passes through a seawater-cooled heat exchanger. On a diesel-propelled vessel, the high-temperature freshwater cooling pump serves as the jacket-water-circulating pump and typically circulates fresh water through the propulsion-engine jackets, turbochargers, and a freshwater generator (evaporator). On some vessels, freshwater in the high-temperature loop also passes through a heat exchanger in which it is cooled by freshwater in the low-temperature loop. Alternatively, however, the need for this heat exchanger is eliminated in some central freshwater cooling systems with a control valve that allows some of the freshwater in the low-temperature loop to enter, mix with, and cool the hotter water in the high-temperature loop.

SEAWATER SERVICE PUMPS A seawater service pump takes suction from a sea chest and supplies seawater to heat exchangers in which this water serves as the cooling medium. This can sometimes include refrigeration and air-conditioning condensers, various lubricating-oil coolers, and air-compressor coolers. On newer vessels that have a central freshwater cooling system, however, the seawater cooling pumps supply seawater only to large freshwater coolers. With either arrangement, after leaving the heat exchangers that it passes through, the seawater is ordinarily directed overboard.

Two or more horizontally or vertically mounted electric-motor-driven single-stage centrifugal seawater service pumps (these pumps are also sometimes referred to as seawater cooling pumps or auxiliary seawater pumps) are usually installed on a vessel. A seawater pump is typically located low in a vessel so it can operate with a flooded suction. In some cases, the pump's suction line may be connected to both a lower sea chest and an upper sea chest. The use of the lower sea chest, which often receives seawater through an opening in the bottom of a vessel's hull, reduces the potential for air to enter a seawater pump's suction line (especially when the vessel is rolling in rough seas). However, when the vessel is in shallow water, using the upper sea chest, which is typically connected to an opening in the side of a vessel's hull, can reduce the amount of silt, mud, and other contaminants entering the seawater-pump suction. In addition to the sea-chest connections, on a diesel-propelled vessel, the largest seawater-cooling pump may also have an emergency bilge suction line through which the machinery space can be dewatered in case of flooding.

To help protect a seawater pump from foreign material that may enter a sea chest, a strainer is frequently installed in a seawater pump's suction line. In addition, when a mechanical-seal flushing line is used with a seawater pump, it is sometimes fitted with a cyclone abrasive separator. In a typical installation, a recirculation line from the discharge side of the pump's casing is connected to the inlet on the side of the separator. Clean water leaving the top of the separator is directed to the mechanical seal's flushing connection, which is usually in the seal's gland. Dirty water discharged from the bottom of the separator is returned to the suction side of the pump's casing.

LUBRICATING-OIL TRANSFER PUMPS Rotary gear pumps are frequently used to transfer lubricating oil from a vessel's lubricating-oil (LO) storage tanks to tanks in various locations throughout the vessel where the oil is stored for use in auxiliary machinery. In addition, a rotary pump may be used to add lubricating oil directly to oil-lubricated machinery.

Most vessels have one or more LO purifiers that can be used to remove water and other contaminants from the lubricating oil contained in machinery LO sumps (for example, LO sumps for diesel engines, turbogenerators, and main-propulsion-turbines). A LO purifier

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may also be used to transfer lubricating oil from a vessel's LO storage tanks to separate LO settling tanks. A rotary gear pump is often used to deliver lubricating oil to a LO purifier. In many cases, the pump is mounted on and driven by the purifier. A second attached rotary pump may also be used to boost the pressure of the clean oil leaving a purifier. Alternatively, independent electric-motor-driven pumps are sometimes used with LO purifiers.

STERN-TUBE LUBRICATING-OIL PUMPS An electric motor-driven rotary screw, gear, or vane pump is required to circulate lubricating oil through a vessel's stern-lube bearings and seals when these components are oil lubricated. One pump is typically provided for each propulsion shaft. The pressure on the discharge side of a stern-tube lubricating-oil (LO) pump is ordinarily established by the elevation of a head tank that is connected to the system. (Stern-tube LO pumps are not required on vessels that have seawater-lubricated stern-tube bearings and seals.)

AUXILIARY AND EXHAUST-GAS (WASTE-HEAT) BOILER PUMPS Diesel-propelled vessels are frequently fitted with exhaust-gas boilers in which heat contained in the engine exhaust-gas is utilized to generate steam for various purposes, such as heating fuel, liquid cargo, and water. On some vessels, a portion of the steam produced may even be superheated and supplied to turbine-driven equipment, such as turbogenerators. To permit steam to be generated during periods of low engine load or when an engine is secured (for example, in port), most of these vessels are also fitted with auxiliary oil-fired boilers that can operate alone or parallel to the exhaust-gas boilers.

In a typical installation, a feed pump is used to return condensate from a drank tank to the boilers. In addition, a separate pump is generally required to circulate water through the boiler generating tubes. Furthermore, when condensing turbines are installed in an auxiliary steam system, a condensate pump is used to transfer condensate from the condenser's hotwell to the drain tank. Electric-motor-driven, single and multistage centrifugal and regenerative turbine pumps are typically used in these applications.

A fuel oil (FO) service pump is usually required to supply fuel oil to an oil-fired auxiliary boiler. Electric-motor-driven gear pumps are often used in this application.

HYDRAULIC-SYSTEM PUMPS Positive-displacement pumps are ordinarily used to pressurize and circulate the hydraulic fluid in a vessel's hydraulic systems. Typical examples of shipboard hydraulically powered machinery include steering gear, anchor windlasses, cranes, hatch covers, and valves. The hydraulic pumps, which can be installed in a self-contained system that is an integral part of the hydraulically powered component or in a larger central system that provides fluid to all of the hydraulic equipment on a vessel, are usually driven by electric motors or diesel engines.

In a constant-pressure hydraulic system, one or more variable-displacement rotating-piston pumps are often used to supply hydraulic liquid at essentially a constant pressure to the components that are powered by the system. If the capacity delivered by a pump exceeds the system's demand, the pump discharge pressure will include and an automatic controller will reduce the pump's stroke and the capacity pumped until the discharge pressure returns to the set point. If the system demand increases and the pump discharge pressure drops, the pump's stroke and the capacity delivered will be increased. The pump stroke and the pumped capacity are, therefore, automatically adjusted to match the demand on the hydraulic system. In some systems, hydraulic pumps may also be automatically started and stopped as needed.

The hydraulic fluid in a constant-flow variable-pressure hydraulic system is typically circulated by gear, vane, multiple-screw, or fixed-displacement rotating-piston pumps. These pumps, which often operate continuously, take suction from a sump or tank and deliver pressurized oil to the components powered by the system. If the capacity delivered by a system's pumps exceeds the system requirement, excess oil is recirculated through an unloading valve back to the sump.

FIRE PUMPS A fire pump takes suction from a vessel's sea chest and discharges seawater through a fixed fire main to hydrants located throughout the vessel. Most vessels have at least two fire pumps that are each capable of delivering a minimum capacity of seawater

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simultaneously to a specified number of the most remote hydrants at a specified pressure. Although fire pumps are sometimes used for other purposes, at least one fire pump on a vessel should always be available to provide water to the fire main. In addition, some of a vessel's fire pumps, together with their sea connections and sources of power, are ordinarily installed in separate spaces and arranged so a fire in any one location cannot incapacitate all of the vessel's fire pumps. Also, controls are sometimes provided that enable a fire pump's suction and discharge valves to be opened and the pump to be started from outside of the machinery space. A pressure gage and a relief valve are usually installed in a fire pump's discharge line. (See Section 12.4.)

Single-stage and, in some cases, two-stage centrifugal pumps may be used in fire service. Vertically mounted centrifugal fire pumps are generally driven by electric motors. Horizontal fire pumps, however, can be driven by electric motors, steam turbines, or diesel engines. In addition, smaller centrifugal fire pumps and the electric motors that typically drive them are sometimes furnished in a close-coupled configuration. A centrifugal fire pump is ordinarily installed low in a vessel so it operates with a flooded suction. Alternatively, a vertical turbine pump is sometimes used in this application, which enables the fire-pump impellers to be submerged within a tank.

Some vessels also carry portable fire pumps that can be moved throughout the vessel by the crew. A typical portable unit consists of a single-stage end-suction centrifugal pump that is close-coupled to a gasoline engine. An integral vacuum priming pump is often included with a portable unit to enable the fire pump to be used in areas of the vessel that are above the waterline.

BILGE PUMPS Bilge pumps remove liquid from machinery-space bilges, tank tops, the shaft alley, and watertight compartments located throughout a vessel. Fluid discharged from a bilge pump is typically directed, depending on its content and applicable regulations, to an oily-waste holding tank or overboard. Most vessels have more than one bilge pump. In addition to a common suction main that is connected to branches that lead to suction wells (sometimes referred to as rose boxes) located throughout a vessel, one or more independent bilge-pump suction lines are generally provided in machinery spaces. In addition, a vessel's bilge pumps, together with their sources of power, may be installed in various watertight compartments throughout a vessel so the flooding of one compartment will not incapacitate all of the vessel's bilge pumps. Alternatively, some vessels have an emergency bilge pump that is suitable for use even when the pump is submerged. The submersible bilge pump, which is normally powered through the vessel's emergency switchboard, can normally be operated remotely from outside of the vessel's machinery spaces.

Horizontally and vertically mounted centrifugal pumps are frequently used for bilge service. Centrifugal bilge pumps are frequently driven by electric motors. Some bilge pumps, however, are driven off a vessel's propulsion machinery. Because bilge pumps frequently operate with a suction lift, a non-self-priming centrifugal bilge pump is usually connected to a vacuum-priming pump. The vacuum pump can be part of a central system that is used to prime multiple pumps on a vessel (Figure 9), or it may be a dedicated unit that is used only with the bilge pump. When a dedicated vacuum pump is provided, it is sometimes electric-motor-driven and mounted on the same baseplate as the bilge pump that it primes. Alternatively, a dedicated vacuum pump may be mounted on and driven directly off the bilge pump.

To eliminate the need for a separate priming pump, self-priming centrifugal pumps are sometimes used in bilge service. A typical self-priming centrifugal pump (Figure 10) has a casing with an enlarged suction chamber that retains liquid whenever the unit is stopped. When the pump is started with air in its suction line, the liquid in the suction chamber is pumped through the impeller and enters a discharge chamber installed at the top of the casing. The evacuation of the suction chamber creates a vacuum that draws air from the suction line into the pump. The liquid in the discharge chamber is returned to the pump's impeller through either an internal port or an external recirculation line and mixes with this air. The liquid-and-air mixture is then pumped through the impeller and into the discharge chamber where the air is vented from the pump. The liquid is again returned to the impeller, and the cycle is repeated. Each time the stored liquid is recirculated through the pump, an additional amount of air is removed from the suction line.

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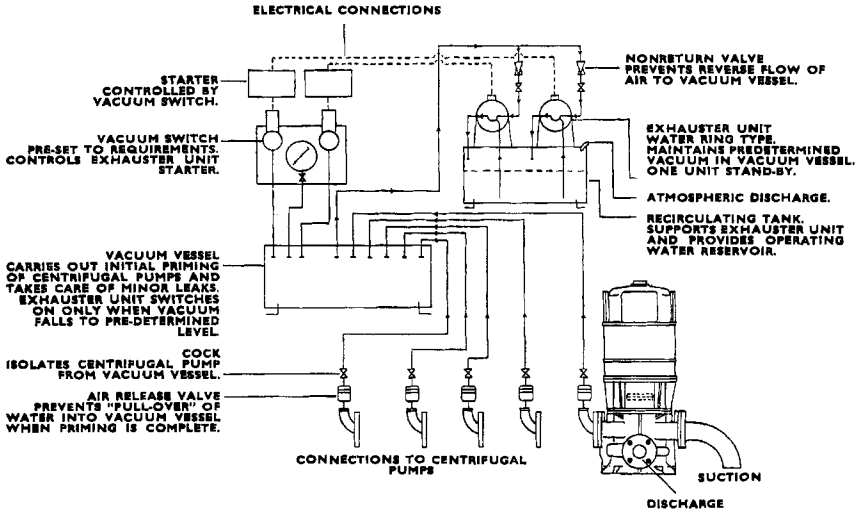


FIGURE 9 Central priming system (Flowserve Corporation)

1 Casing
2 Suction chamber
3 Recirculation port
4 Impeller
5 Discharge chamber
6 Shaft
7 Impeller nut
8 Impeller-nut insert
9 Gland
10 Packing
11 Lantern ring (seal cage)

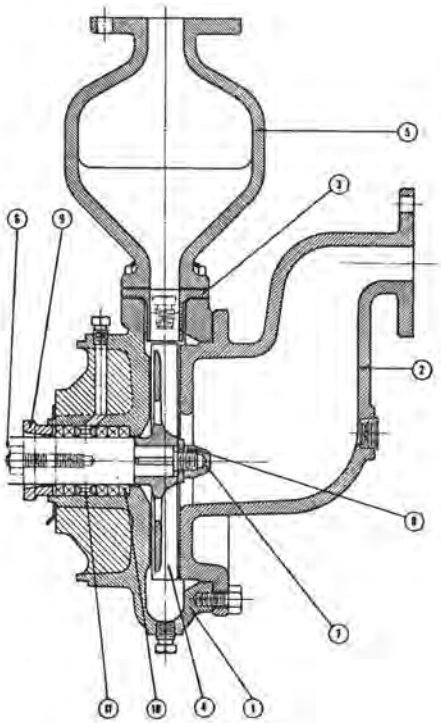


FIGURE 10 Self-priming centrifugal pump (Flowserve Corporation)

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After this occurs, the pump will discharge liquid in a normal fashion. However, priming times can be excessive when large amounts of air must be removed from a suction line; consequently, self-priming centrifugal pumps are usually only used for bilge service when the length of the bilge suction piping is relatively short.

Additional types of bilge pumps that do not require priming pumps include motor-driven sump pumps that are submerged within the bilge drain wells; rotary-type vane pumps; air-operated diaphragm pumps; and motor-, steam-, or air-driven piston-type reciprocating pumps.

An oily-water separator (OWS) is used to remove oil from bilge water so the water can be discharged overboard. A typical OWS is fitted with one or more dedicated pumps that take suction either from an oily-water holding tank or directly from bilge suction wells and circulate the oily water through the separator, discharge clean water overboard, and transfer separated oil to a waste-oil tank. Electric-motor-driven progressing-cavity pumps are often used with an OWS.

BALLAST PUMPS A ballast pump is used to transfer seawater into and out of a vessel's ballast tanks to adjust list, trim, draft, and stability. Ballast pumps may also be used during a voyage to exchange water contained within ballast tanks to prevent the introduction of nonindigenous aquatic species into coastal and inland waterways. A ballast pump must, therefore, have the capability to take suction either from a sea chest or from ballast tanks that are being emptied. In addition, it must be possible to direct seawater discharged by a ballast pump either to ballast tanks that are being filled or overboard. Furthermore, the seawater discharged by a ballast pump may be used as the motive fluid in eductors that are provided to strip ballast tanks. On some vessels, pumps used for ballasting and deballasting may also be used for other purposes, such as bilge dewatering (that is, a bilge and ballast pump) or fire fighting. Vessels with large segregated ballast tanks, such as many tankers, however, frequently have dedicated ballast pumps.

Horizontally and vertically mounted single-stage centrifugal pumps that are installed in a vessel's machinery space are often used for ballast service. Although many of these pumps are driven by electric motors, larger ballast pumps are sometimes driven by steam turbines. A centrifugal ballast pump is normally located low in a vessel so it will operate with a flooded suction when it is lined up to the sea chest or begins to empty a ballast tank that is full. As the water level in a ballast tank being emptied drops, however, the submergence of the pump's impeller is continuously reduced. To enable it to be reprimed if suction is lost prematurely during operation with a suction lift, a centrifugal ballast pump may be connected to a priming system.

As an alternative to centrifugal ballast pumps installed in a machinery space, hydraulic-motor-driven centrifugal ballast pumps that are submerged directly in the ballast tanks are used on some vessels. In addition, some vessels have vertical line-shaft deep-well ballast pumps. So the pump can take suction from multiple locations, a deep-well ballast pump is frequently installed in a suction can that is connected to the ballast-system suction piping. To enable air and vapor to be removed from the can and suction piping, a deep-well ballast pump is often fitted with self-priming valves.

Hotel Service Pumps

FLASH-DISTILLING-PLANT PUMPS Many steam ships have multistage flash distilling plants that are used to generate freshwater from seawater. Several electric-motor-driven single-stage centrifugal pumps are used with a typical flash unit. Smaller pumps are often provided in a close-coupled configuration.

A distiller-feed pump takes suction from a sea chest and supplies seawater to the distilling plant's first-stage flash chamber. This water often passes through various heat exchangers, such as a distillate cooler, distillate condensers, air-ejector condenser, and seawater heater before it enters the flash chamber. The distiller-feed pump is usually located sufficiently below the vessel's waterline so it operates with a flooded suction.

The hot high-salinity brine remaining in a flash distilling plant's last stage is typically removed by a brine pump that discharges it overboard. A line is often provided to permit

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a brine pump's shaft seals to be flushed with seawater discharged from the distiller-feed pump. In addition, a vent line is typically connected from a brine pump's suction to the last-stage flash chamber.

The freshwater or distillate produced by a flash distilling plant is removed from the last-stage distillate condenser by a distillate pump. During normal operation, water discharged from this pump ordinarily passes through a cooler and is then directed to a vessel's distilled-water (reserve-feedwater) or potable-water tanks. A salinity cell in the pump discharge line, however, will typically trip a three-way valve that dumps the distillate to the bilge if salinity of this water is excessive. A vent line may be connected from a distillate pump's suction to the last-stage distillate condenser.

An additional pump is sometimes provided to remove condensate from the hotwell of a flash distilling plant's seawater heater and transfer it to a freshwater drain tank. Alternatively, however, this condensate may be returned directly to a main or auxiliary condenser through a vacuum-drag line. With this latter arrangement, the feed-heater condensate pump is not required.

Because the distillate, brine, and feed-heater condensate pumps each take their suction from a chamber in which the pumped liquid is at or near its vapor pressure, pumps used in these applications should have low NPSH requirements. In addition, to increase the submergence of their impellers, these pumps are often located as far below the distilling plant assembly as practicable.

HEAT-RECOVERY DISTILLING-PLANT PUMPS It is common for a diesel-propelled vessel to be fitted with one or more heat-recovery distilling plants in which jacket water from the vessel's diesel engines serves as the heating medium. In a typical heat-recovery distilling plant, this jacket water is pumped through either a plate or a submerged-tube evaporator.

Seawater is often delivered to a heat-recovery distilling-plant by a dedicated distiller-feed pump (sometimes referred to as an eductor or ejector feed pump) that takes its suction from a sea chest. In some units, this seawater initially passes through the distilling plant's condenser where it absorbs heat from the distilled vapor generated in the evaporator. A portion of this seawater then enters the evaporator section of the shell as feed. The remaining portion of the seawater frequently serves as the motive fluid for an eductor that removes brine from the evaporator and air from the condenser portion of the distilling plant's shell. After leaving the eductors, this seawater is directed overboard with the brine and air. (In some units, the mixture of seawater, brine, and air passes through the distilling plant's condenser before being discharged overboard.) The feed pump is typically located low in a vessel to enable it to operate with a flooded suction.

A distillate pump is used to remove the freshwater collected in a heat-recovery distilling plant's condenser and discharge it either to the vessel's freshwater tanks or, when the salinity is high, to the bilge. Both the eductor-feed pump and the distillate pump are generally electric-motor-driven single-stage centrifugal-type pumps. In addition, they are often furnished in a close-coupled configuration.

POTABLE-WATER PUMPS A potable-water pump typically takes suction from a vessel's potable-water or domestic tanks and discharges freshwater either directly to sinks, showers, and other potable-water fixtures located throughout the vessel or to an air-charged pressure tank, sometimes referred to as a hydrophore or a hydropneumatic tank, that is frequently included in a potable water system. Most vessels have at least two full-capacity potable-water pumps. Electric-motor-driven single-stage centrifugal pumps are often used in this application. In addition, regenerative turbine pumps are sometimes used. Smaller pumps are often furnished in a close-coupled configuration. To improve suction conditions, a potable-water pump may be installed as far as practicable below a vessel's potable water tanks.

In a typical potable-water system in which a hydrophore is connected to the pump discharge line, a potable-water pump may be cycled on and off automatically by a pressure switch installed on the hydrophore. During periods when the pump is not running, any potable-water usage will result in a drop of the water level within the hydrophore, the expansion of the compressed air located in the upper portion of this tank, and a reduction in the hydrophore pressure. However, the force exerted by the compressed air ordinarily

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prevents the potable-water system pressure from dropping too suddenly. If the potable-water usage continues, the pressure exerted by the compressed air within the hydrophore will continue to drop until it reaches the cut-in pressure for the potable-water pump's motor. At this point, the pump will be started and the water level and pressure within the hydrophore will normally increase. This will continue until the pressure switch's cut-out pressure is reached and the potable-water pump is stopped.

Although it is less common, on a vessel with a high potable-water demand, a potable-water pump may be operated continuously. When a potable-water pump operates continuously, however, a recirculation line is typically provided to prevent the pump from overheating during periods of low water usage.

HOT-WATER CIRCULATION PUMPS Some of the freshwater discharged from a vessel's portable-water pumps is circulated through a heater and is then directed to sinks, showers, and other hot-water fixtures located throughout a vessel. One or more hot-water circulation pumps are ordinarily used to recirculate unused water contained in the hot-water distribution piping through the heater so the water will remain hot. Electric-motor-driven single-stage centrifugal pumps are typically used in this application. In addition, most hot-water circulating pumps, which generally operate continuously, are furnished in a close-coupled configuration.

SANITARY PUMPS Seawater is sometimes used as the flushing medium in bathrooms on a vessel. When this is the case, a sanitary or flushing-water pump may be provided to take suction from a sea chest and discharge seawater to the vessel's flushometer valves. Two electric-motor-driven single-stage centrifugal pumps are frequently used in this application. When each pump is sized to meet peak-demand requirements, one pump can be cycled on and off automatically by a discharge-pressure switch while the second pump serves as a standby unit. To reduce the number of sanitary-pump starts and stops, an air-charged pressure tank similar to the potable-water-system hydrophore described previously is often included in a sanitary system. A sanitary pump is frequently located low in a vessel so it can operate with a flooded suction.

SEWAGE PUMPS When a vessel has a sewage holding tank, a sewage pump typically takes suction from the holding tank and discharges the sewage and waste water, based on applicable regulations, overboard or to an above-deck connection for transfer ashore. Electric-motor-driven single-stage centrifugal pumps are frequently used in this application. Some of these pumps have casings with large waterways and are fitted with special non clog impellers. Alternatively, some vessels have sump-type centrifugal pumps that are submerged within the sewage tank and are driven by submersible motors or through vertical line shafting by above-tank motors. With either arrangement, two pumps are often provided for each holding tank.

Instead of a sewage holding tank, many vessels now have an onboard sewage treatment plant, referred to as a marine sanitation device (MSD). At least two pumps are ordinarily provided with each MSD to discharge the treated effluent overboard. Standard electric-motor-driven single-stage centrifugal pumps are frequently suitable for this purpose.

Some vessels also have one or more lift stations in which sewage and wastewater is collected before being transferred to an MSD or a larger sewage holding tank. Macerator pumps are frequently used to transfer a lift station's contents to the MSD or holding tank. A macerator grinds the sewage and breaks up solids into small particles that are more easily treated. Lift-station transfer pumps are usually started and stopped automatically by a float switch in the lift station. Two pumps are often provided for each lift station.

AIR-CONDITIONING CHILLED-WATER PUMPS Some vessels utilize freshwater as a secondary refrigerant for air conditioning. With this arrangement, a chilled-water pump circulates the fresh water through a chiller where it is cooled by the system's primary refrigerant. The water then passes through duct-mounted cooling coils, where it absorbs heat from air being supplied to temperature-controlled spaces located throughout the vessel. The chilled water may also be used to cool electronic components. Electric-motor-driven single-stage

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centrifugal pumps are frequently used in this application. A pressurized or elevated expansion tank that is typically installed on the suction side of the system maintains a minimum pressure at the inlet to the pumps.

Cargo Pumps and Associated Pumps

Cargo pumps are used to discharge the liquid cargo transported in a vessel's tanks. The types of vessels that carry liquid bulk cargo include ultra-large and very-large crude carriers (ULCC's and VLCC's), multi-petroleum product and chemical carriers, and liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers. In addition, some freighters and supply vessels have tanks in which they can carry a limited amount of liquid cargo.

As liquid cargo is discharged from a vessel, the level in a cargo tank being emptied and, therefore, the suction head to the cargo pump are continuously reduced. This, as well as the fact that many cargoes have relatively high vapor pressures, means cargo pumps must often operate with relatively low values of available net positive suction head (NPSH). In addition, as a cargo tank is being emptied, vortices can form on the surface of the liquid in the tank, and air or inert gas (the space in a cargo tank above a flammable liquid is typically filled with inert gas) from the tank's atmosphere can be drawn through a vortex and into the cargo pump's suction. To reduce the potential for cavitation and vortex formation, the flow rate discharged by a cargo pump is frequently reduced by the use of either manually controlled or motorized remote-operated discharge valves during the latter stages of a pump-out cycle.

The materials used in the construction of a cargo pump's wetted components must be compatible with all of the fluids that the pump will discharge. In addition to the cargoes that the vessel will carry, this can also include seawater if the cargo pumps will be used to remove slops during tank washing. In addition, if fluids that are flammable or explosive will be pumped, components with contacting surfaces should be constructed from nonsparking materials.

A description of some of the different types of pumps used in marine cargo service follows.

CENTRIFUGAL CARGO PUMPS Single-stage centrifugal pumps are used to discharge liquid cargo on many crude-oil carriers. In addition, they are used on some large product carriers that transport a limited number of different liquid cargoes. Typically, three or four centrifugal cargo pumps are installed in one or more pump rooms located low in a vessel. Each pump can frequently remove cargo from multiple tanks through interconnected suction piping.

A centrifugal cargo pump may be furnished with a double-suction impeller mounted in a between-bearings configuration (Figure 11). Pumps of this type, which must be fitted with two shaft seals, are installed in both horizontal and vertical configurations. Alternatively, a vertically mounted centrifugal cargo pump design with an impeller that is overhung on the end of a shaft (Figure 12) is used on some vessels. This latter configuration enables both of the pump's external bearings and the single shaft seal to be located above the impeller, which facilitates maintenance. A centrifugal pump's bearing housings are often fitted with explosion-proof intrinsically safe resistance temperature detectors (RTDs) that permit the operator to detect a hot bearing before it leads to a serious casualty. In addition to the suction and discharge connections, one or more vents are often provided in the suction area of a centrifugal cargo pump's casing so vapor and gas mixed with the liquid entering the pump during a pump-out can be removed from the casing.

Although centrifugal cargo pumps are frequently driven by steam turbines, they are also sometimes driven by diesel engines or electric motors. To isolate it from the explosive vapor that is often present in a pump room, a centrifugal cargo pump's driver is usually installed in a separate machinery space located adjacent to the pump room. With this arrangement, each driver is typically flexibly coupled to the cargo pump that it drives through an intermediate jackshaft. The opening in the pump-room bulkhead (for a horizontal pump, Figure 13a) or overhead (for a vertical pump, Figure 13b) through which the jackshaft passes is ordinarily sealed with a gas-tight stuffing box to prevent explosive

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1	Casing	6	Thrust bearing
2	Shaft	7	Bearing housings
3	Casing wearing ring	8	Line bearing
4	Impeller	9	Auxiliary stuffing box
5	Vent / stripping connection	10	Shaft nut
		11	Mechanical seal

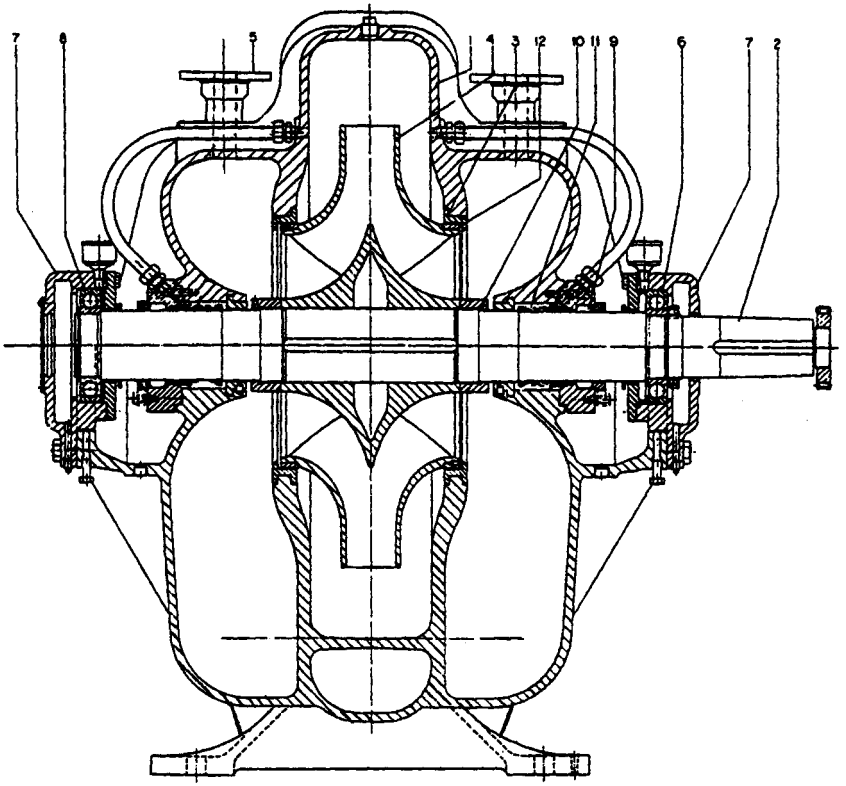


FIGURE 11 Between bearings centrifugal cargo pump (Flowserve Corporation)

vapor in the pump room from entering the machinery space. A thrust bearing is also frequently provided in the driver to support the weight of a jackshaft used with a vertically mounted cargo pump.

As the liquid level in a cargo tank being emptied approaches the inlet to the suction tailpipe (referred to as a suction strum), air or inert gas from the tank's atmosphere can be drawn into the cargo pump through vortices that form on the surface of the liquid. This entrained gas can cause a centrifugal cargo pump to lose suction before the cargo tank has been emptied. Although a separate stripping pump may then be used to remove the liquid remaining in the cargo tank, stripping pumps typically deliver much lower capacities than large centrifugal cargo pumps. Therefore, self-priming/stripping systems are sometimes employed to increase the amount of cargo that can be discharged by a centrifugal cargo pump. A recirculation-priming system and an automated-vacuum-stripping system are two types of systems that are used.

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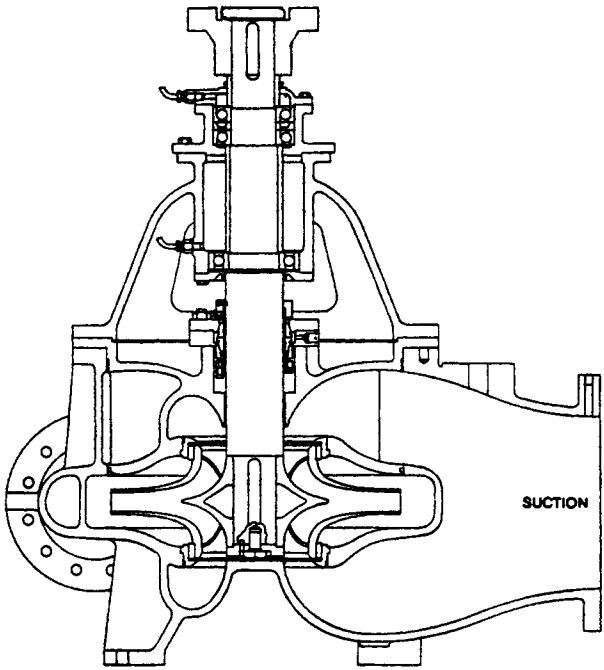


FIGURE 12 Vertical overhung centrifugal cargo pump (Flowserve Corporation)

In a typical recirculation-priming system, a recirculation tank is mounted between the cargo-system suction piping and the inlet to the cargo pump. In addition, one or more priming valves are mounted in the cargo discharge line near the outlet from the pump. Also, a recirculation line is connected from these valves back to the recirculation tank at the pump's inlet, a check valve is installed in the cargo-pump discharge piping above and

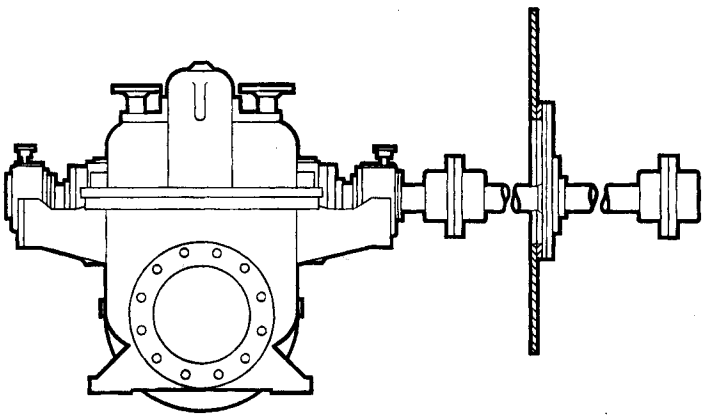


FIGURE 13A Horizontal centrifugal cargo pump with jackshaft (Flowserve Corporation)

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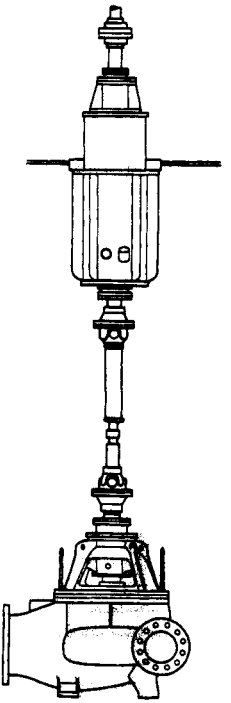


FIGURE 13B Vertical centrifugal cargo pump with jackshaft (Flowserve Corporation)

downstream from the priming valves, and a vent line with a second check valve is connected from the top of the recirculation tank to the cargo-pump discharge line just upstream of the discharge check valve. A branch from the recirculation tank's vent line is ordinarily piped to the vent/stripping connections in the cargo pump's casing.

At the beginning of a pump-out, the recirculation tank is filled with liquid cargo and the priming valves are closed. However, if, as the level in the cargo tank being emptied is reduced, the percentage of gas in the liquid being transferred increases sufficiently for the cargo pump to lose suction, the priming valves in the discharge line will open. This permits the cargo contained in the discharge line between these priming valves and the discharge check valve, which closes when the flow through it stops, to be returned through the recirculation line to the recirculation tank. As this liquid drains through the open priming valves, a vacuum is created in the discharge line. The gas contained in the recirculation tank is displaced by the returning liquid and is drawn by the discharge-line vacuum through the vent line and into the portion of the discharge piping that has been evacuated.

As the recirculated liquid cargo enters the recirculation tank, the liquid level within the tank rises until the submergence of the cargo pump's impeller is sufficient for the pump to regain suction. After this occurs, the flow of liquid through the pump will resume, the priming valves and the check valve in the vent line will close, and the gases that have accumulated in the discharge line will be forced through the discharge check valve. The priming cycles will continue until enough gas has been removed from the cargo suction line for the cargo pump to operate normally. As the level in the cargo tank being emptied continues to be reduced, the amount of gas entering the suction strum will increase, and the cargo pump will lose suction more frequently. When the amount of liquid remaining in the cargo tank is not sufficient for the cargo pump to be reprimed, the pump should be stopped.

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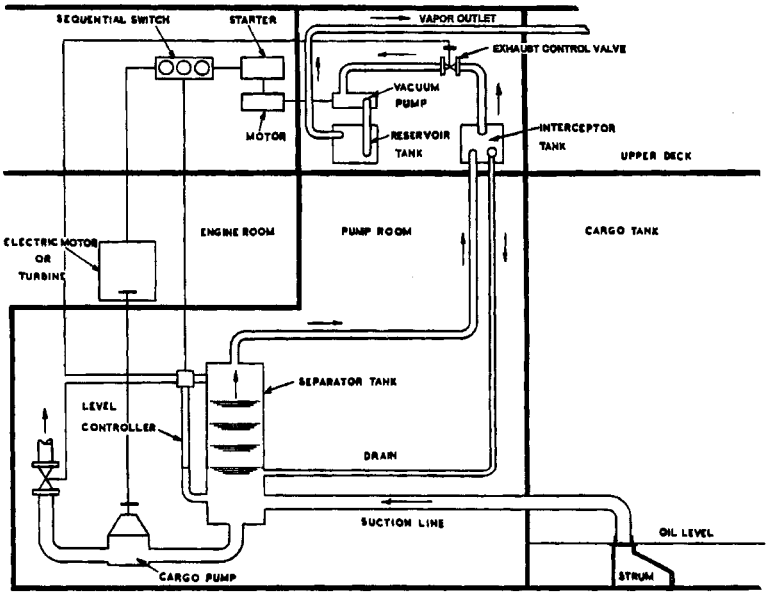


FIGURE 14 Automated-vacuum-stripping system (Flowserve Corporation)

The key components used in a typical automated-vacuum-stripping system (Figure 14) include a separator tank located at the cargo pump's inlet, a vent line connected to the top of the separator tank, an automatic exhaust control valve mounted in the vent line, one or more electric-motor-driven vacuum pumps, a cargo-pump discharge valve with actuator, and associated controls. As with the recirculation system, a branch from the separator tank's vent line is ordinarily piped to the vent connections in the cargo-pump casing.

At the beginning of a pump-out, the separator tank is ordinarily filled with liquid cargo. However, as the pump-out progresses, the liquid level in the cargo tank being emptied and the cargo-pump's suction pressure will both be reduced. Eventually, cargo vapor will typically begin to accumulate in the top of the separator tank and the liquid level in the separator tank will drop. At a specified point, the exhaust control valve in the vent line opens to permit vapor in the separator tank to pass through the vent line before it can enter the cargo pump. If the venting process is not sufficient, the vacuum pump will start and draw vapor out of the separator tank. After the vapor has been removed from the separator tank, the separator-tank liquid level may rise sufficiently for the vacuum pump to stop and the exhaust control valve to close. The exhaust control valve will, however, reopen and the vacuum pump will restart automatically if, because of the reaccumulation of vapor in the top of the separator tank, the separator tank level again drops. The exhaust control valve will continue to open and close and the vacuum pump will continue to cycle on and off automatically, as needed, based on the level in the separator tank.

With further reductions in the cargo-tank liquid level, the amount of cargo that vaporizes in the cargo-pump suction line and separator tank will increase. In addition, as the cargo-tank liquid level approaches the suction strum, vortices will typically form on the surface of the liquid in the cargo tank and draw gas from the tank's atmosphere into the suction line. As the percentage of vapor and entrained gas in the liquid cargo being unloaded increases, the frequency of the exhaust-control-valve/vacuum-pump cycles described above will increase. Eventually, near the end of the pump-out, it will usually become necessary for the exhaust control valve to remain open and for the vacuum pump to operate continuously.

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In addition to operating the exhaust control valve and the vacuum pump, the automated-vacuum-stripping system will normally automatically throttle the cargo pump's discharge valve whenever the vacuum pump is started. In addition, when permitted by the type of driver used, the cargo pump's operating speed may be reduced. Controls may also be included in the system to automatically shut down the cargo pump driver when the pump-out has been completed.

DEEP-WELL CARGO PUMPS Vertical line-shaft deep-well pumps are used to discharge liquid cargo on some multi product and chemical carriers. Therefore, a deep-well cargo pump must frequently be suitable to transfer a wide range of liquids having different specific gravities, vapor pressures, viscosities, and temperatures. Some cargoes, such as lubricating oils, waxes, and other viscous cargoes, may be heated to improve pumpability. With certain cargoes, such as molten sulfur, steam, or a heated liquid may even be circulated through jackets that surround the deep-well pump to prevent the cargo from solidifying within the pump. In addition, deep-well pumps are sometimes used to discharge cryogenic cargoes, such as liquefied petroleum gas (LPG).

A deep-well cargo pump can be driven by a vertical electric or hydraulic motor mounted on top of the pump's discharge head. Alternatively, a deep-well cargo pump may be driven through a right-angle gear mounted on the discharge head by a horizontal motor, steam turbine, or diesel engine. Although a deep-well cargo pump's discharge head is often mounted on a vessel's main deck, on some vessels, the deep-well-cargo-pump discharge heads are located below deck in a pump room. In addition, in some cases, a deep-well cargo pump's horizontal driver is located in an adjacent space and is coupled to the pump's right-angle gear with a jackshaft that passes through a bulkhead separating the pump's discharge head from the driver. The opening for the jackshaft in the bulkhead is ordinarily sealed with a gas-tight stuffing box so the driver can be isolated from any explosive vapor that may be emitted from the pup. Some designs are also available with submersible motors that allow the entire pump-driver assembly to be located at the bottom of the cargo tank.

Many deep-well cargo pumps are furnished with a multistage bowl assembly that has single-suction impellers. When high-vapor-pressure or high-viscosity cargo will be pumped, an inducer is sometimes mounted on the lower end of the bowl assembly's impeller shaft to reduce the pump's net positive suction head requirements. Alternatively, a deepwell pump may be fitted with a special low-NPSH first-stage impeller, or, in some cases, a double-suction first-stage impeller. A spool piece is sometimes installed between the top of a deep-well pump's bowl assembly and the lower end of its column assembly so the bowls can be removed for maintenance while the column and discharge head are still in place.

Hydraulic axial unbalance resulting from the use of single-suction impellers can result in the generation of a high axial thrust. Some deep-well pumps are fitted with a balancing device, such as a balancing drum or front and back impeller wear rings, to reduce this thrust. Alternatively, a thrust bearing is frequently used to absorb the axial thrust acting on a deep-well pump's shaft. This bearing is sometimes installed in a housing that is an integral part of the pump's discharge head. In many cases, however, the thrust bearing is in the vertical driver or right-angle gear (when a horizontal driver is used) that is mounted on top of the pump's discharge head. With this latter arrangement, the deep-well pump's top shaft is typically secured to the vertical driver or gear either with a rigid coupling when the driver or gear has a solid shaft or with an adjusting nut when a hollow-shaft driver or gear is used. It is always important that the pump shaft be raised the proper amount during assembly to prevent contact between the pump's rotating and stationary parts during operation.

Mating sections of a deep-well pump's line shaft are typically connected with threaded or keyed couplings. Radial bearings that support the line shaft are frequently mounted in brackets, sometimes referred to as spiders, that are sandwiched between mating sections of column pipe. Impeller- and line-shaft bearings are often lubricated by the pumped cargo. Consequently, bearing materials must typically be compatible with all of the fluids that will be pumped. The bearings should also be able to tolerate operation with loss of suction if the deep-well pump will be used for stripping or during tank cleaning. Although bronze bearings are common, bearings constructed from carbon, polytetrafluoroethylene (PTFE) compounds, and various composites and plastics have also been used.

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The use of a single shaft seal at the location where a deep-well pump's top shaft penetrates the discharge head is sometimes suitable. A multiple sealing arrangement consisting of two packed stuffing boxes or a double mechanical seal may be utilized, however, when a deep-well pump will discharge a volatile cargo. In addition, if a deep-well pump is fitted with a double mechanical seal, a static seal may also be provided to prevent vapor from escaping around the pump's shaft during periods when the mechanical seals are being replaced. A reservoir tank that stores liquid for seal lubrication is often mounted on the side of the discharge head of a deep-well pump that has one or more mechanical seals.

On some multi-product and chemical carriers, a separate deep-well pump is installed in each cargo tank. This arrangement eliminates the need for suction piping, and it can result in a high degree of cargo segregation. On a double-bottom vessel, the deep-well pump's suction opening is frequently submerged in a suction well located in the bottom of the cargo tank. In-tank supports are often provided to stabilize the pump during periods of pitch and roll. However, to prevent these supports from restricting thermal expansion and distorting a pump's column or line shaft as a vessel flexes, they are generally not rigidly attached to the pump.

To prevent fluid in a deep-well pump's discharge head, column pipe, and bowl assembly from draining back into a cargo tank after the pump is stopped, a deep-well cargo pump's inlet opening is sometimes fitted with a nonreturn suction valve (Figure 15). During normal operation, the valve is open. However, after the discharge cycle has been completed, the suction valve closes automatically. The deep-well pump's driver is then stopped, the pump's above-deck discharge valve is closed, and compressed air or inert gas is injected into the pump through a connection in the discharge head. The gas forces the cargo contained within the deep-well pump out through a bypass line connected to the lower portion of the bowl assembly and into the vessel's piping.

On some vessels, such as those that carry either a limited number of different cargoes or cargoes that are not sensitive to contamination, each deep-well pump may be used to discharge cargo from several of the vessel's tanks. When this arrangement is used, each deep-well pump is generally mounted in a suction tank or can that is connected to multiple cargo tanks via suction piping.

A deep-well cargo pump that is installed in a suction can is frequently fitted with automatic priming valves that enable the pump to remove gas and vapor from the can and from the attached suction piping. The operation of these priming valves and their associated components is often similar to the operation of the recirculation system used to reprime centrifugal cargo pumps. A typical self-priming deep-well pump is fitted with two or three priming valves that are mounted in the pump's column pipe just above the bowl assembly (Figure 16). When the pump is discharging fluid, these valves are closed. However, when the deep-well pump loses suction, the priming valves open and permit the liquid contained within the pump's column and discharge head to drain into the suction can. As this liquid drains through the open priming valves, a vacuum is created in the discharge head and column pipe. The gas in the bottom of the can, which is displaced by the returning liquid, is drawn by the vacuum through a vent line that is connected from the top of the suction can to the pump's discharge head and into the evacuated portion of the discharge head and column. When the liquid level in the suction tank rises sufficiently for the deep-well pump to regain suction, the flow of liquid through the pump resumes, the priming valves and a check valve in the vent line close, and the gases that have been drawn into the column and discharge head are forced out through a check valve installed at the pump's discharge connection. The priming cycles will continue until enough gas has been removed from the cargo suction line for the deep-well pump to operate normally. As the level in the cargo tank being emptied continues to be reduced, the amount of gas entering the suction can will increase, and the deep-well pump will lose suction more frequently. When the amount of liquid remaining in the cargo tank is not sufficient for the deep-well pump to be reprimed, the pump should be stopped.

HYDRAULIC-MOTOR-DRIVEN SUBMERSIBLE CARGO PUMPS Hydraulic-motor-driven submersible pumps are used to discharge cargo on many chemical and multi-petroleum-product carriers. In addition, they are used for cargo discharge on some crude carriers. A

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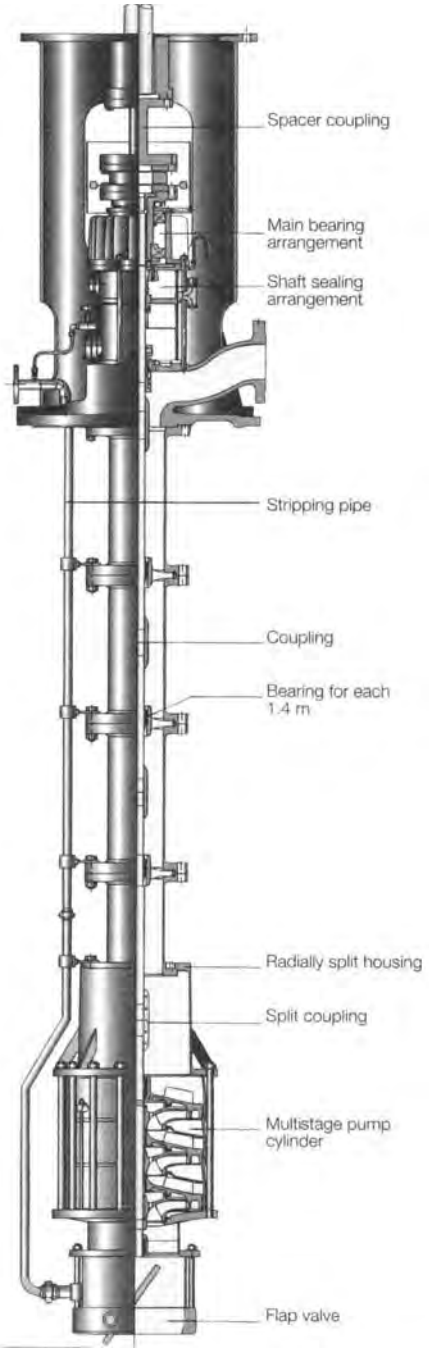


FIGURE 15 Vertical deep-well cargo pump (Svanejoh International A/S)

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600	Discharge Head
604	Adjusting nut
608	Head or top shaft
616	Stuffing box
617	Stuffing-box bushing
618	Gland
620	Packing
624	Bypass pipe
641	Top column pipe
644	Bottom column pipe
646	Line shaft
649	Line-shaft coupling
652	Bearing retainer or bracket
653	Line-shaft bearing
655	Impeller-shaft coupling
660	Pump or impeller shaft
666	Top bowl
670	Intermediate bowl
672	Bowl bearing
673	Impeller
674	First-stage (priming-stage) impeller
677	Impeller taper lock
685	Adapter case
689	Suction bell
690	Suction-bell bearing
702	Autoprime (self-priming) valve housing
704	Autoprime (self-priming) valve spring
705	Autoprime (self-priming) valve
706	Air-release (vent) check valve
707	Air-release (vent) line
708	First-stage (priming-stage) impeller
762	Column-flange bolt
764	Bowl cap screw
779	Stuffing-box gasket

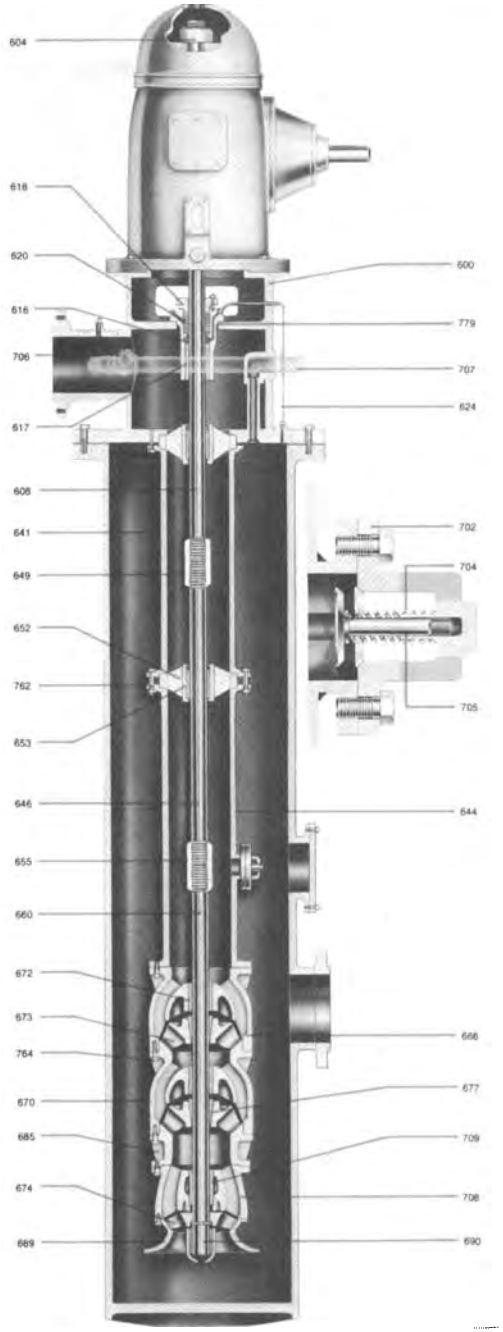


FIGURE 16 Self-priming vertical deep-well cargo pump (ITT/Gould Pumps, Inc.)

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separate pump is usually installed in each cargo tank, which eliminates the need for suction piping. In addition, to prevent different cargoes carried by a vessel from mixing, each pump may be connected to an independent above-deck discharge line. Although the pressurized oil required to drive the hydraulic motors for all of the cargo pumps on a vessel is frequently supplied by a central hydraulic system, a self-contained hydraulic power pack is furnished for each cargo pump on some vessels.

Each submersible unit usually consists of a single-stage end-suction centrifugal pump with a short shaft that is driven, often through either a splined connection or a coupling, by a hydraulic motor (Figure 17). The pump's shaft is typically supported by antifriction ball or roller bearings that are submerged in and lubricated by the hydraulic oil that drains from the motor. The pump is mounted on the lower end of a vertical support pipe that is suspended from a top plate installed on the vessel's main deck. The hydraulic-oil supply and return lines are ordinarily enclosed within this support pipe. Fluid discharged from the pump's volute-type casing typically passes through a second vertical pipe that terminates at an above-deck discharge connection. A control valve that can be used to vary the flow of hydraulic oil to the motor and, therefore, the pump's operating speed valve is usually mounted on the above-deck top plate.

Mechanical or lip-type seals are generally used at the shaft penetrations in a submersible pump-and-hydraulic-motor assembly to prevent hydraulic oil from leaking into a cargo tank and to prevent cargo from mixing with hydraulic oil. Air or inert gas can usually be circulated through a void space or cofferdam that runs through the vertical support pipe that surrounds the hydraulic supply and return lines, the housing that surrounds the hydraulic motor, and a chamber between the pump's seals. After leaving the cofferdam, this gas frequently passes through an above-deck trap in which liquid is separated so a hydraulic-oil or cargo-seal leak can be detected. Compressed air or inert gas is also frequently injected into a submersible pump's above-deck discharge connection after a pumpout has been completed to force cargo contained within the vertical discharge pipe out through a small bypass line connected to the base of the pump. However, because a submersible pump's inlet is generally not fitted with a nonreturn valve, this operation must be performed before the driver is stopped.

It is often possible to disassemble a submersible pump and remove it from a cargo tank for maintenance without disturbing the vertical support pipe or top plate. In most cases, vessels fitted with hydraulic-motor-driven submersible cargo pumps carry one or more portable hydraulically driven submersible pumps that can be lowered into a cargo tank with a winch and used to discharge cargo if the main pump in the tank is inoperable.

In addition to being used for cargo discharge, a submersible cargo pump is sometimes operated while a vessel is underway to circulate cargo through a diffuser so sediment in the liquid does not settle in the cargo tank or through an above-deck heater to prevent the cargo from cooling. Also, to eliminate the need for separate drop lines, on some vessels, liquid is loaded into small cargo tanks by allowing it to flow backwards through a vessel's submersible cargo pumps. When this is done, a nonreverse brake is generally mounted on the submersible pump's shaft to prevent reverse rotation during loading. However, because of the resistance created by the pump's impeller, loading through a submersible cargo pump can increase loading times.

ELECTRIC-MOTOR-DRIVEN SUBMERSIBLE CARGO PUMPS Electric-motor-driven submersible pumps are often used to discharge cargo from liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers. Each submersible unit typically consists of a vertical centrifugal pump with either one or two impellers that are mounted on the lower end of an electric motor's shaft. In addition, an inducer is often installed at the inlet to the first-stage impeller (the only impeller in a single-stage pump) to reduce the pump's net positive suction head requirements.

Pumps used for cargo unloading are frequently installed directly within a vessel's cargo tanks. Cargo typically enters an electric-motor-driven submersible pump through an opening in the bottom of the pump. After being discharged by the pump's impellers, the cargo often enters an annular passage formed between the motor frame and an outer casing that surrounds the motor. A portion of this liquid usually passes through openings in the motor frame and flows through the motor. In addition to cooling the motor, this bypass

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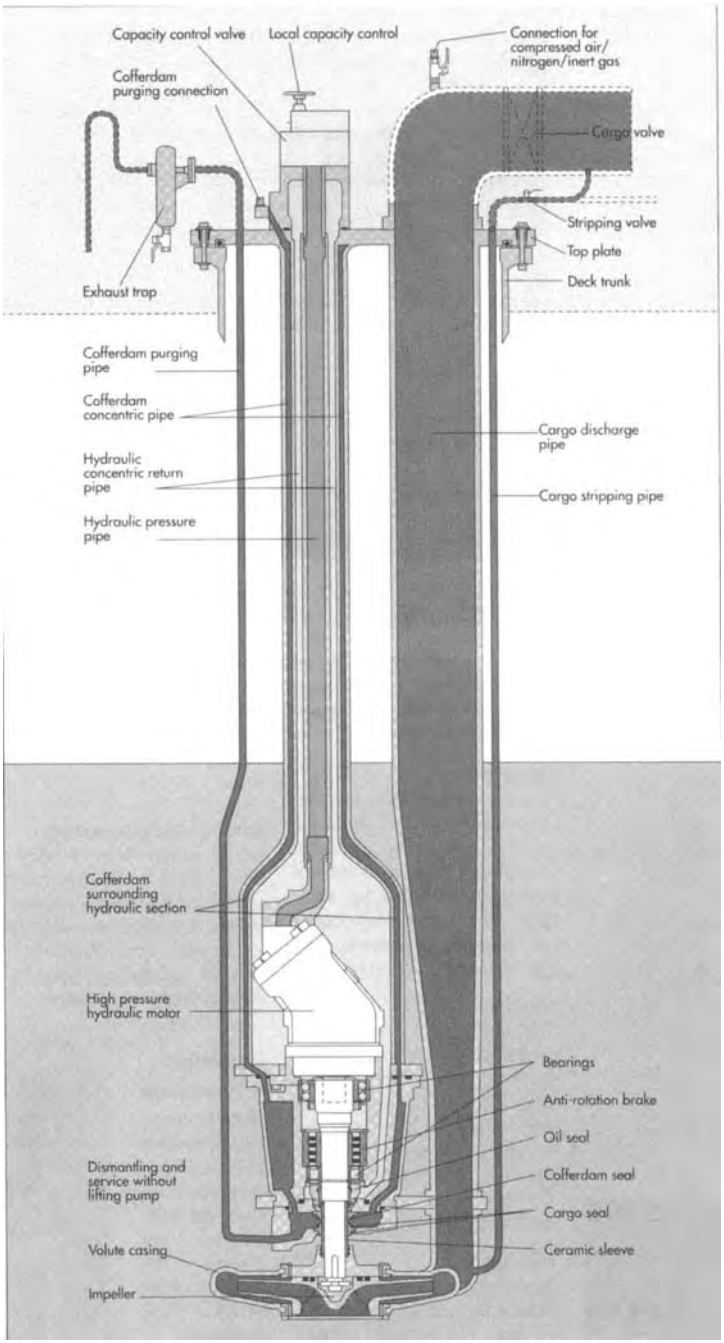


FIGURE 17 Hydraulic-motor-driven submersible cargo pump (Frank Mohn AS)

LIBERTY EXHIBIT 1013, Page 1456

flow lubricates the ball bearings that support the common pump and motor shaft. After leaving the top of the annular passage, the cargo that has been discharged by the pump passes through a connection located above the motor and enters a vertical pipe that leads to the main deck.

In addition to the main cargo unloading pumps, smaller spray or cool-down pumps are often installed on an LNG carrier. When an LNG carrier is unloaded, some cargo is typically left in each tank. During the voyage back to the loading terminal, the spray pumps circulate this LNG through a cool-down header and spray nozzles in each tank. The LNG vaporizes as it passes through the spray nozzles and absorbs heat from the cargo tanks. This enables the tanks to be kept cold until the vessel is reloaded with additional cargo.

ROTARY CARGO PUMPS Some vessels that carry high-viscosity cargoes have multiple-screw- or lobe-type rotary main cargo unloading pumps. In addition, vessels that have centrifugal-type main cargo pumps sometimes also have lower-capacity screw, lobe, or sliding-vane pumps that are used to strip cargo tanks.

A rotary main cargo or stripping pump may be installed in a pump room located in the lower part of a vessel. With this arrangement, the pump can typically take its suction from multiple cargo tanks through interconnected suction piping. So the driver can be isolated from explosive vapor in the pump room, it is frequently installed in an adjacent space and is coupled to the pump with a jackshaft that passes through a bulkhead stuffing box. A typical rotary cargo pump is driven either by a variable-speed driver or by a constant-speed driver through a fluid coupling, which enables the pump speed and, therefore, the capacity delivered to be changed during a pump-out.

Some multiple-screw rotary cargo pumps are furnished in a deep-well configuration (Figure 18), which can eliminate much of the suction piping in a cargo system. Although a vertical driver can be used to drive a deep-well rotary pump, many deep-well rotary pumps are driven through right-angle gears by horizontal motors or engines to facilitate maintenance and reduce vertical height requirements. Typically, the output shaft of the above-deck vertical driver or right-angle gear is connected to the power rotor in the pump through a line shaft that is enclosed within a vertical column pipe. Cargo discharged by a deep-well rotary pump may pass through the column, or it may pass through a separate vertical pipe mounted adjacent to the column. Bearings that support the line shaft, together with the pump's bearings and timing gears, when used, are sometimes lubricated by the pumped fluid. Alternatively, a deep-well rotary pump may be furnished with a pressurized forced-feed lubrication system.

RECIPROCATING CARGO PUMPS Direct-acting reciprocating pumps are used to strip cargo tanks on some vessels. These units typically have double-acting pistons in both the liquid- and the drive-end cylinders. Some reciprocating stripping pumps are mounted in a pump room and are connected to the vessel's cargo tanks through suction piping. With this arrangement, one pump can be used to strip multiple tanks. Pumps installed in this fashion are frequently duplex units (that is, two liquid cylinders) that are mounted vertically and are driven by steam (Figure 19).

When used on multi-product carriers, a separate reciprocating stripping pump may be used for each cargo tank. These stripping pumps are frequently driven by compressed air or inert gas. On some vessels, a horizontal duplex reciprocating pump is mounted in the bottom of each cargo tank. Alternatively, a vertical simplex (i.e., one liquid cylinder) pump with a liquid cylinder that is submerged within a cargo tank and a drive cylinder that is mounted on deck is sometimes used. With this latter configuration, the liquid-end piston is coupled to the piston in the drive cylinder through a long intermediate shaft.

INERT-GAS SYSTEM PUMPS To reduce the risk of explosion and fire, the space in a cargo tank above a flammable liquid must typically be kept filled with an inert gas. Some vessels use flue gas from either a fossil-fueled steam boiler or a dedicated oil-fired inert-gas generator to inert cargo tanks. A pump is usually required to deliver seawater to a scrubber where the water is used to cool, clean, and desulfurize the inert gas. A separate pump is also frequently required to supply seawater to a wet-type deck seal that is used to prevent

LIBERTY EXHIBIT 1013, Page 1457

1 Pumping screws
2 Timing gears
3 Coupling
4 Line shaft or jackshaft
5 Column pipe (standpipe)
6 Discharge head with carrying plate
7 Bearing bracket
8 Column pipe (standpipe)
9 Bearing housing
10 Long pump shaft
11 Short pump shaft
12 Pump body or housing
13 Suction connection / bearing housing

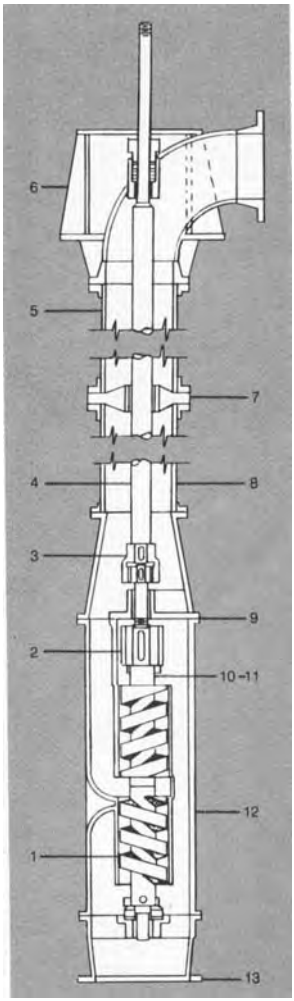


FIGURE 18 Deep-well screw pump (IMO-Warren Pump)

vapor in the vessel's cargo tanks from flowing backwards through the inert-gas supply piping and into the machinery spaces. It is common to use single-stage electric-motor-driven centrifugal pumps for both of these applications.

TANK-CLEANING PUMPS On some vessels, hot seawater is used to clean cargo tanks of residue that remains in the tanks after the liquid cargo has been discharged. A typical tank-cleaning or tank-washing pump takes suction from a sea chest and discharges seawater through a heater in which it is often heated to a temperature of approximately 200°F (93°C). The hot seawater then passes through nozzles in tank washing machines and is sprayed onto the sides of the cargo tanks being cleaned. Single- and two-stage centrifugal pumps that are driven by steam turbines, electric motors, or hydraulic motors are frequently used in this application. A tank-washing pump may also be used for other purposes, such as fire fighting.

LIBERTY EXHIBIT 1013, Page 1458

1 Steam cylinder
2 Steam piston rod
3 Liquid cylinder
4 Liquid-cylinder liner
5 Liquid piston rod
6 Liquid piston
7 Liquid piston packing
8 Liquid valve service
9 Air chamber (pulsation dampener)

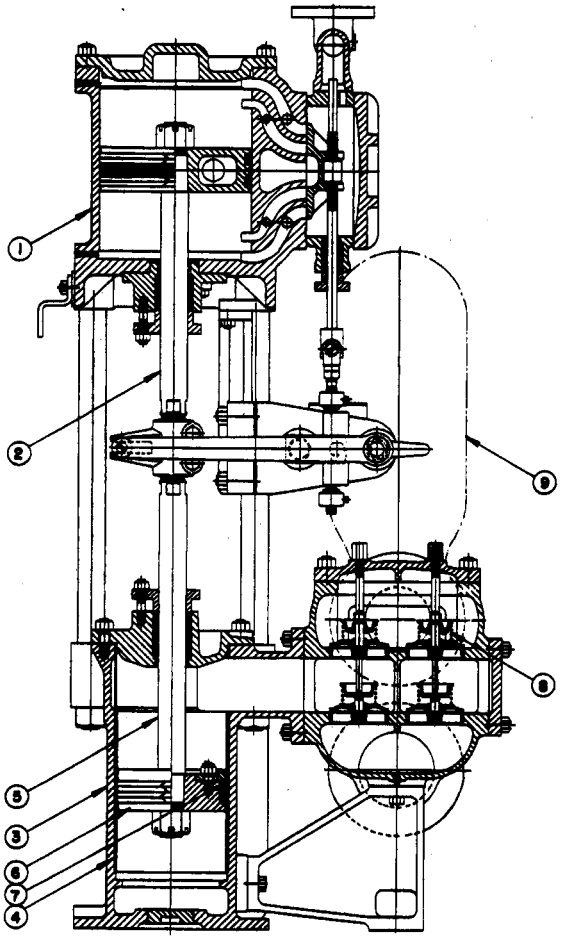


FIGURE 19 Duplex direct-acting reciprocating pump (Flowserve Corporation)

As an alternative to washing cargo tanks with seawater, crude carriers frequently clean tanks with crude oil, referred to as crude-oil washing (COW). A general-service pump is often used on a crude carrier to deliver pressurized crude oil to the vessel's tank-washing machines. The pump may receive this oil from a holding or slop tank on the vessel. The general service pump may also be used to deliver cargo to an eductor that strips cargo tanks during the final stages of unloading or during crude-oil washing. This cargo, which serves as the motive fluid in the eductor, mixes with the fluid being removed from the cargo tank and is then usually discharged by the eductor into one of the vessel's slop tanks. On a typical crude carrier, the general-service pump is similar in configuration to the vessel's main cargo pumps but is smaller.

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SECTION 12.13
HEATING AND AIR CONDITIONING

BURT RISHEL

HEATING AND AIR CONDITIONING

Heating, as defined herein, is the warming while air conditioning is the cooling of people, space, and equipment. These subjects will be discussed with reference to their use of pumps.

Two great developments have had a profound effect upon these fields. They are digital control and variable speed drives. Microprocessor-based controls have replaced pneumatic control systems for improved reliability, response time and system feedback, while new software tools provide the flexibility to address any conceivable control problem. Digital control technology is at work in onboard boiler and chiller control as well as variable speed drives that operate pumps and fans. Standard building automation system protocols and internet capability have further enabled communications between manufacturers.

Digital controls also contribute to energy improvements by providing the necessary platform for adaptive control techniques. Adaptive control algorithms manage operation of individual components like boilers, chillers, pumps, and cooling towers so that the total energy usage is minimized. Adaptive algorithms eliminate the operation of the system based on fixed, preset parameters. Rather, they automatically change set points for addition/subtraction of equipment as conditions change; this minimizes energy consumption.

Figure 1 provides an example of how adaptive algorithms control two variable speed pumps to achieve minimum energy use. Under original system conditions, the second pump would be started at 12 kW and 1140 gpm. As the system conditions change, the algorithm calculates that the second pump should be added at 10 kW and 1080 gpm with adaptive control. Concluding, adaptive control has eliminated the forced operation of an HVAC system to preconceived control parameters.

Variable speed drives have been developed for almost any size pump motor. Integral motors for pumps from 5 to 750 hp are usually for electrical service of 460 V 3 phase 60 Hz while large motors are designed for medium voltages of 2300 or 4160 V. The popularity of these drives has reduced their cost so that they can be applied to any heating or cooling system that has a variable flow requirement. This has eliminated the need for manual adjusting devices such as balance valves and multiple duty valves and special circuiting of piping such as crossover bridges. These hydronic systems require much less energy today with variable speed drives. Conversion of constant speed pumping systems to variable speed has reduced the pumping energy to less than half that required with constant speed pumps. Also, the number of pumps required has been reduced substantially. For

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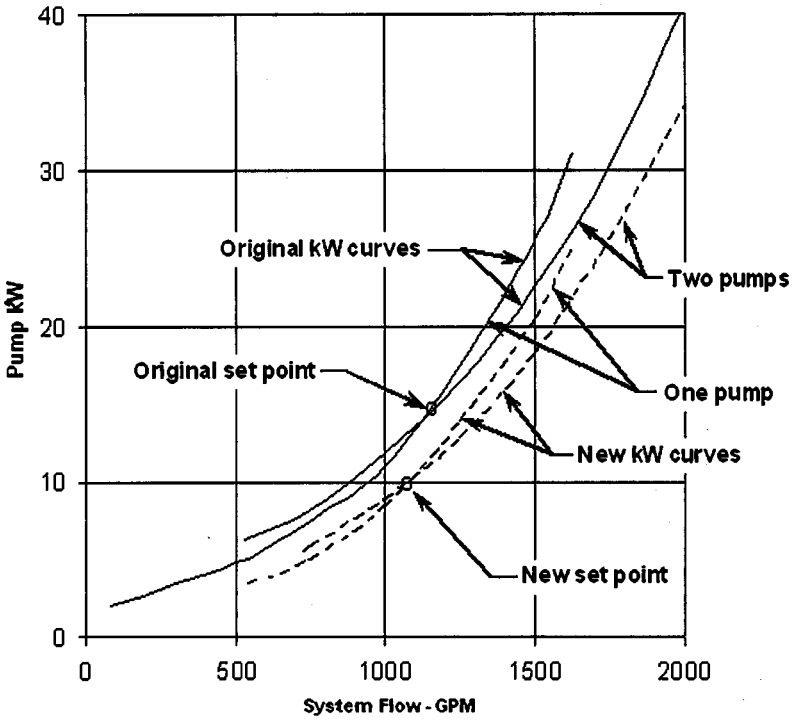


FIGURE 1 Adaptive control of two centrifugal pumps (tekWorx, LLC, Cincinnati, OH)

example, the preferred pumping arrangement was primary/secondary, Figure 2, but it is now replaced by variable primary pumping, Figure 3, on most systems.

Heating The majority of residential and commercial heating is now accomplished with forced air systems that do not use pumps. Larger commercial, public buildings, hospitals, educational facilities, and institutions use hot water heating. Most of these installations are heated with hot water boilers. Smaller hot water systems use in-line circulators as shown in Figure 4 while larger systems may use end suction or axial split case pumps as described in Section 2.2. A recent development is the double suction, in-line pump as shown in Figure 5 with capacities as high as 8000 gpm.

The boilers can be noncondensing or condensing. Condensing boilers are those that operate at lower water temperatures so that the water vapor in the flue gases can be condensed; this results in combustion efficiencies in the range of 85 to 95%. Figure 6 describes a general combustion efficiency curve for condensing boilers based upon return water temperature. Noncondensing boilers are designed of nonferrous materials that do not rust in the presence of water vapor and carbon dioxide (weak carbonic acid). The water temperatures used with these boilers vary from 80 to 150°F (26 to 65°C). Another significant advantage of condensing boilers is the ability to operate at any water flow rate. They do not have a minimum flow rate to eliminate problems such as thermal shock. Therefore, variable speed pumps can operate easily with them. A simple pumping system, known in the industry as variable primary pumping and shown in Figure 3, is used with these boilers.

Noncondensing boilers are those that have been in use for many years; they include cast iron and steel fire tube or water tube boilers. They operate on natural gas with water temperatures above 140°F (60°C) to avoid condensation in the flue collectors. Many of them cannot withstand variable flow, so primary/secondary pumping as shown in Figure 2

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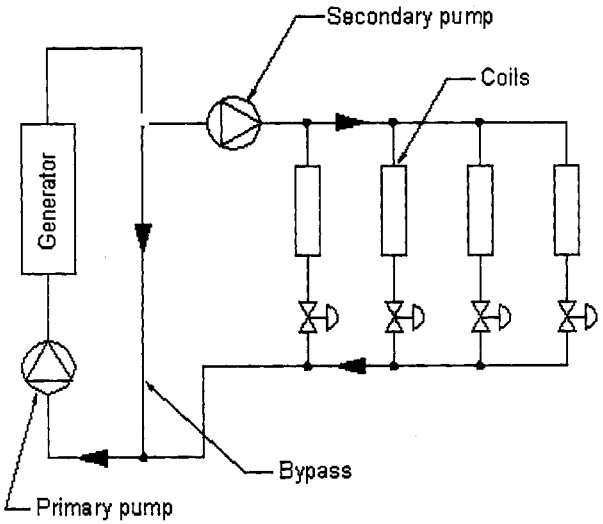


FIGURE 2 Primary/secondary system (Reference 1)

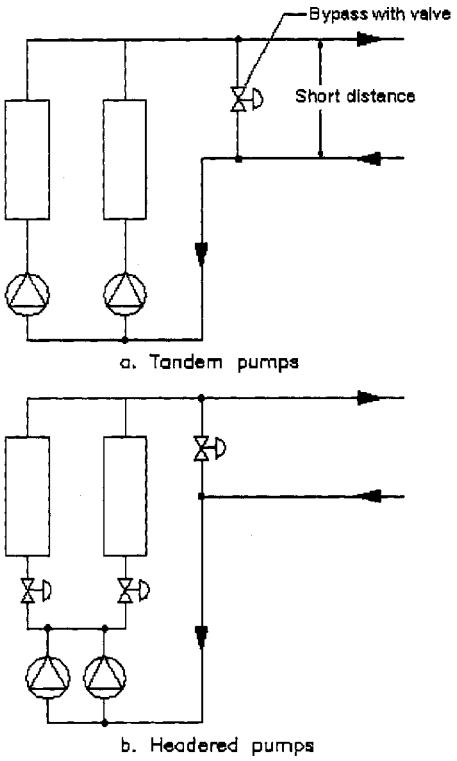


FIGURE 3 Piping arrangements for variable primary pumps (Reference 1)



FIGURE 4 In-line circulator (Flowserve Corporation)

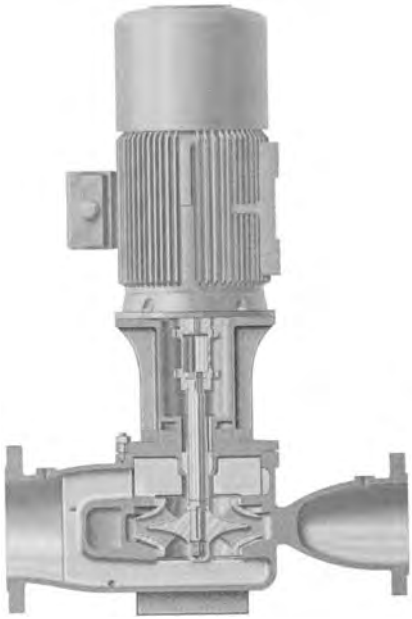


FIGURE 5 Double-suction in-line pump (Flowserve Corporation)

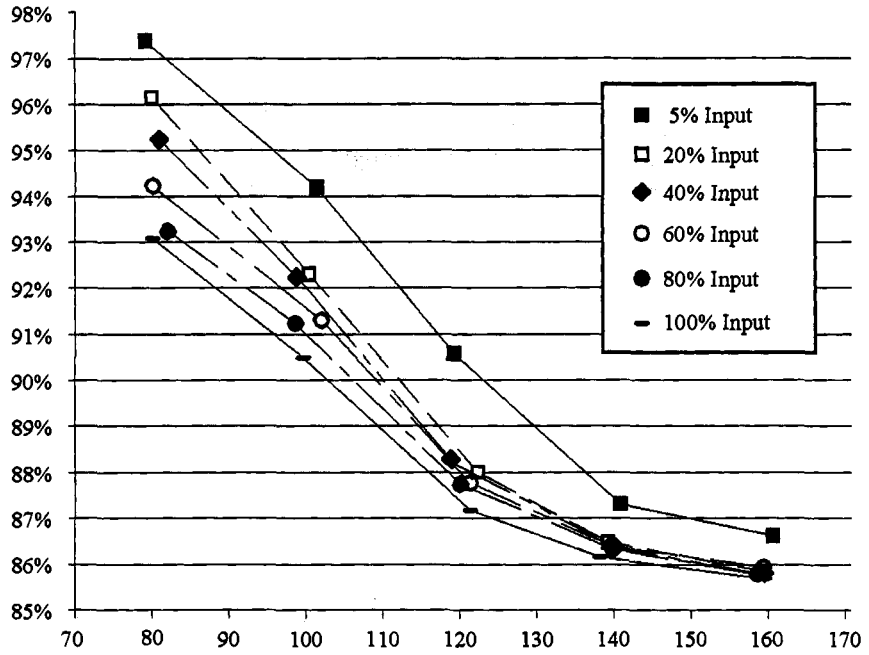


FIGURE 6 Typical thermal efficiency of condensing-type boilers (AERCO International, Inc., Northvale, NJ)

is needed to avoid thermal shock in the boilers. Their combustion efficiency on natural gas is in the range of 78 to 85% at design firing rate. They are being supplanted by condensing boilers due to this lower combustion efficiency.

Air Conditioning Most air conditioning in larger commercial, industrial, and institutional facilities is accomplished by air-cooled or water-cooled chillers. Installations with cooling requirements up to 200 tons utilize air-cooled chillers with closed circuit coolers. Their coefficient of performance (COP) will be in the range 2 to 3. If higher efficiency is required, these installations will have water-cooled chillers with a coefficient of performance of 5 to 6. Continued development of water-cooled chillers has resulted in performance at 0.5 kW/ton or a COP as high as 7. With variable speed drives, these chillers have provided performance at 0.35 kW/ton or COP of 10 at part load.

One of the most important developments in air conditioning is production of water-cooled chillers with the capability of varying the flow in their evaporators (coolers). Digital control of the refrigerant has enabled the chiller designer to allow a flow variation in them from a minimum velocity to design flow. The minimum allowable velocity is subject to the leaving water temperature of the chiller as well as other design criteria. The chiller manufacturer can provide the allowable flow range for a specific machine.

Most water-cooled chillers are electric motor driven even though models are available gas fired or diesel engine driven. The need to acquire higher efficiency has resulted in greater popularity of variable speed units and others with multiple compressors. Figure 7 describes a typical single chiller installation. The chilled water pump is variable speed with a bypass valve to insure minimum allowable evaporator flow. The condenser pump is constant speed excepting on very large installations where the condenser water piping has a sizeable amount of friction.

Multiple chiller installations can be installed with the evaporators in series, parallel, or series/parallel configurations as shown in Figure 8. The size, load range, leaving

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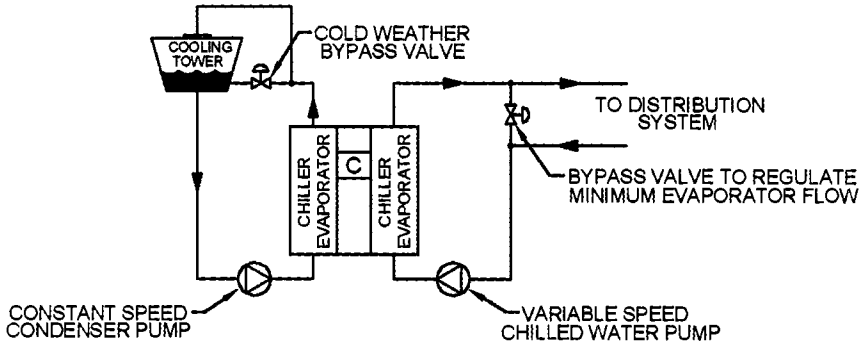


FIGURE 7 Typical installation of a water-cooled chiller

temperature, and temperature differential determine the optimum chiller arrangement. Most installations of two to four chillers are usually installed in parallel. Chilled water pumps were installed in tandem with the chillers in the past. Now, with the capability of great energy savings with variable speed, the pumps are installed in parallel in a header as shown in Figure 8b and 8c.

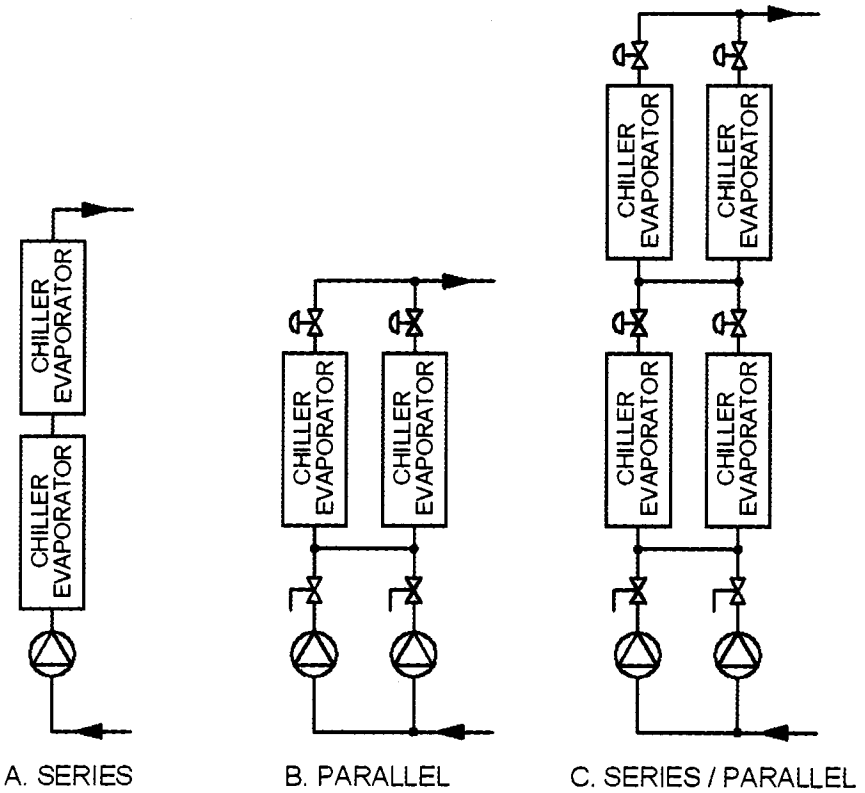


FIGURE 8 Installation of chillers

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A significant effort in energy conservation is the recovery of heat from chillers rather than eliminating it by cooling towers. Heat recovery chillers are now available in several different configurations. The design of low temperature heating systems has enabled the use of this recovered heat for space and domestic water heating.

COOLING TOWERS Open cooling towers provide most of the heat rejection means for water-cooled chillers. Closed circuit coolers are available, and first cost determines the type selected unless there are extenuating circumstances such as plume or noise abatement with open cooling towers. Internal sumps may be desirable on extremely cold installations. The internal sump must have a capacity to hold all of the condenser water when the condenser pumps are shut down. It should be located near the top of tall buildings to reduce the required head for the condenser pumps.

Condenser pumps are of the end suction type on small installations; medium size applications use axial split case pumps, while large installations will have vertical turbine or mixed flow pumps in a sump below the towers. All of these pumps are described in Section 2.2 of this book. NPSH available and required must be checked for all condenser pumps operating with open cooling towers or sumps.

HEAT PUMPS Heat pumps have offered an economical first cost installation for multiple housing and school heating and cooling. A typical installation is shown in Figure 9. Their future popularity is contingent upon the cost of electric power. One way of reducing power costs is through the use of a geothermal field as shown in Figure 10.

Distribution of Chilled Water A principal use of pumps in the HVAC field is the distribution of chilled water. Before variable speed and digital control, many complicated circuits using many pumps were used to control the flow and pressure in chilled water systems. Now, variable speed has reduced these systems to simply one set of pumps as shown in Figure 3 on many installations. Campus-type or multiple building such as college or airport applications may require building pumps to reduce the overall energy consumption. Distributed pumping as shown in Figure 11 accomplishes this without excessive pressure on any of the buildings. Small circulators are required at the chillers to maintain the minimum flow needed for the chillers.

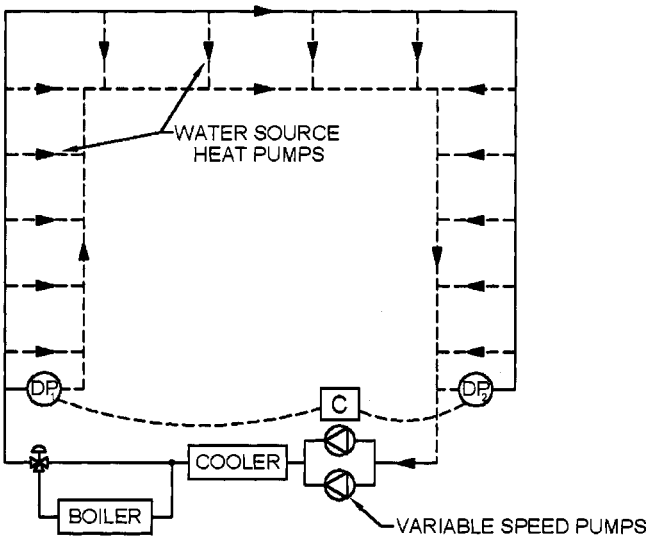


FIGURE 9 Water source heat pumps with closed circuit cooler (Reference 1)

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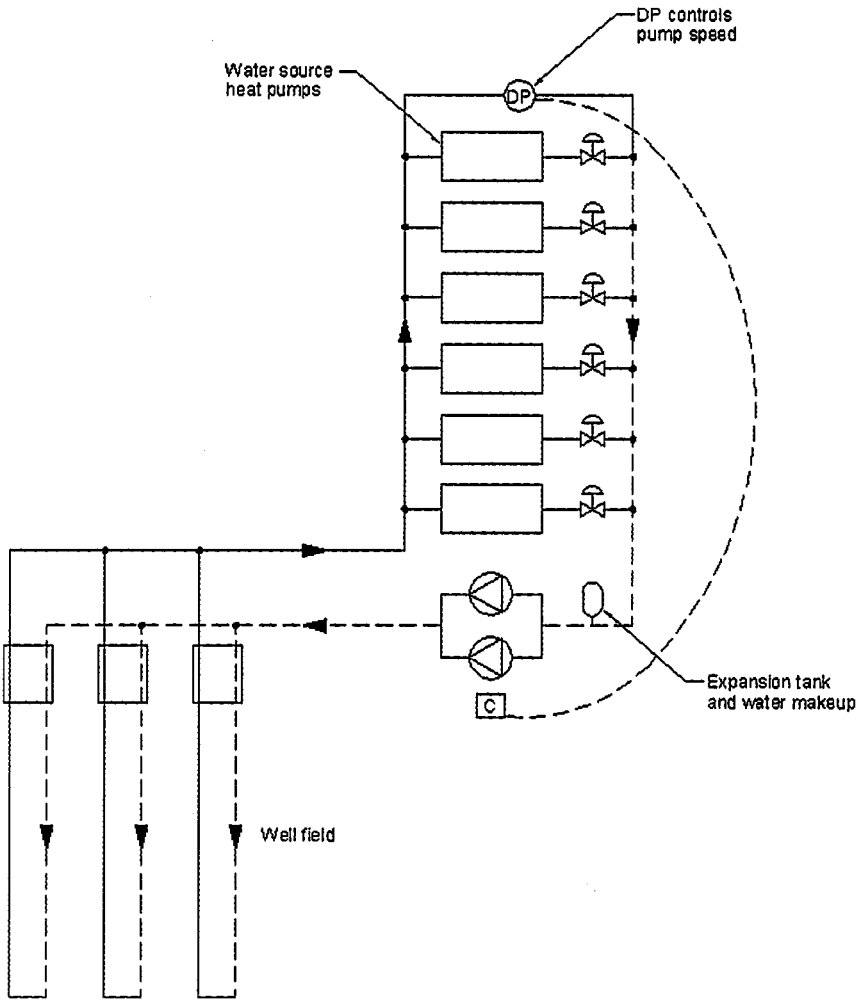


FIGURE 10 Geothermal heat pump installation for low-rise building (Reference 1)

USGBC Green Buildings In an effort to improve the use of energy in buildings, the United States Green Building Council (USGBC) was formed. They have developed the LEED program, Leadership in Energy and Environmental Design. They develop standards and levels of quality to reduce the use of energy in public buildings. They provide Green Building Awards to actual buildings that comply with their standards.

The use of energy in heating and cooling systems is of great concern and, along with USGBC, the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) is developing procedures and practices that will conserve energy in all types of buildings.

Summation The HVAC market has become important to the centrifugal pump industry due to the growth of air conditioning. There are many installations of large axial split case pumps as well as vertical turbine and mixed flow pumps. The energy consumed by

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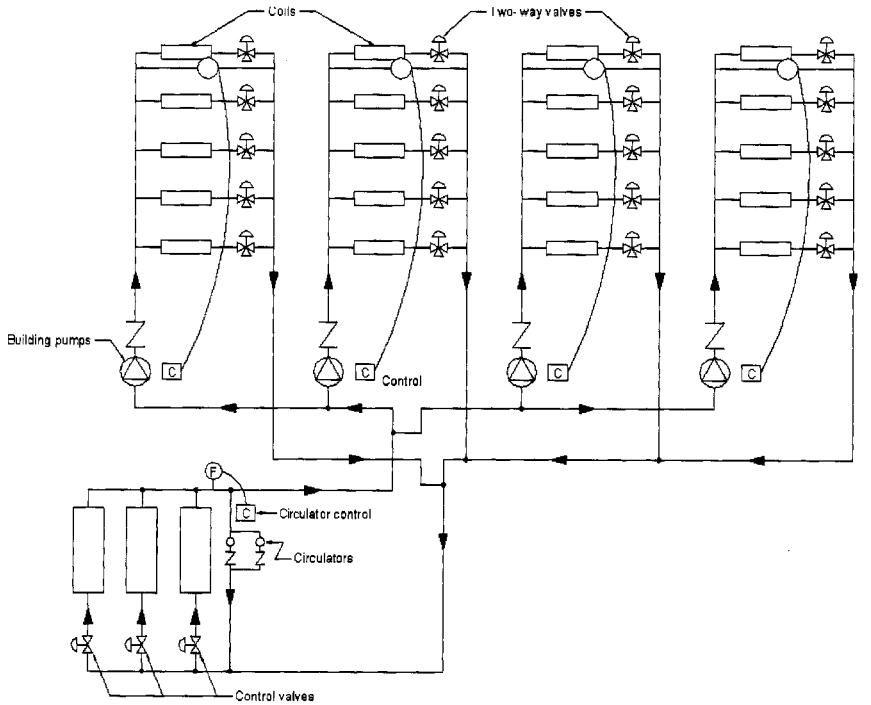


FIGURE 11 Distributed pumping system for four separate buildings (Reference 5)

these pumps along with chillers has resulted in the development of kW/ton for the entire chiller plant, not just the chillers themselves. Figure 12 describes the instrumentation required to acquire a continuous indication of kW/ton for a chiller plant. Due to these efforts, the pumps now used in HVAC applications must be of the same quality and efficiency as those in the general industrial market.

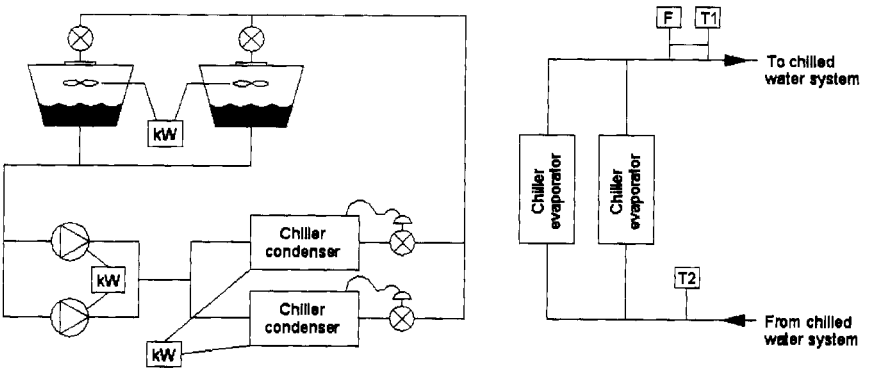


FIGURE 12 Kilowatts per ton instrumentation for central chilled water plant (Reference 1)

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SECTION 12.14
PUMPED STORAGE

C. SAMUEL MARTIN

INTRODUCTION

The variable demand for electrical energy during the day and weekly and the concomitant cost necessitates networks with not only base loads from thermal plants but peaking plants to meet the instantaneous electricity requirement. Ideally, direct storage of electricity would be the most desirable, but currently there are no means of storing large quantities of energy via batteries or other electrical or chemical devices. Presently, peaking loads are satisfied by employing gas turbines or hydroelectric plants, often by means of pumped storage. Pumped storage is a simple concept, requiring two bodies of water at suitable differences of elevation with means of pumping from lower to higher during periods of low electrical demand, followed by generation of power through a hydraulic turbine during periods of high demand.

Compared to steam plants (fossil or nuclear) hydroelectric installations—either conventional or pumped storage—are much faster in meeting electrical demands. An obvious value of pumped storage is its ability to regulate the timing of power generation. Pumped storage accounts for nearly all the effective electrical storage in place around the world, which corresponds to approximately 3% of total electrical generating capacity. For industrialized countries the pumped storage contribution ranges from 2.4% in the United States to approximately 16% for Austria.

The highly reliable and controllable pumped storage technology for fast load change, use as spinning reserve, frequency regulation, and voltage regulation as a synchronous condenser has been recognized and implemented for more than half a century. In Europe, there has been a recent trend of employing reversible pump/turbines rather than separate pumps and turbines—tandem units. In contrast, the two countries with the greatest installed pumped storage capacities—United States and Japan—have utilized only reversible pump/turbines. Figure 1 shows the number of pumped storage plants in selected countries compared with those with pump/turbines.

Pumped storage plants are often designed with utilization of an existing body of water as the lower reservoir in conjunction with a man-made upper pool—termed "pure pumped storage." The term "mixed pumped storage" refers to installations where the power plant may be on a river, or both reservoirs may be natural with water inflow to the upper one. Projects have been planned utilizing abandoned mines or the sea for the lower reservoir. Hence pumped storage need not be restricted to mountainous areas. Many environmental concerns can be addressed by locating all structures underground.

Recently, there has been considerable activity in the planning and construction of pumped storage plants around the world, in particular in Austria, Germany, Japan, and China, as

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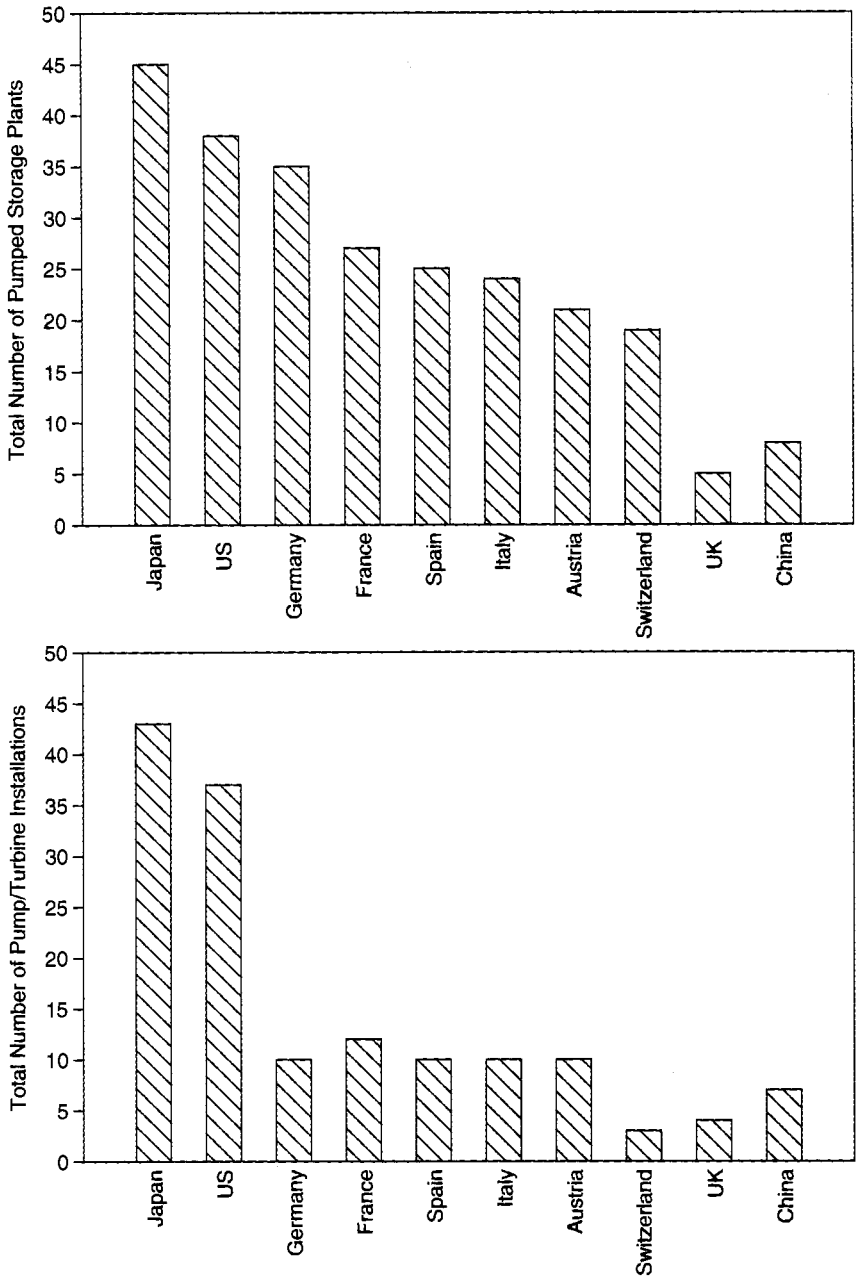


FIGURE 1 Distribution of pumped storage plants and pump/turbines in major countries

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well as in smaller countries, but not in the United States. Several large installations in the planning stage in the United States have been postponed or cancelled because of the effect of deregulation of electrical energy generation. However, increasing fuel prices associated with gas turbines employed for peak loads may have an effect in the future.

PHYSICAL LAYOUT OF TUNNELS AND PENSTOCKS

Basically, the pumped storage installation consists of an upper reservoir with intake, inclined or vertical shaft, upper tunnel, possible upper surge tank, penstock(s), powerhouse, possible lower surge tank, tailrace tunnel, and lower reservoir. Three typical conduit configurations are illustrated as schematics in Figure 2. Typography and local geological conditions will play an important role in dictating the configuration. In order to compare the differences from plant to plant Table 1 is prepared to summarize the configuration of eleven pumped storage installations in the United States. For Figure 2a (Bath County and Castiac) the upper tunnel is on a slight slope leading up to the inclined shaft. In both examples the powerhouse is connected directly to the lower reservoir through the draft tube. The existence of the upper surge tank is required to prevent negative pressures in the piping at the knee. Local conditions permitting, a vertical shaft from the upper reservoir to the upper tunnel, as shown in Figure 2b, would eliminate the upper surge tank. A lower surge tank was in fact installed at Raccoon Mountain, but not for Bad Creek. Both plants have an underground powerhouse. Both upper and lower surge tanks were necessary for Plant Helms. The remaining seven plants listed in Table 1 correspond to the schematic layout illustrated in Figure 2c.

CONFIGURATION OF POWERHOUSE MACHINERY

In alpine countries of Europe pumped storage installations before the 1960s were traditionally configured with separate pumps and turbines—called tandem units. Some very old installations utilized the same water conveyances, but perhaps separate powerhouses, necessitating four machines—pump, motor, turbine, and generator. This configuration is referred to as **quaternary** (base 4). The advantage is the speed of changing from generating to pumping and back to generation. Many installations were (and are) configured to be **ternary** (base 3), with pump, motor/generator, and turbine—also referred to as tandem. Figure 3 is a photograph of a double-suction two-stage storage pump in a tandem arrangement. The horizontal axis arrangement for Plant Säckingen with tandem units is shown in Figure 4. Both pump and turbine (Francis or Pelton) rotate in the same direction, and are either directly coupled or linked by torque converter and/or clutch. Fast switching from pump mode to turbine mode is a clear advantage. Disadvantages are (1) requirement of extra hydraulic machine and mechanical equipment, (2) large powerhouse, and (3) maintenance costs. Figure 5 is a vertical axis arrangement of a **ternary** configuration comprised of impulse turbine, motor/generator, torque converter, and storage pump. Table 2 lists the various machine arrangements, coupling, and orientation of axes.

As shown by Figure 1, however, most of the pumped storage installations in the world are comprised of reversible pump/turbines, defined as **binary** (base 2). The two machines are pump/turbine and motor/generator, directly coupled, but normally required to rotate in opposite directions. The exception is a turbomachine that rotates in the same direction in both pump and turbine modes, named Isogyre. Although there are a few installations, this design is restricted to lower capacity applications. The main advantage of the **binary** unit compared to **ternary** is the fact only one hydraulic machine is required. However, there is a sacrifice in efficiency when operating in the turbine mode, but this is becoming less with modern designs influenced by computational fluid dynamics (CFD). Disadvantages to **binary** are (1) switching modes of operation and (2) starting the pump/turbine in the pumping mode. A typical pump/turbine impeller is illustrated by the photograph in Figure 6.

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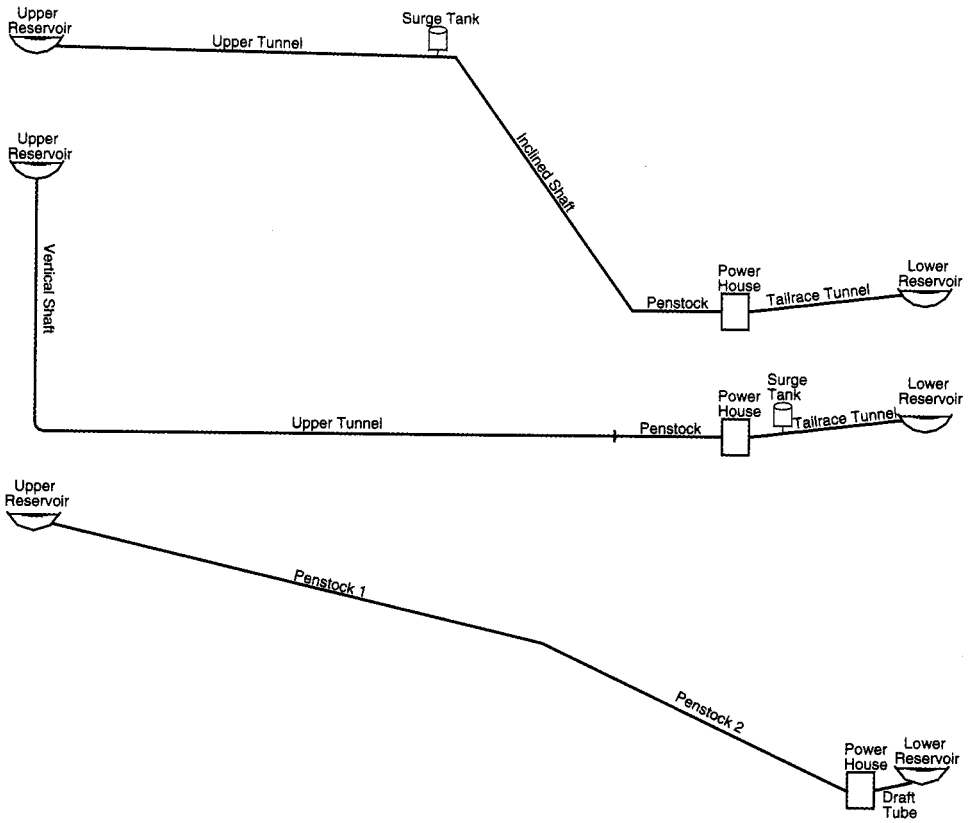


FIGURE 2 Conduit layouts: a (top), b (middle), c (bottom)

TABLE 1 Lengths of tunnels and penstocks of selected U. S. plants

Plant	Units	Tunnel or penstock length (m) (Angle of tunnel or penstock)								Upper length	Lower length
Bear Swamp	2	228 (–90°)	125 (~0°)	53 (~0°)		152 (~0°)	–	–	–	406	152
Ludington	6	158 (–3°)	246 (–18°)		–	–	–	–	–	396	0
Fairfield	8	115	220		–	–	–	–	–	335	0
Blenheim-Gilboa	4	316 (–90°)	274 (~0°)	597 (~0°)		–	–	–	–	1,186	0
Northfield Mountain	4	259 (–90°)	104 (–0°)	1,563 (–0°)		–	–	–	–	1,926	0
Rocky Mountain	3	173 (–90°)	590 (~0°)	147 (~0°)		–	–	–	–	910	0
Bad Creek	4	278 (–90°)	1,128 (~0°)	121 (~0°)		136 (~0°)	290 (~0°)	–	–	1,527	426
Castiac	6	11,514		1067		–	–	–	–	12,581	0
Bath County	6	1,103 (–2°)		302	1,054	320		–	–	2,779	0
Raccoon Mountain	4	274 (–90°)	297		48		480	–	–	571	528
Helms	3	1,293	2,806		686		82		1,180	4,785	1,262

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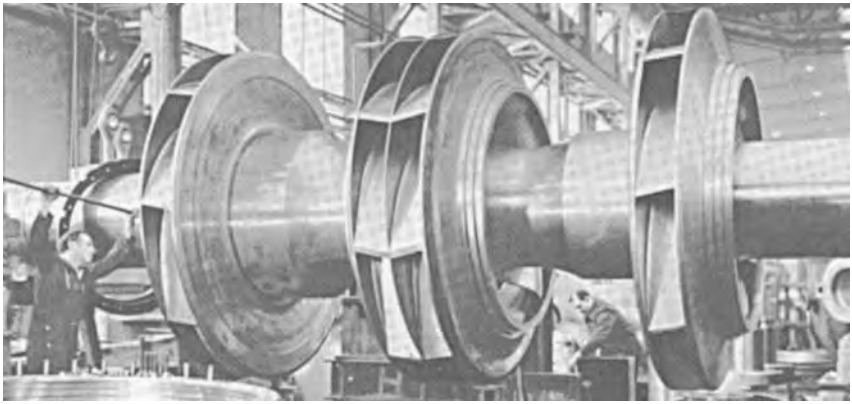


FIGURE 3 View of double-suction two-stage storage pump impeller

PUMP/TURBINE INSTALLATIONS

Selected pump/turbine installations around the world will be listed to emphasize the various aspects of the machines and the various trends. Of the some 38 U.S. plants Table 3 is included to list the salient pump/turbine characteristics of the same installations in Table 1. The last installation finished was Rocky Mountain (1995). Since then, however, numerous plants have been constructed around the world. Figure 7 is the arrangement for a **binary** arrangement of a five-stage pump/turbine for Plant Edolo in Italy. Selected plants with total power capacity of 1200 MW or more are listed in Table 4. Sites with multistage pump/turbines and storage pumps are summarized in Table 5.

PUMP/TURBINE DESIGN

The pump impeller is typically designed for pump mode, resulting in acceptable performance in turbine mode. With considerable enhancements in CFD in recent years, however, improvements are also possible for turbine mode operation. Multistage machines have

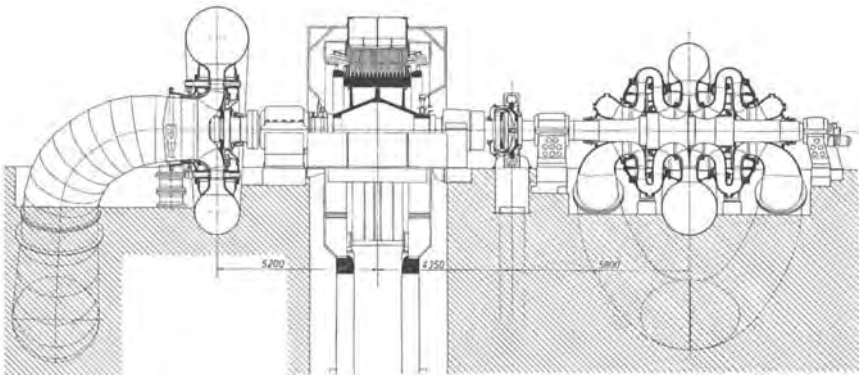


FIGURE 4 Tandem arrangement with Francis turbine and two-stage double-suction storage pump (Plant Säckingen)

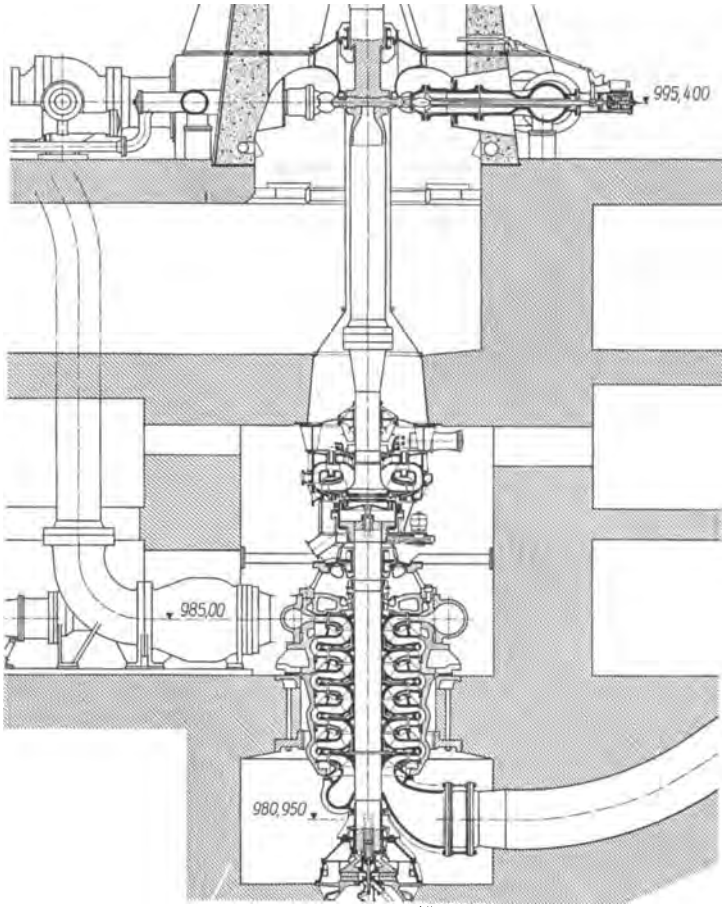


FIGURE 5 Tandem arrangement with Pelton turbine and five-stage storage pump (Plant Lünersee)

TABLE 2 Definition of machine sets

Machine set	Machines	Coupling	Rotation direction	Pump casing	Orientation	Example
ternary	SP-MG-FT	Rigid	Same	Dewatered	VA	Waldeck II
ternary	SP-MG-IT	TC	Same	Filled	VA	Lünersee
ternary	SP-MG-IT	TC	Same	Filled	HA	Säckingen
binary	PT-MG	Rigid	Opposite	Dewatered	VA	Ludington
binary	PT-MG-MS	Rigid	Opposite	Filled	VA	Edolo
binary	Isogyre PT-MG	Rigid	Same	Filled	VA	Malta

SP: Storage Pump; PT: Pump/Turbine; FT: Francis Turbine; IT: Impulse Turbine; MS: Multistage; G: Generator; M: Motor; MG: Motor/Generator; TC: Torque Converter; C: Clutch; VA: Vertical Axis; HA: Horizontal Axis

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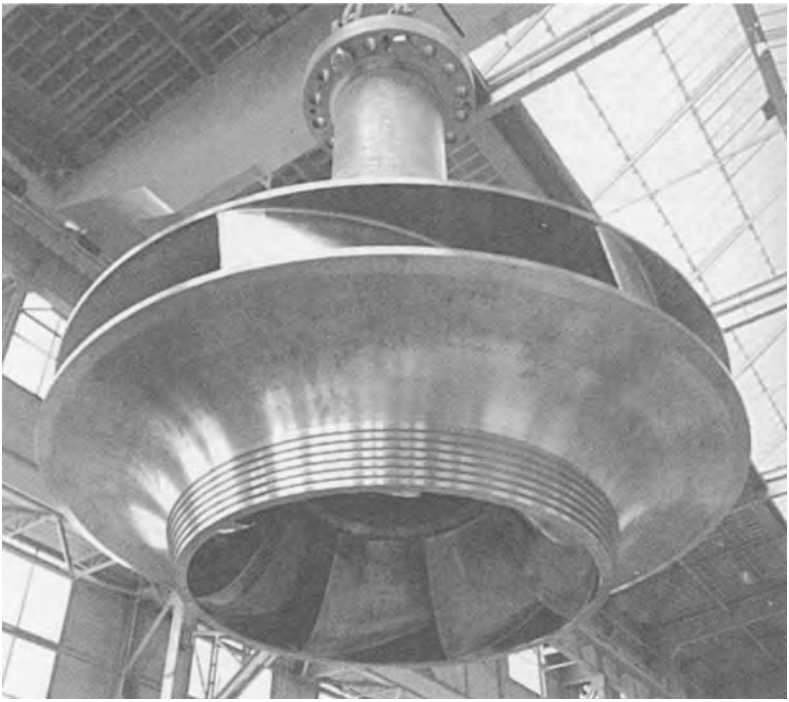


FIGURE 6 Pump/turbine impeller for Plant Palmiet

TABLE 3 Pump/turbine characteristics of selected U. S. plants

Plant	Year	Units	N (rpm)	Unit MW (Pump)	Plant MW	Static head (m) (Pump)	Flow (m^3/s) (Pump)	Specific speed N_{Qp}
Bear Swamp	1974	2	225	300	600	209	125	45.8
Ludington	1973	6	112.5	308	1,980	87	315	70.1
Fairfield	1978	8	150	62	512	49	136	94.4
Blenheim–Gilboa	1973	4	257.1	270	1,000	358	65	25.2
Northfield Mountain	1972	4	257.1	260	1,080	226	102	44.5
Rocky Muntain	1995	3	225	250	760	187	146	53.8
Bad Creek	1991	4	300	270	1,000	380	51	24.8
Castiac	1973	6	257.1	255	1,250	325	64	26.8
Bath County	1985	6	257.1	380	2,100	335	119	35.8
Raccoon Mountain	1978	4	300	400	1,530	305	109	42.9
Helms	1984	3	360	360	1,205	457	71	30.7

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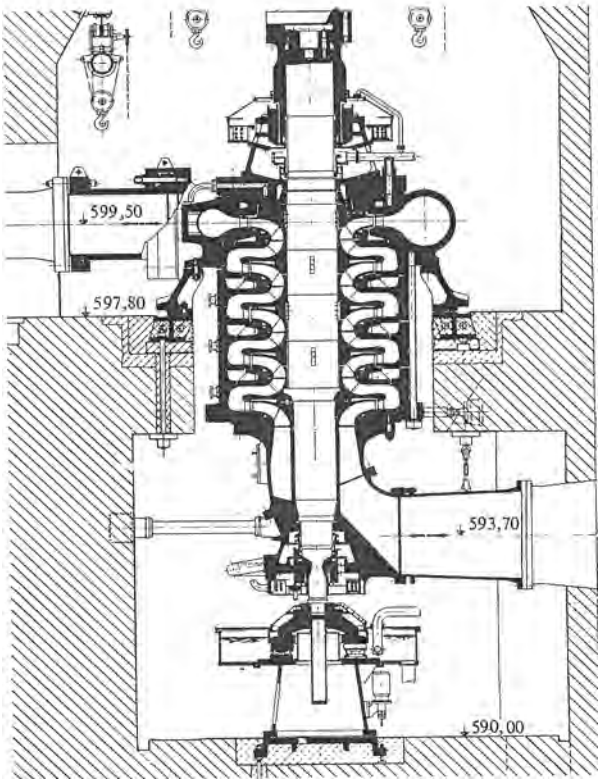


FIGURE 7 Binary arrangement with five-stage pump/turbine (Plant Edolo)

TABLE 4 Selected plants with total pump power exceeding 1,200 MW

Plant	Country	Year	Units	Plant power (MW)	Head (m)	Speed (rpm)
Blenheim-Gilboa	USA	1973	4	1,200	358	257
Ludington	USA	1973	6	1,980	108	112.5
Okutataragi	Japan	1974	4	1,930	380	300
Raccoon Mountain	USA	1978	6	1,530	305	300
Castiac	USA	1978	6	1,250	325	257
Tagasegawa	Japan	1978	4	1,280	229	214
Okuyoshino	Japan	1978	6	1,200	530	514
Dinorwig	UK	1982	6	1,800	545	500
Helms	USA	1984	3	1,200	457	360
Bath County	USA	1985	6	2,100	335	257
Tamahara	Japan	1986	4	1,200	518	428
Ohkawachi	Japan	1993	4	1,280	395	333 to 390
Mingtan	Taiwan	1994	6	1,620	403	400
Tianhuangping	China	1998	6	1,800	608	500
Kazunogawa	Japan	1999	4	1,600	779	500
Guangzhou	China	2000	4	1,200	550	500
Kannagawa	Japan	2005	6	2,820	653	500

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TABLE 5 Selected plants with multistage pumps or pump/turbines

Plant	Country	Machine type	Stages	Unit pump power (MW)	Head (m)	Speed (rpm)
Chiotas	Italy	PT	4	152	1,070	600
Roncovalgrande	Italy	SP-IT	4	94	750	500
Grand Maison	France	PT	4	151	900	600
Edolo	Italy	PT	5	130	1,287	600
La Coche	France	PT	5	80	940	600
Ste Helene	France	PT	5	81	910	600
Lunersee	Austria	SP-IT	5	43	950	750
San Fiorono	Italy	SP-IT	6	110	1,440	500

SP: Storage Pump; PT: Pump/Turbine; FT: Francis Turbine; IT: Impulse Turbine

already exceeded the 1400 m limit. The improvement in hydraulic design through CFD over the years has resulted in an increase in maximum head for single-stage pump/turbines, attaining values approaching 800 m in the pumping mode. This has also led to upgrading of pump/turbine installations around the world. For pump/turbines the specific speed can be defined for either the pump or turbine mode. Since the machine is designed as a pump the definition is based on pump performance to be

$$N_{Qp} = \frac{N\sqrt{Q}}{H^{3/4}} \tag{1}$$

where metric units (rpm · m^3, s, m) are used. The more general dimensionless representation is

$$\Omega_s = \frac{\omega\sqrt{Q}}{[gH^{3/4}]} \tag{2}$$

Correlation of pump/turbine head versus specific speed for a few selected plants is presented in Figure 8. For each plant there are the two limits to head, and corresponding values of specific speed in metric terms. For Francis turbines past correlations have been shown to be approximately represented by

$$H = \frac{\lambda^2}{N_{Qp}^2} \tag{3}$$

where λ is a constant, apparently not formally defined. This correlation is plotted in Figure 8 for λ = 600 and 800, yielding slopes of –2.0 on log-log plots. Although not precisely calculated the data roughly correspond to a slope of –0.5.

ADJUSTABLE SPEED PUMP/TURBINES

Another enhancement of pump/turbine performance is the implementation of adjustable speed motor/generators, yielding a wider range of higher efficiencies in the pumping mode. Table 6 lists the chronology of variable-speed units, starting with Japan, and recently being employed in Germany (Goldisthal 2002), to be followed in Austria with Limberg II (2012). Variable-speed units can widen the cavitation characteristics in the pumping mode, as well as provide easier start-up in the pumping mode. Moreover, the broader range of performance provides more operating flexibility.

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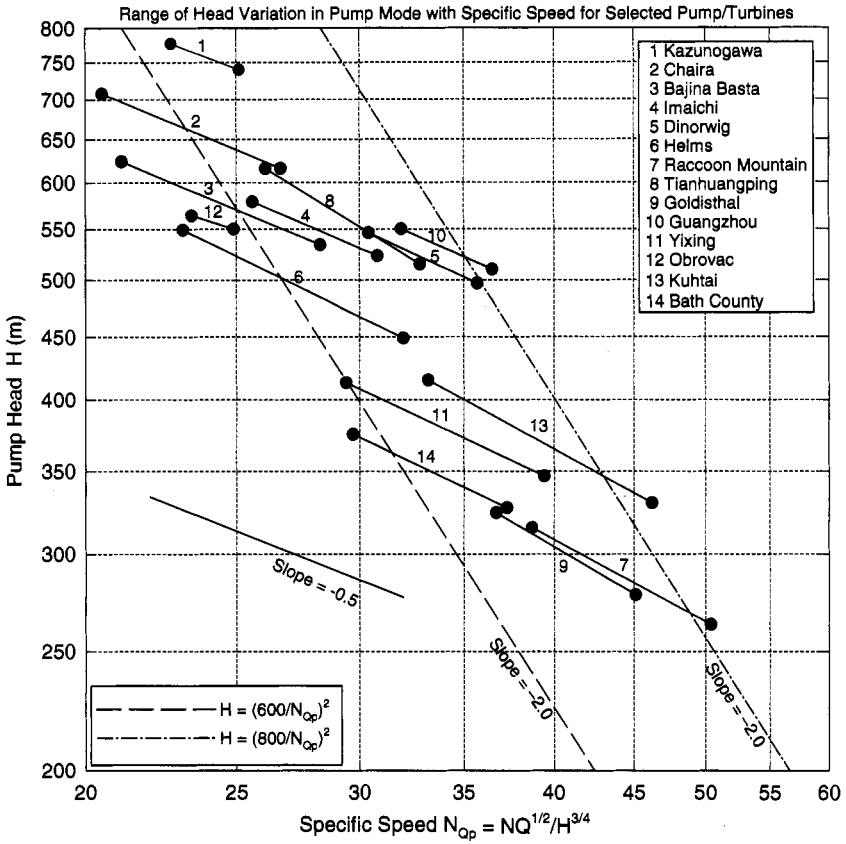


FIGURE 8 Pump head versus specific speed for selected pump/turbines

TABLE 6 Selected plants with adjustable speed motor/generators

Plant	Country	Year	Pump power (MW)	Pump head (m)	Speed range (rpm)
Yagisawa	Japan	1990	53 to 82	63 to 113	130 to 156
Takami	Japan	1993	44 to 140	86 to 118	200 to 254
Ohkawachi 3 & 4	Japan	1993	240 to 400	432	330 to 390
Shiobara 3	Japan	1995	200 to 330	330 to 387	356 to 394
Okukiyotsu 2	Japan	1996	230 to 340	445 to 494	407 to 450
Kazunogawa	Japan	2001	482	650 to 708	500+/–
Okumino 5 & 6	Japan	2000	270	520	514+/–
Goldistahl	Germany	2004	168 to 289	279 to 334	300 to 347
Limberg II	Austria	2012	240	360	–

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Finally, for heads exceeding 800 m the designer has to choose between (1) multistage pump/turbines or (2) tandem units comprised of multistage (storage) pumps and either impulse or Francis turbines. Examples of unregulated machines are summarized in Table 5.

STARTING PUMPS

The starting of storage pumps or pump/turbines in the pumping mode to an eventual synchronous speed can be accomplished in various ways. For **ternary** units with storage pump and Francis turbine rigidly connected on same shaft, rigid pump starting can be accomplished with dewatered pump casing (Waldeck II: Table 2). For **binary** sets pump/turbines are most often started with the pump casing dewatered. Plants Chiotas and Edolo are exceptions in that either a dedicated Francis turbine is used or back-to-back employing adjacent impulse turbine with the pump casing filled. Various techniques for pump starting are:

- Across the line
- Asynchronous starting
- Back to back using adjoining turbine unit or dedicated Francis or Pelton turbine
- Back to back for multistage unregulated pump/turbine with spherical valves with bypass valve for fine control
- Reduced voltage starter
- Use of pony motor with dewatered pump casing

FOUR QUADRANT CHARACTERISTICS

For transient simulation hydraulic characteristics are required in both pump and turbine regions of operation, as well as in transitory regions. Reverse flow and reverse rotation are also possible for storage pumps. For pump/turbines, however, steady-state operation in pump and turbine mode are definitely required. Typically, model studies are conducted for pump/turbines over a wide range of machine operation for wicket gate (guide vane) openings. The model data are generally represented in terms of nondimensional parameters in so-called four quadrants, with the axes being speed and flow. The same data are sometimes correlated in terms of head coefficient and flow coefficient.

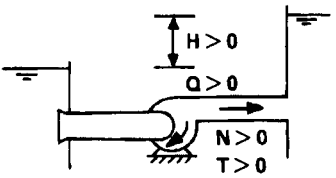
Figure 9 depicts eight zones of possible operation of a pump or pump/turbine in four quadrants. As listed in Table 7 Quadrant I contains normal pumping via Zone A, while normal turbining is Zone G in Quadrant III. In a model test stand the machine can be made to operate in all four quadrants, even with those zones with negative head. For a prototype unit operating as a pump, loss of power will mean the pump starts in Zone A (Quadrant I), enters Zone H (Quadrant IV) as flow reverses, then Zone G (Quadrant III) as the machine reverses rotation, during which time the wicket gates are normally closing. If the transient persists until the pump passes zero torque (runaway) before the gates are closed then the unit enters so-called braking, called Zone F. If the servomotor were to fail, or if the pump/turbine was initially generating (Zone G in Quadrant III) there is a possibility of the machine entering Zone E (Quadrant II), called reverse pumping.

As shown in Table 7 Zones B, C, and D correspond to negative head, a condition only possible to be contrived in the laboratory model. Consequently, model data are typically only reported for positive heads, represented by Zones A, H, G, F, and E. In order to compute speed change during the transient not only head-flow data are needed, but also torque.

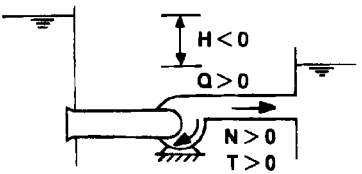
HYDRAULIC TRANSIENT ANALYSIS

The potential for waterhammer exists in pumped storage installations mainly due to disconnections from the electrical grid. For preliminary analyses the three worst scenarios

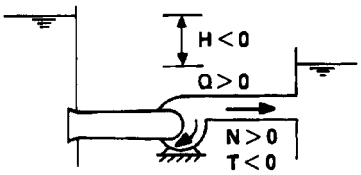
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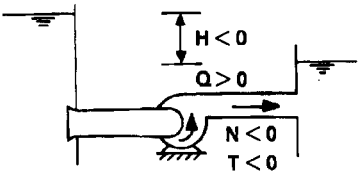
Zone A. Normal Pumping (I)



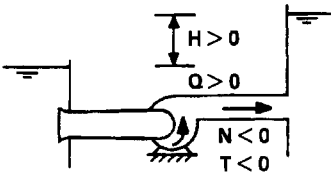
Zone B. Energy Dissipation (I)



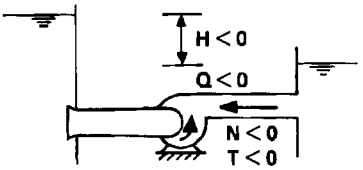
Zone C. Reverse Turbine (I)



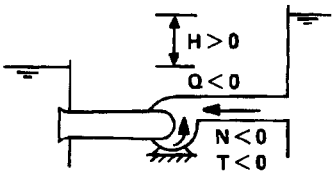
Zone D. Energy Dissipation (II)



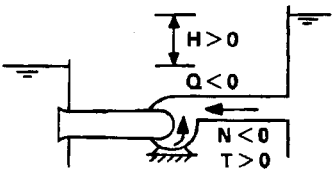
Zone E. Reverse Rotation Pumping
Radial–Flow Machine (II)



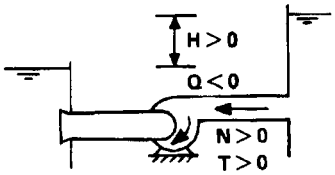
Zone E. Reverse Rotation Pumping
Mixed-or Axial-Flow Machine (III)



Zone F. Energy Dissipation (III)



Zone G. Normal Turbine (III)



Zone H. Energy Dissipation (IV)

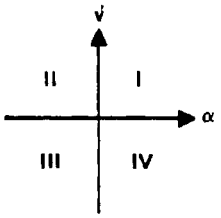


FIGURE 9 Definition of zones and signs of variables for four quadrants

are (a) full load rejection of all operating units, (b) pump power failure, and (c) load acceptance. The possibility of negative pressures in the upper tunnel and shaft must be investigated for pump power failure and load acceptance. For the layout in Figures 2a and 2c an upper surge tank may be necessary. For long tailrace tunnels load rejection and/or pump start-up from no load, or other transient scenarios, may require a lower surge tank.

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TABLE 7 Definition of zones and quadrants

Quadrant	Zone	Mode	Energy	Rotation	Flow	Head	Torque
I	A	Normal pumping	Productive	$N > 0$	$Q > 0$	$H > 0$	$T > 0$
I	B	Energy dissipation	Dissipative	$N > 0$	$Q > 0$	$H < 0$	$T > 0$
I	C	Reverse turbining	Productive	$N > 0$	$Q > 0$	$H < 0$	$T < 0$
II	D	Energy dissipation	Dissipative	$N < 0$	$Q > 0$	$H > 0$	$T < 0$
II	E	Reverse pumping	Productive	$N < 0$	$Q > 0$	$H > 0$	$T < 0$
III	F	Braking	Dissipative	$N < 0$	$Q < 0$	$H > 0$	$T < 0$
III	G	Normal turbining	Productive	$N < 0$	$Q < 0$	$H > 0$	$T > 0$
IV	H	Energy dissipation	Dissipative	$N > 0$	$Q < 0$	$H > 0$	$T > 0$

Raccoon Mountain is an example. Plants Helms (USA) and Kannagawa (Japan) needed both upper and lower surge tanks. (See Section 11.3).

For final analyses more transient cases may be warranted, as listed below

- Full load rejection with guide vane closure
- Full load rejection with main inlet valve (MIV) closure and locked guide vanes
- Full load acceptance from speed-no-load
- Pump power failure with guide vane closure
- Pump power failure with main inlet valve closure
- Pump start-up from synchronization
- Normal turbine shutdown
- Normal pump shutdown
- Reduction of generating load
- Increase in generating load
- Load rejection followed by load acceptance
- Load acceptance followed by load rejection
- Load acceptance followed by normal shutdown
- Malfunction of governor cushioning stroke, resulting in higher closure rate over entire stroke
- Guide vane closure from speed-no-load
- Governor stability
- Resonance phenomena

Load Rejection and Pump Power Failure Example Bad Creek Plant in the United States is chosen as an example for hydraulic transient simulation. The piping layout and pertinent pump/turbine data are listed in Tables 1 and 3, respectively. The vertical shaft is 278 m, followed by a nearly horizontal tunnel of 1128 m, leading to four penstocks of 121 m each. From hydraulic model test data for eleven wicket gate openings ranging from 3 to 40 mm four-quadrant curves are plotted in Figures 10 and 11, for unit flow versus unit speed, and unit torque against unit speed, respectively. Nondimensional definitions are employed for all three quantities, as follows. For a pump impeller with diameter D rotating at ω the peripheral speed is U, yielding a speed coefficient K_{u1}

$$K_{u1} = \frac{U}{\sqrt{2gH}} = \frac{\omega D}{2\sqrt{2gH}} = \frac{1}{\sqrt{\Psi}} \tag{4}$$

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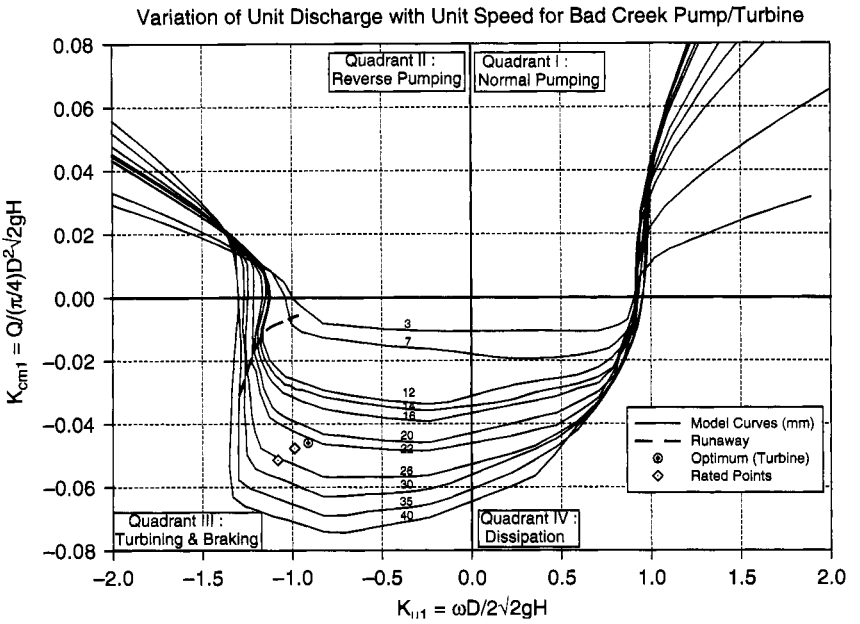


FIGURE 10 Correlation of unit discharge with unit speed for Bad Creek pump/turbine

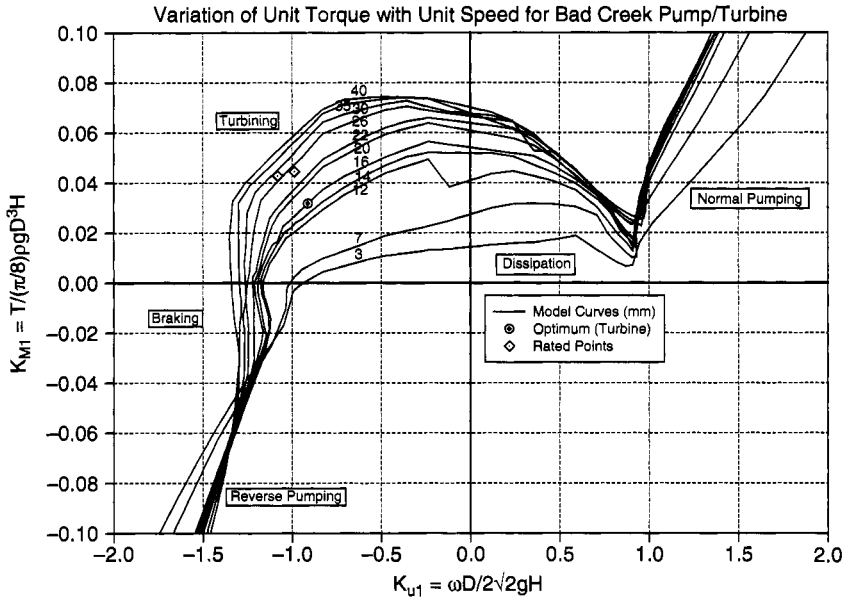


FIGURE 11 Correlation of unit discharge with unit torque for Bad Creek pump/turbine

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Dividing the flow Q by an area based on diameter D results in a unit discharge parameter K_{cm1}

$$K_{cm1} = \frac{V}{\sqrt{2gH}} = \frac{4Q}{\pi D^2\sqrt{2gH}} \tag{5}$$

The unit torque is defined as

$$K_{M1} = \frac{8T}{\pi\rho g D^3 H} \tag{6}$$

Figure 10 shows the four quadrants and Zone A (Quadrant I), Zone H (Quadrant IV), Zones G and F (Quadrant III), and Zone E (Quadrant II). The dashed runaway line in Quadrant III is the line of zero torque, separating Zones G and F. Reverse pumping is Zone E in Quadrant II. The white space between Zones A and E corresponds to negative head, which is not tested for pump/turbines. The correlation of the wicket gate opening with the servomotor stroke is plotted in Figure 12 in model units. Although the model was tested for wicket gate openings up to 40 mm, the full gate or stop position for the prototype is 35 mm in model units.

For four 300 rpm units generating at full gate (35 mm) at static head of 375 m the unit flow is 121.3 m^3/s with a power output of 358 MW. Full load rejection results are plotted in Figure 13, showing maximum speed of 475 rpm. The path of all four machines in Figure 14 falls on the 35 mm curve until the delayed wicket gate action at T = 12 s. The pump/turbines pass through runaway into the braking region, then back into positive torque. Finally, the units progress into reverse pumping before attaining a gate opening of 3 mm.

Pump power failure results are plotted as time histories in Figure 15 and transient path in unit flow unit speed four quadrant diagram in Figure 16. For this mode the flow is 67.3 m^3/s for a static head of 375 m.

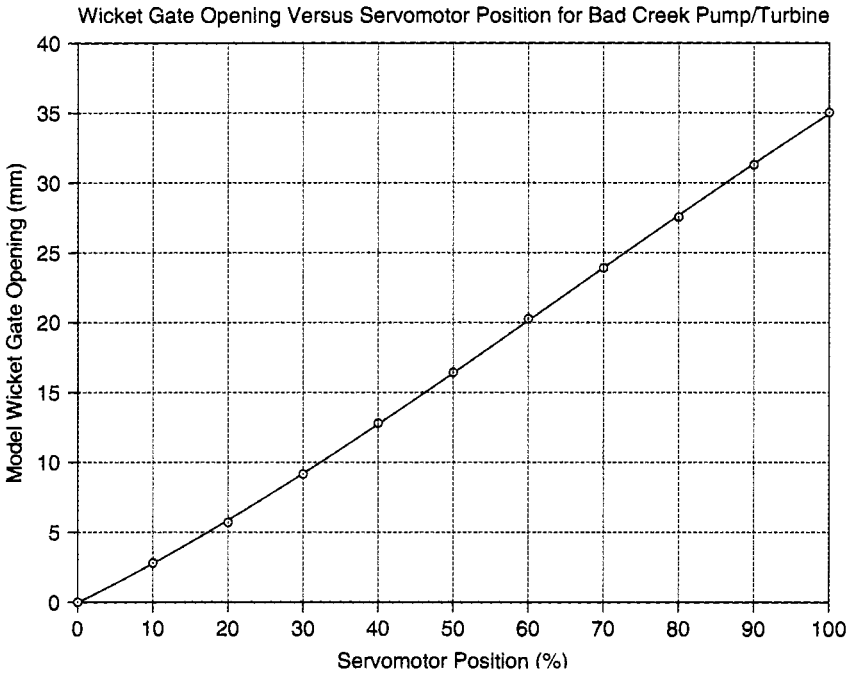


FIGURE 12 Correlation of wicket gate opening with servomotor position

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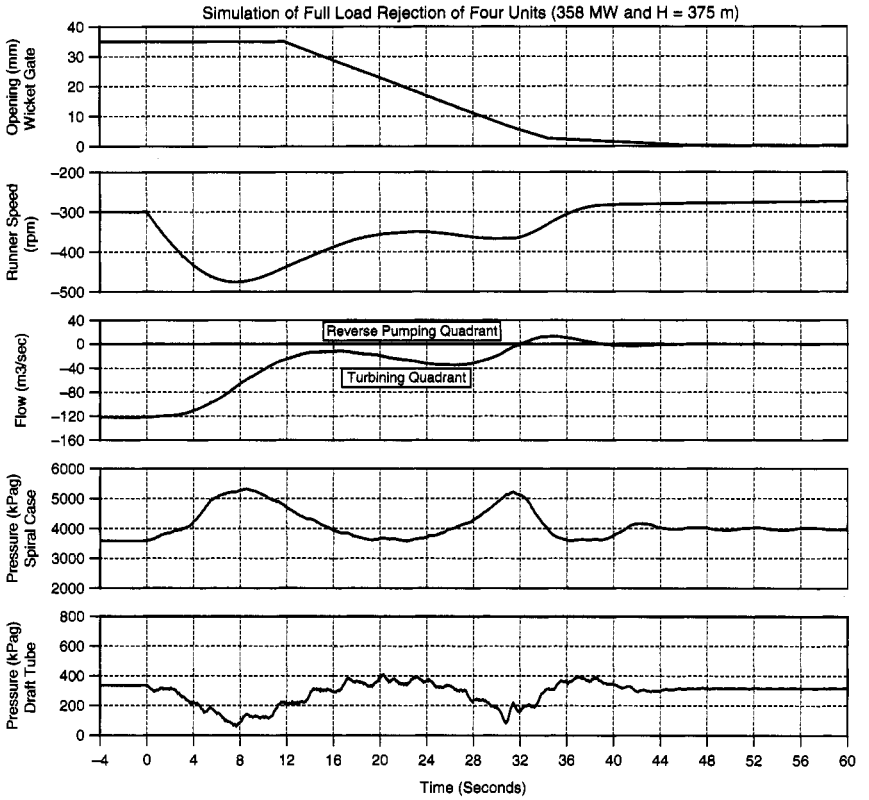


FIGURE 13 Time histories of four unit load rejection for Bad Creek

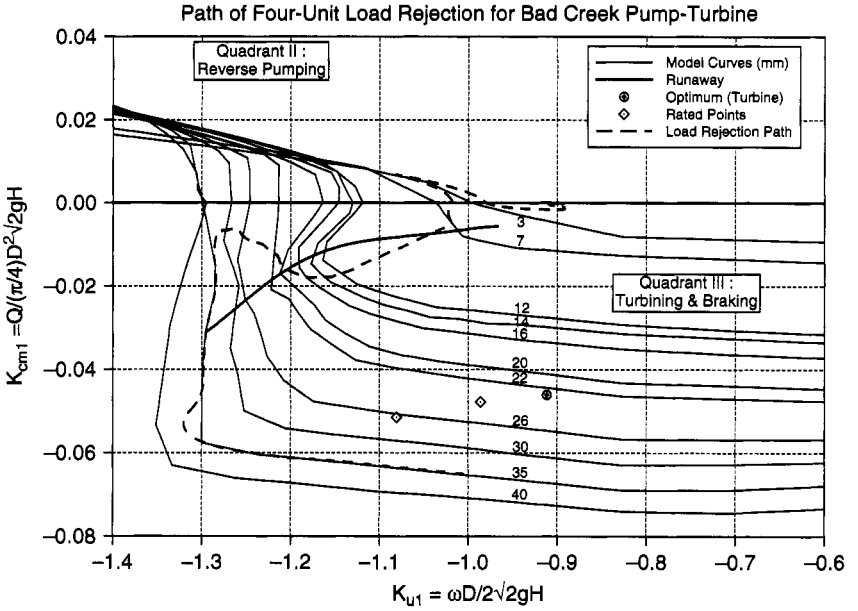


FIGURE 14 Four-quadrant path for four unit load rejection for Bad Creek

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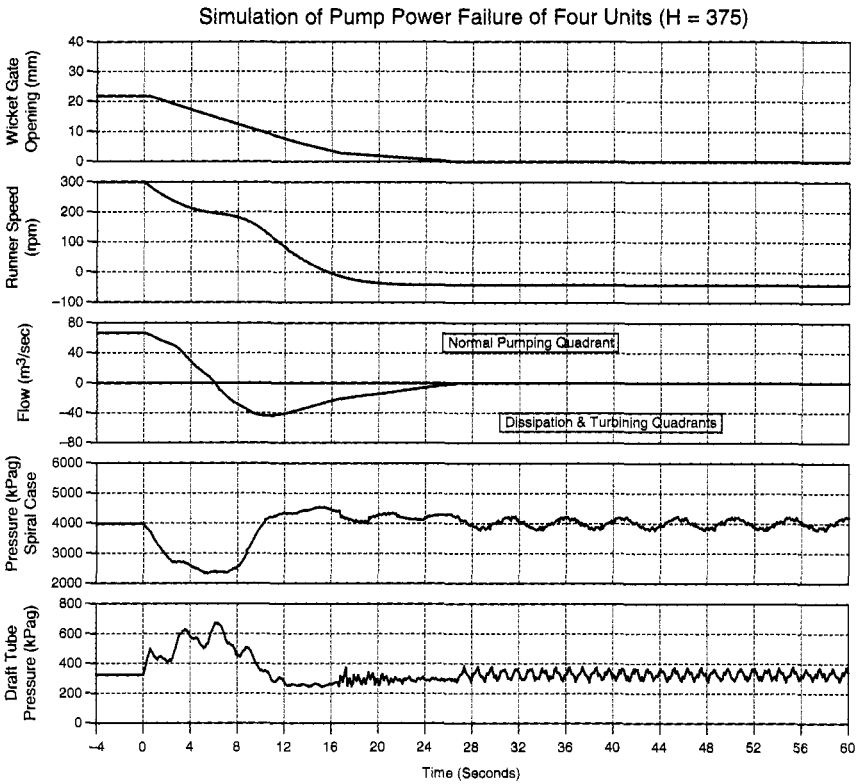


FIGURE 15 Time histories for four unit pump power failure for Bad Creek

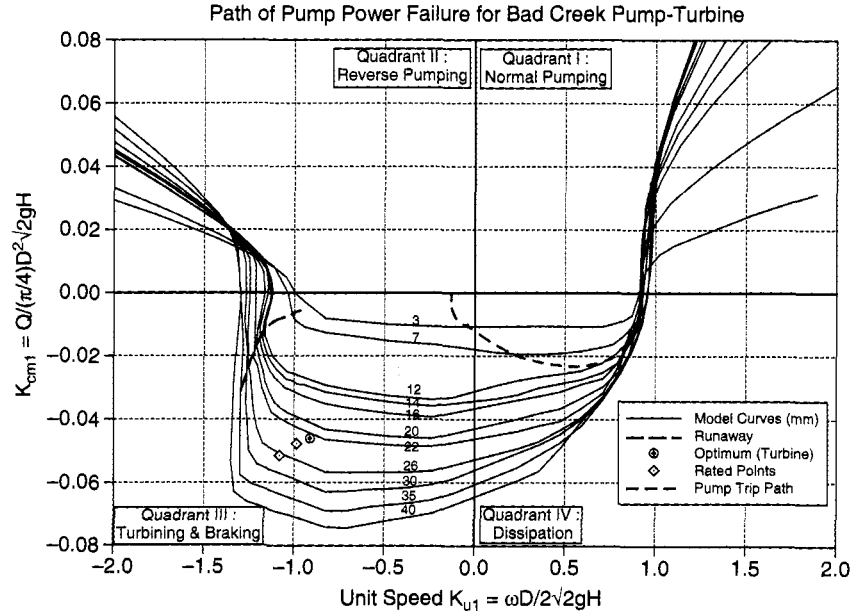


FIGURE 16 Four-quadrant path for four pump power failure for Bad Creek

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SECTION 12.15
METERING

ROBERT H. GLANVILLE
KEN W. TAYLOR

METERING OR PROPORTIONING

Conventional reciprocating pumps can be adapted to function as metering or proportioning devices in the transfer of liquids. The principal adaptation is the addition of a means of varying the pumping rate and predicting what that rate will be, which makes the modified units suitable for use as final control elements in continuous-flow processes. Metering pumps are often employed where two or more liquids must be proportioned or where mixture ratios must be controlled. These effects are achieved by changing the displacement per stroke (by moving the crankpin by special linkages or by partial stroking) or by changing the stroking speed through the use of variable-speed transmissions or electric motors. Accurate cyclic volume compensation can also be performed using electronic calibration.

Four basic types of positive displacement reciprocating pumps, and several variations, are used for this service: packed plunger pumps, pumps with a mechanically actuated diaphragm, pumps with a hydraulically actuated diaphragm, and pumps with hydraulically actuated pistons.

Packed Plunger The packed plunger pump is the most commonly used type because of its relatively simple design and wide range of pressure capability. It is an adaptation of the conventional reciprocating transfer pump (Figure 1). Its advantages are

1. Relatively low cost
2. Pressure capability to 50,000 lb/in^2 (345 MPa) gage
3. Mechanical simplicity
4. Wide capacity range, from a few cubic centimeters per hour to 20 gpm (4.5 m^3/h)
5. High accuracy, better than 1% over a 15:1 range
6. Only slightly affected by changes in discharge pressure

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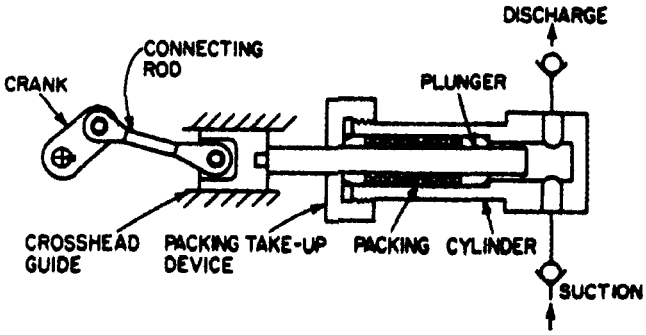


FIGURE 1 Packed plunger pump

Its disadvantages are

1. Packing leakage, making it unsuitable for corrosive or dangerous chemicals
2. Packing and plunger wear and the resulting need for gland adjustment
3. Inability to pump abrasive slurries or chemicals that crystallize

Mechanically Actuated Diaphragm The mechanically actuated diaphragm pump is commonly used for low-pressure service where freedom from leakage is important. This pump utilizes an unsupported diaphragm, which is moved in the discharge direction by a cam and returned by a spring (Figure 2). Its advantages are

1. Relatively low cost
2. Minimum maintenance at 6- to 12-month intervals
3. Zero chemical leakage
4. Ability to pump slurries and corrosive chemicals

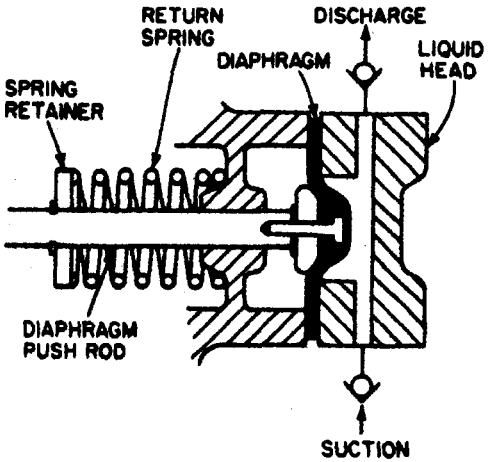


FIGURE 2 Mechanically actuated diaphragm pump

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Its disadvantages are

1. Discharge pressure limitation of 125 to 150 lb/in^2 (860 to 1030 kPa) gage
2. Accuracy in 5% range and as much as 10% zero shift with change from minimum to maximum discharge pressure
3. Capacity limit of 12 to 15 gph (0.045 to 0.057 m^3/h)

Hydraulically Actuated Diaphragm The hydraulically actuated diaphragm pump is a hybrid design that provides the principal advantages of the other types. A packed plunger is used to pulse hydraulic oil against the back side of the diaphragm. The reciprocating action thus imparted to the diaphragm causes it to pump in the normal manner without being subjected to high pressure differences. A flat diaphragm design is shown in Figure 3 and a tubular diaphragm design in Figure 4.

The advantages of the hydraulically actuated diaphragm pump are

1. Pressure capability to 5000 lb/in^3 (34.5 MPa) gage
2. Capacities to 20 gpm (4.5 m^3/h)
3. Minimum maintenance
4. Zero chemical leakage
5. Ability to pump slurries and corrosive chemicals
6. Accuracy around 1% over 10:1 range

Its disadvantages are

1. Subject to predictable zero shift of 3 to 5% per 1000 lb/in^2 (6.9 MPa) gage
2. Higher cost

Hydraulically Actuated Pistons This type of design is used when high accuracy at relatively low pressure is required. Figure 5 shows the complete pumping and metering unit, whereas Figure 6 shows a sectioned view of the metering unit. The pumping unit shown in Figure 6 can be a gear or vane type, driven by an electric motor. This is integral to a dual-measuring unit (Figure 5), mounted on top of the pumping unit. Alternatively, a centrifugal pump located remotely in the product storage tank can provide the pumping function. The pumping unit is run at a constant speed of about 850 rpm. A bypass valve in the pumping unit varies the output from 0 to 100% in response to a control valve located downstream of the meter.

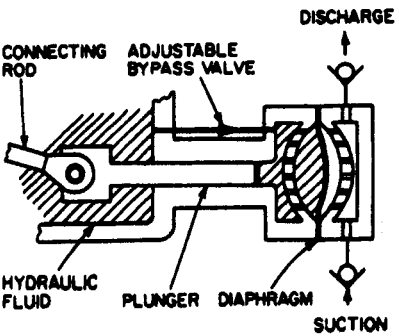


FIGURE 3 Hydraulically actuated diaphragm pump with flat, circular diaphragm

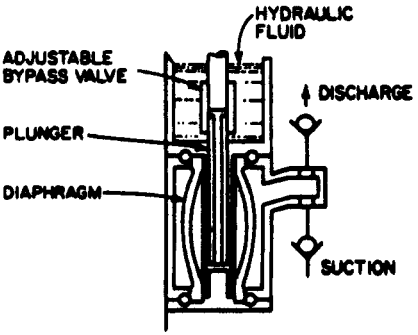


FIGURE 4 Hydraulically actuated diaphragm pump with tubular diaphragm

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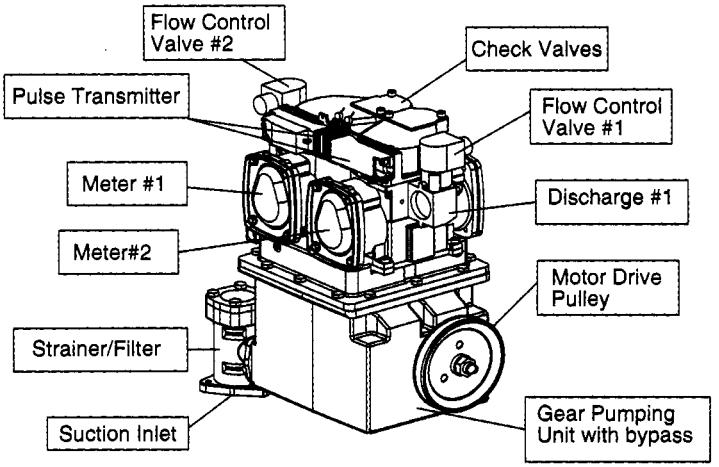


FIGURE 5 "Duplex" metering pump (Dresser-Wayne)

The metering unit shown in Figure 5 is a dual unit providing two separate measured outputs. Each meter unit has two pistons and three chambers. Scotch yokes that drive a rotary valve that ports the fluid in and out of the measuring chambers connect the pistons. A shaft encoder and calibration unit is integral to the meter to provide accurate adjustment of the pumped volume together with batch and running volume totals. The advantages are

1. High accuracy (0.25%) across complete flow range to 25 gpm (6 m^3/h)
2. Compact, totally integrated multifunction design.

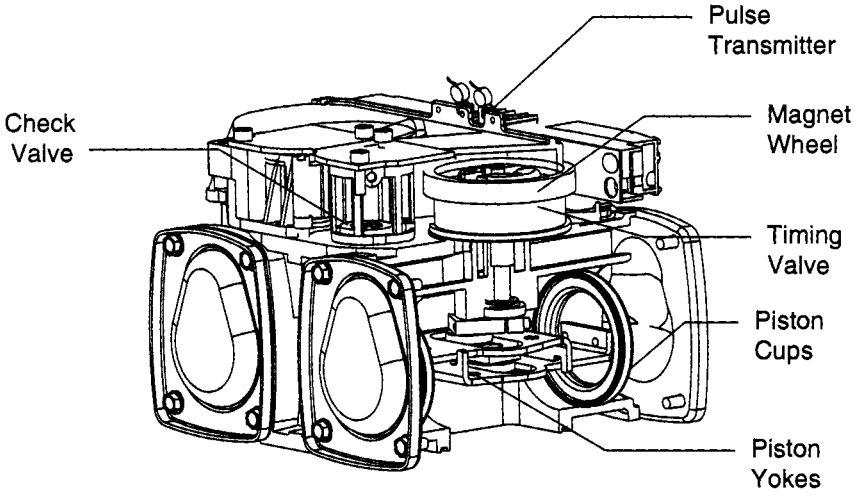


FIGURE 6 "Duplex" meter (Dresser-Wayne)

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3. Minimum maintenance every 1.3–1.5 million gallons (5000–6000 m^3)
4. Low initial and ownership cost

Its disadvantages are

1. Pressure limitation of 100 lb/in^2 (700 kPa).
2. Maximum flow rate of approximately 25 gpm (6 m^3/h)
3. Only suitable for low viscosity noncorrosive products.

CAPACITY CONTROLS

There are five methods commonly used to adjust the capacity of metering pumps. The choice of which method to use is determined by the application for which the pump is intended.

Manually Adjustable While Stopped This control is generally found on packed plunger pumps of conventional design. Capacity changes are effected by moving the crankpin in or out of the crank arm while the pump is not in motion. This is the least expensive method and is used where frequent changes in pump displacement are not required.

Manually Adjustable While Running This feature is most frequently found on mechanically actuated diaphragm pumps, where it is accomplished by limiting the return stroke of the diaphragm with a micrometer screw. On packed plunger pumps, stroke adjustment while the pump is running is relatively complicated. Stroke length is set by some type of adjustable pivot, compound linkage, or tilting plate, which is manually positioned by turning a calibrated screw. On hydraulically actuated diaphragm pumps, relatively simple control is provided by manually adjustable valving that changes the amount of the intermediate liquid bypassed at each stroke.

Pneumatic Meeting pumps used in continuous processes must be controlled automatically. In pneumatic systems, the standard 3- to 15-lb/in^2 (21- to 103-kPa) gage air signal is utilized to actuate air cylinders of diaphragms directly connected to the stroke-adjusting mechanism.

Electric On electric control systems, stroke adjustment is through electric servos that actuate the mechanical stroke-adjusting mechanism. These accept standard electronic control signals.

Variable Speed This method of adjusting capacity is achieved by driving a reciprocating pump with a variable-speed prime mover. Because it is necessary to reduce stroke rate to reduce delivery, discharge pulses are widely spaced when the pump is turning slowly. Surge chambers or holdup tanks are used when this factor is objectionable.

In control situations involving pH and chlorinization, two variables exist at once; for example, flow rate and chemical demand. This is easily handled by a metering pump driven by a variable-speed prime mover. Flow rate can be adjusted by changing the speed of the pump, and chemical demand by changing its displacement.

CALIBRATION

Calibration, which is a fine adjustment of capacity, can be performed in the same way as the capacity controls previously discussed by using a fine stroke adjustment mechanism. It can better be achieved using electronics linked to the pumped volume display. A shaft encoder is connected to the meter that outputs pulses proportional to the volume pumped.

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The number of pulses is selected relative to the display resolution required. If $\frac{1}{100}$ th of a liter were required on a nominal cyclic volume of 1 liter, 105 pulses would be provided per revolution. The pulses are fed into counting and processing electronics. The electronics are put in calibrate mode and a fixed measure pumped into a calibrated vessel. The electronics counts the number of pulses received for the known volume pumped, compares that to the number of pulses it should have received, and calculates a calibration factor for that meter. This factor is stored in nonvolatile memory, and used by counter to "discard" extra pulses when the pump is in normal delivery mode. The 105 pulses per revolution is used to make sure that there are always extra pulses to discard allowing a ±5% manufacturing variation on the nominal cyclic volume. The calibrating electronics can also store batch and lifetime volumes pumped totals, and supply this information to a local or remote display.

SERVICE AND MAINTENANCE

Proportioning pumps utilizing manual capacity controls can be installed, serviced, and operated by plant personnel. Pneumatic capacity controls are more sophisticated, but after their construction and function are understood, maintenance and service should become routine. Electric capacity controls require a basic understanding of electric circuits for installation, operation, and routine service. Modular construction allows repair service by replacement of component groups, thus diminishing the task of trouble-shooting.

All proportioning pumps utilize suction and discharge check valves. These require regular maintenance and service because they are used frequently, encounter corrosion, and must be kept in good working order to ensure accurate pump delivery. Service periods are greatly dependent on liquids pumped, pump operating speed, and daily running time. Usual service periods run from 30 days to 6 months or longer. Check valves are designed to facilitate service with a minimum of downtime, thus allowing replacement of wearing parts at minimum expense. Good design allows this service to be accomplished without breaking pipe connections to the pump.

Packed plunger pumps require periodic adjustments of the packing takeup device to compensate for packing and plunger wear. Packing has to be replaced periodically, and regular lubrication of bearings and wear points is required.

The lubricating oil in speed-reduction gearing, either separate integral units or built in, must be changed at six-month intervals.

Diaphragm pumps usually require replacement of the diaphragm as part of routine service at six-month intervals. On hydraulically actuated diaphragm pumps, the hydraulic fluid must also be changed.

Pneumatic and electric controls generally present no special maintenance problems. The frequency of routine cleaning and lubrication is dependent on environmental conditions and should be consistent with general plant maintenance procedures.

INSTALLATION

Proper installation of metering pumps is very important if reliable pump operation is to be obtained.

NPSH must be kept as high as possible, and the manufacturer's recommendations as to pipe size and length, strainers, relief valves, and bypasses must be observed.

MATERIALS OF CONSTRUCTION

With the tremendous variety of liquid chemicals used in industry today, an all-inclusive guide to suitability of materials for pump construction is virtually impossible. For the great majority of common chemicals, pump manufacturers publish data on materials of

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construction. In general, packed plunger and hydraulically actuated diaphragm pumps are available as standard construction in mild steel, cast or ductile iron, stainless steel, and plastic. Mechanically actuated diaphragm pumps are usually available as standard construction in plastic and stainless steel. Almost any combination of materials can be furnished on special order.

Diaphragms are available as standard construction in Teflon, chemically-resistant elastomers, and stainless steel from various manufacturers.

See also Section 3.6, "Diaphragm Pumps," and Section 3.5, "Displacement Pump Flow Control."

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SECTION 12.16
CRYOGENIC PUMPS FOR LIQUEFIED GAS SERVICE

JINKOOK LEE

Cryogenic fluids have normal boiling points below 100°F (or −73°C), and include liquid oxygen, hydrogen, nitrogen, helium, argon, methane, and ethane.

Since the start of the twentieth century, there has been increasing demand for high quality cryogenic pumps for space and missile technologies as well as commercial applications. It is always critical for cryogenic pump designers to consider the rate of thermal shrinkage of each pump component to be certain of sufficient clearances that allow the pump to run smoothly with high efficiency at very low temperatures.

LIQUEFIED HYDROCARBON

Since World War II, liquid rocket engine development has advanced significantly in military and commercial applications. On these engines, high-speed centrifugal, turbine-driven cryogenic pumps supply liquid oxygen as an oxidizer and various types of fuel such as liquid hydrogen, kerosene, H_2O_2 to be burned in the engine. The inducer suction specific speed N_{ss} for liquid hydrogen pumps is reported to run in excess of 35,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$) and as high as 70,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$) because this liquid is near its thermodynamic critical point. (See Section 12.18.2.)

After the oil shocks in the 1970s and the increasing worldwide demand for energy coupled with the international clean air policies set in the 1980s, gas is regarded as the key transition fuel of the twenty-first century because of its environmental benefits in reducing greenhouse gas emissions. The U.S. government's Energy Information Administration (EIA) predicts that world gas consumption will nearly double by 2020, with its share of the total energy supply increasing from 23% in 1999 to 28% in 2020 and that, by 2030, developing countries will account for more than 60% of the increase in world energy. The cryogenic fluids in this service are primarily liquefied natural gas (LNG), which is normally a mixture of more than 90% methane at −162°C or −260°F, ethylene, and ethane; along with the liquefied petroleum gases (LPG) propane and butane. Therefore, demand of LNG and LPG as clean energy sources will increase the demand for cryogenic pumps in the import, transportation, and export gas markets.

Over its 45-year history, the submerged motor-driven cryogenic pump has been used in the LNG business because cryogenic fluids are excellent dielectrics with low lubricity.

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A portion of the pumped fluid is guided through the motor gap between the rotor and stator to cool the motor and flows downward to lubricate the bearings. Motors are running at 2-pole and 4-pole speeds at 50 and 60 Hz depending on the application, such as pumps in a land-based receiving terminal or marine cargo pumps in LNG carrier tankers.

What Is Natural Gas? Natural gas is a flammable gaseous mixture made up essentially of hydrocarbons (chemical compounds that contain carbon and hydrogen), which is found along with coal and petroleum. Natural gas is a fossil fuel, which exists in a gaseous form in either sedimentary rock in the earth's crust or in layers of organic matter that has accumulated at the bottom of bodies of water from the Precambrian to the Conozoic Era.

Natural gas is obtained through the four stages of exploring, confirmation of its existence, drilling, and mining. Natural gas, depending on where it is obtained, can contain as much as 90% methane (CH_4) with small quantities of ethane (C_2H_6) and even smaller amounts of propane (C_3H_8), butane (C_4H_{10}), pentane (C_5H_{12}), and other alkines. Natural gas is extracted and refined into fuel that provides approximately 25% of the world's energy supply.

OVERVIEW OF LNG PRODUCTION AND SUPPLY CHAIN

The simplified LNG production process and supply chain can be summarized as follows:

Offshore Gas Production → Onshore Gas Processing → LNG Export Terminal → Shipping → LNG Receiving Terminal → End User

Gas Production Facilities Natural gas upon being extracted and refined is then liquefied. This processed natural gas is known as LNG. LNG takes up six hundredths of the volume of natural gas, which allows for efficient storage and long-distance transportation.

Liquefaction Methods

1. *Turbo Expander Cycle*

   When natural gas is compressed under high pressure and expanded using turbines, its temperature drops sharply and it enters a liquid state.

2. *Cascade Cycle*

   The cascade cycle uses various refrigerants to liquefy natural gas. First, propane is liquefied by using ethylene as a refrigerant. Then, natural gas is liquefied by using this liquefied propane as a refrigerant.

3. *Multicomponent Refrigerant Cycle*

   In this cycle, natural gas is condensed and expanded by using a mixture of nitrogen and hydrocarbons as a refrigerant until the temperature drops to the liquefaction point.

Liquefaction Facilities

STORAGE TANKS LNG storage tanks at the natural gas liquefaction facilities have the same structure as the ones at the receiving terminals. Storage tanks at the LNG fields are used for temporary storage before LNG is shipped to its destination. LNG storage tanks have double walls like a thermos. Their exterior is made of a poststressed concrete wall, which is designed to absorb external shocks and maintain the tank's cohesiveness. The interior wall is a sealed stainless steel membrane specially designed to store LNG. Between the interior and exterior walls, polyvinyl chloride (PVC) is inserted to insulate the tank and so maintain a low temperature, thereby preventing the LNG from reverting

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back to a gaseous state. The primary (in-tank) pumps in the storage tanks pump liquefied gas to recondensers, and the vaporized gas in the storage tank is sent by pipeline to the boil-off gas (BOG) compressors for reliquefaction.

LOADING SYSTEM After LNG is produced and stored in tanks at production sites, it travels to the final destinations by either moss or membrane-type LNG tanker vessels. The type of vessel is determined by the size of storage tank. Loading systems used to fill the tanker vessels include LNG pumps, pipelines, loading arms, and BOG facilities to handle the gas generated in the process of loading.

GAS RECEIVING TERMINAL FACILITIES The size of an LNG receiving terminal ranges from less than 1 million tons per annum (tpa) to over 10 million tpa of gas sent out. Typical end uses for the imported gas are as follows:

- Power generation for metropolitan areas
- Industrial/domestic fuel for customers in metropolitan areas
- Power generation for regional use
- Industrial/domestic fuel through pipe line for regional use
- Feedstock for manufacturer of other chemicals
- Alternate source of fuel for transportation (CNG/LNG vehicle)
- Others

A typical LNG receiving terminal simplified flow diagram is shown in Figure 1.

UNLOADING SYSTEM At the LNG unloading facilities, the LNG tanker is towed to the production site. Upon docking, the ship is moored to a jetty to minimize movements during the unloading process. Once securely moored, three LNG unloading arms are connected to one NG unloading arm, and LNG is transported to ground storage tanks via pipelines

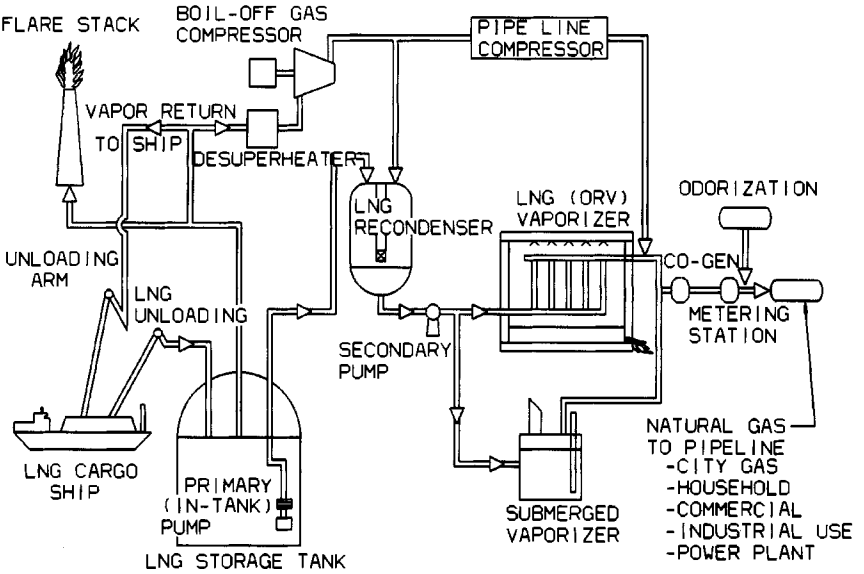


FIGURE 1 Typical LNG receiving terminal simplified flow diagram

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using cargo pumps on the tanker. The NG unloading arm is connected to LNG unloading arms to maintain the pressure inside the storage tank on the LNG vessel. The pressure inside the vessel drops when LNG is unloaded, but the NG arm maintains this pressure by supplying boil-off gas generated at the production site. One LNG tanker vessel can unload its capacity of between 125,000 m^3 (approximately 57,000 tons) and 135,000 m^3 (about 62,000 tons) of LNG within an average of 12 h.

The LNG receiving platform and a typical marine cargo pump are shown in Figures 2 and 3. The LNG unloading system also includes facilities for disconnecting the ship-to-shore interface at the end of the unloading operation, and for undocking the ship prior to its return voyage.

LNG UNLOADING ARMS As the typical unloading time for a full-size tanker (125,000 to 140,000 m^3) is 12 to 15 h, this requires peak unloading rates of about 12,000 m^3/h. This can be accomplished by two 16-in unloading arms, each capable of transferring 6000 m^3/h. The transfer pipe portion of the unloading arm is made of stainless steel, capable of withstanding cryogenic temperatures below 162°C. Typically, the shipboard cargo pumps provide a head of about 100 to 140 m. The jetty unloading platforms, which typically have four 16-in LNG arms and one 16-in vapor return arm, are shown in Figures 4 and 5.

STORAGE SYSTEM The storage system provides a buffer between the LNG discharged from the ships and the vaporizers that regasify the LNG. The system consists of one or more specially designed tanks. Ships carrying the LNG arrive at the terminal at specified intervals. The minimum required storage capacity is the volume of LNG discharged from the largest ship expected at the terminal. While the typical capacity of an LNG tanker is around 140,000 m^3, it is anticipated to grow to 200,000 m^3 in the near future. In practice, the installed storage is larger than this minimum. The extra storage provides a cushion to account for scheduled and unscheduled delays in ship arrival. A typical land-based LNG receiving tank is shown in Figures 6 and 7.

Most large LNG tanks in service today, including all the large LNG tanks in the United States, are of the single-containment, or double-containment design and built in accordance with American Petroleum Institute (API) Code 620, Appendix Q, BS7777, and Recommended Practice In-Ground Storage Tank (RPIS).[4,5] Typical design temperature inside

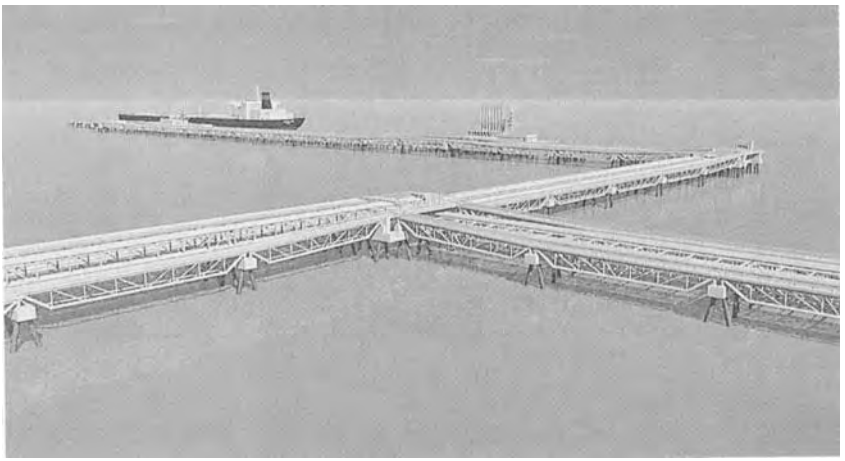


FIGURE 2 Typical LNG receiving platform

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FIGURE 3 Typical shipboard cryogenic pump with flow rates to 1500 m^3/h (Courtesy of Carter Cryogenics)

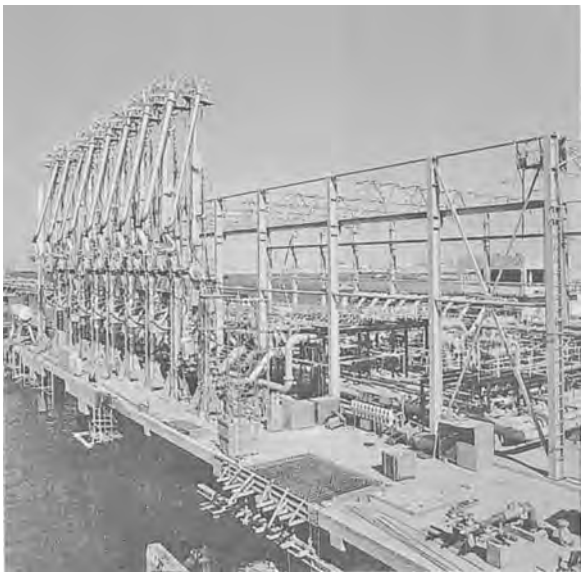


FIGURE 4 Typical receiving jetty arms

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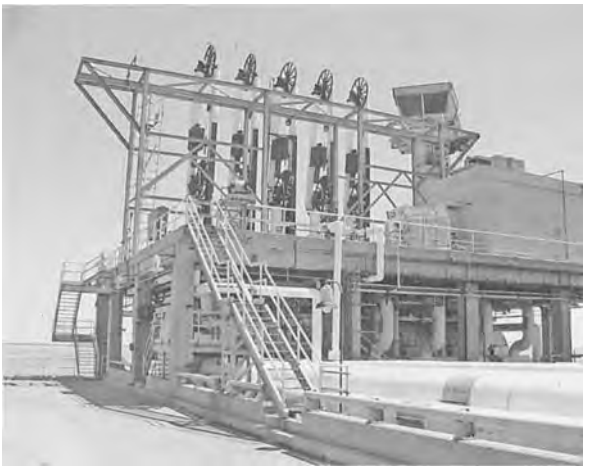


FIGURE 5 Typical receiving jetty arms

of the tank is –170°C; the range of design pressure is from –5 to 450 mbarg; and operating pressure is from 50 to 350 mbarg.

BOG COMPRESSOR BOG compressors literally compress the boil-off gas from LNG storage tanks and pipelines. LNG boils at an ultra-low temperature of –162°C, and can easily boil due to heat transmitted from the outside of tanks and pipelines. BOG compressors reliquefy this vaporized gas by raising pressure. Ethane and propane have the relatively higher boiling points of –88.6°C and –42°C, respectively. Therefore, BOG from LNG is almost entirely methane.

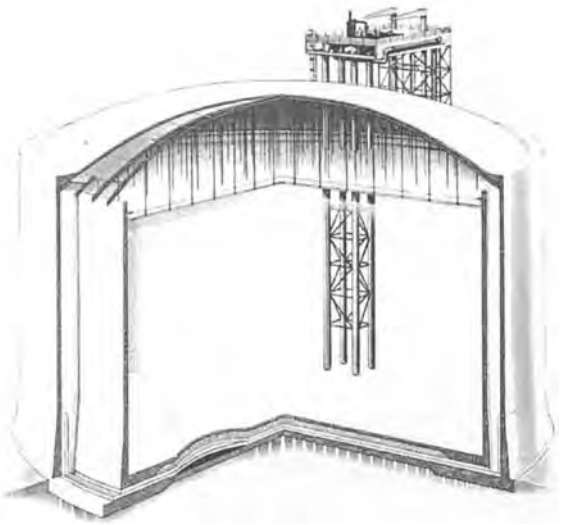


FIGURE 6 Typical land based LNG receiving tank

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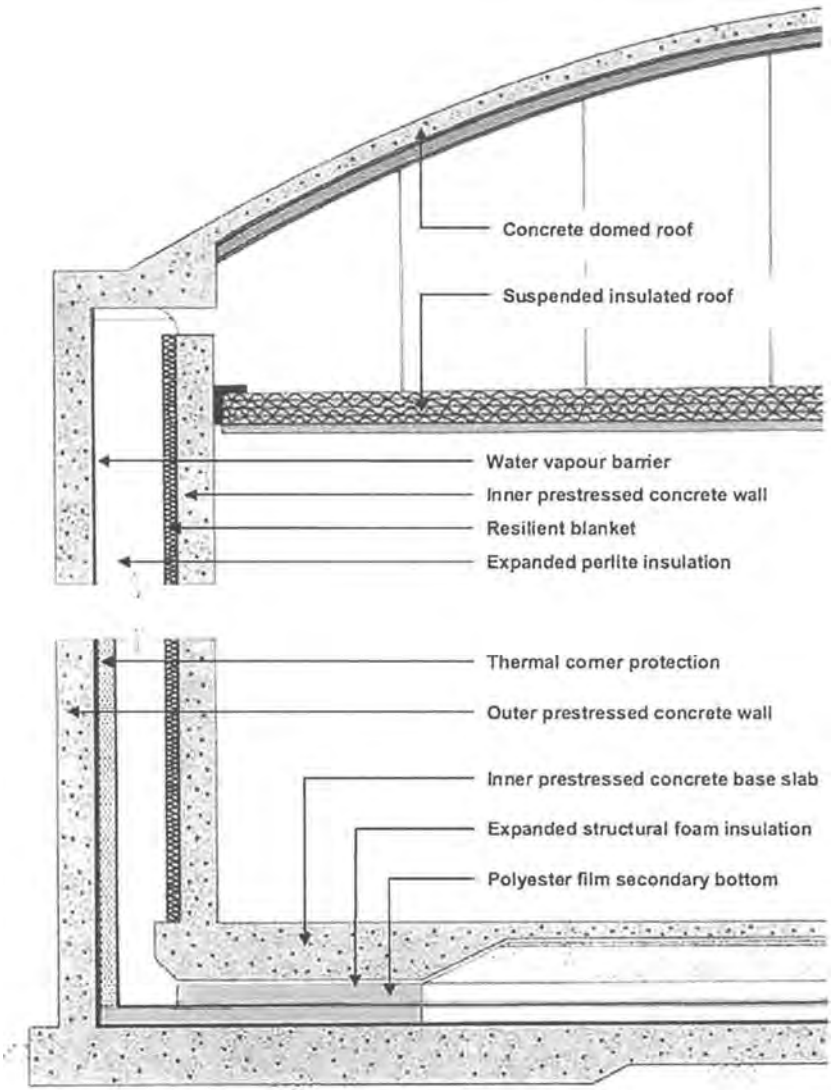


FIGURE 7 Typical all concrete primary containment

RECONDENSERS A recondenser liquefies BOG compressed by the BOG compressor by blending it with LNG. LNG, compressed to a higher pressure by primary (in-tank) pumps in the storage tank, and the BOG from the BOG compressor each separately flow into the condenser and are mixed. The LNG with its low temperature (–150°C to –160°C) and BOG with a relatively high temperature (5°C to 35°C) mix and condense to LNG (–130°C to –150°C) through a heat exchanger reaction in the recondenser, and is then transported to the secondary (ex-tank or sendout) pumps. LNG boils at a higher temperature than –162°C when the pressure is high.

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LNG PUMPING SYSTEM

In-Tank Pump LNG tanks operate at very low pressure, just slightly above atmospheric pressure. The gas from the terminal needs to be delivered at an elevated pressure. Since pumping the liquid is easier and less expensive than compressing gas, the LNG is pumped to the required sendout pressure prior to its regasification. The pump-out system may consist of one or two stages of pumps, depending on the required battery-limit pressure of the gas and on the configuration of the BOG handling system. Pumping of cryogenic liquids, especially at the high rates required in LNG facilities, is specialized technology. In most cases, in-tank pumps are driven by 3-phase, 2-pole, or 4-pole motors at speeds corresponding to 50 and 60 Hz. In general, land-based in-tank pumps operate at 50 Hz. However, with a variable frequency drive (VFD), they can operate at any speed. In modern terminals the first-stage pumps almost always are installed inside the storage tanks, and are referred to as in-tank pumps as shown in Figures 8, 9, 10, and 11.

Ex-Tank Pump A secondary pump (ex-tank pump or sendout pump) pumps compressed LNG from low-pressure primary LNG pumps and recondensers to high pressure. An extremely high pressure level is required to deliver natural gas via a long pipeline network across the nation to the end users. As it is easier to increase the pressure of a liquid than a gas, the liquefied form of LNG is compressed normally up to 80 bars by secondary

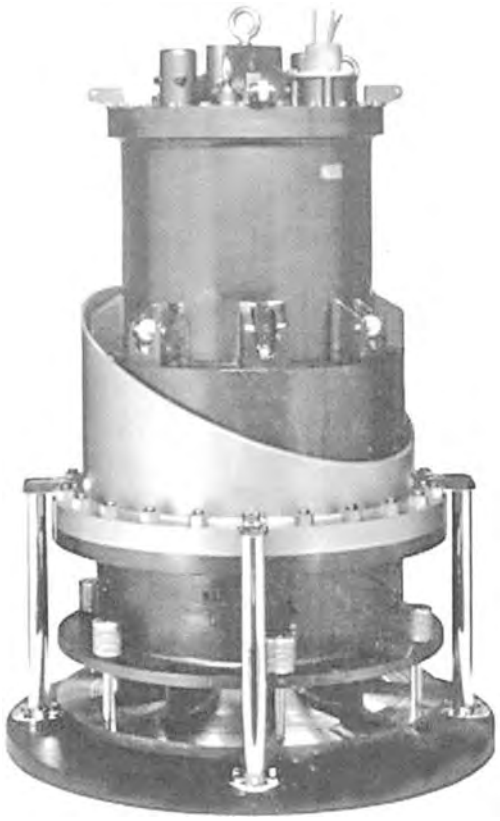


FIGURE 8 Typical removable in-tank cryogenic pump with flow rates to 2200 m^3/h (Courtesy of Carter Cryogenics)

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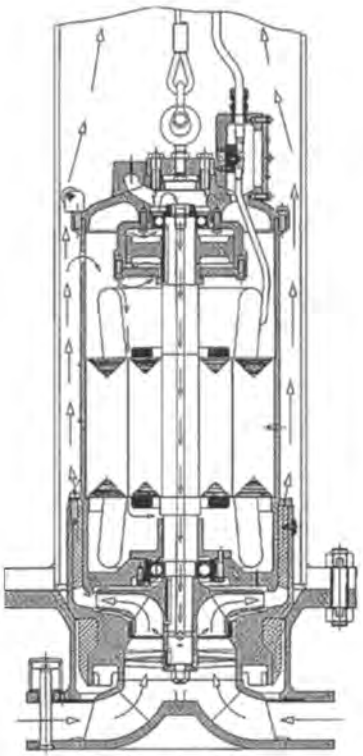


FIGURE 9 Cross-sectional view of typical single-stage cryogenic in-tank pump showing flow passages through the motor (Courtesy of Carter Cryogenics)

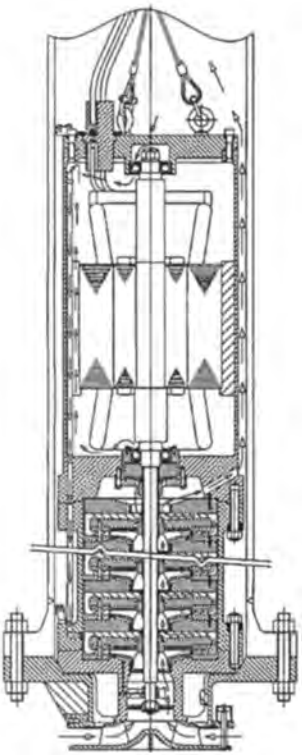


FIGURE 10 Cross-sectional view of typical in-tank multistage cryogenic pump (Courtesy of Carter Cryogenics)

pumps. In addition, fewer calories are consumed in vaporizing gas when the LNG pressure is high. Secondary pumps supply highly compressed LNG to high-pressure vaporization facilities. The number of secondary pumps is determined by the amounts to be delivered through the network pipelines.

These second-stage pumps, when required, are located outside the tanks, in the process area. Second-stage pumps discharge at a pressure sufficiently high to satisfy the battery limit pressure at the terminal fence. Cross-sectional views of single and multistage suction vessel-mounted ex-tank submerged motor pumps are shown in Figures 12, 13, 14, and 15.

Vaporizer A seawater open rack vaporizer supplies vaporized LNG from secondary pumps to the pipeline. There are two types of vaporizers:

- Open rack vaporizer (ORV) that uses the temperature of seawater which is higher than that of LNG
- Submerged vaporizer (SMV) that uses the combustion of gas

The ORV pumps and pours seawater to vaporize LNG, while moving through the thin-layered heat exchanger from bottom to top. The heat exchanger of an SMV is submerged in a water tank, and a burner installed in the tank burns oxygen and natural gas to heat water and vaporize LNG.

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FIGURE 11 Typical removable in-tank multistage cryogenic pump (Courtesy of Carter Cryogenics)

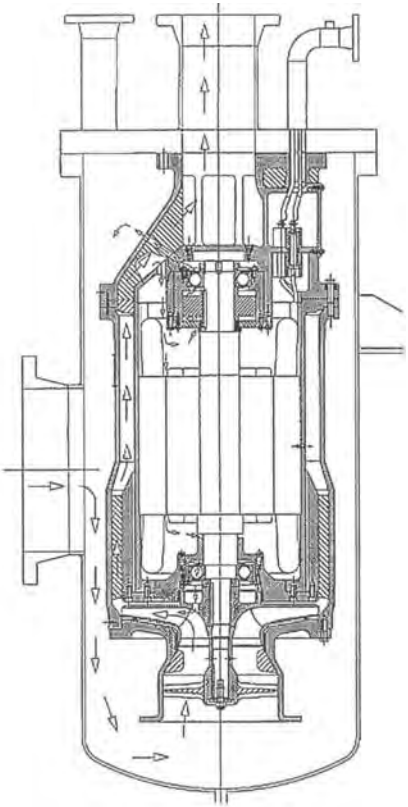


FIGURE 12 Typical suction vessel-mounted ex-tank submerged motor pump—single stage (Courtesy of Cater Cryogenics)

HISTORY AND DEVELOPMENT OF CRYOGENIC PUMPS

It was reported that the first commercial production of LNG was at a liquefaction plant in Cleveland, Ohio, USA, in early 1940s. A LNG carrier, the "Methane Pioneer," made the first successful transport of a commercial cargo of LNG from Lake Charles, Louisiana across the Atlantic Ocean to England in 1959.

This voyage carried an almost token quantity of 5000 m^3 of LNG. This vessel was fitted with deep-well pumps, which were found to be unreliable. An investigation was undertaken to find possible types of pumps that could be used in the transfer of LNG in a safe, simple, and reliable manner. This program identified one of the most suitable types to be a fully submerged electric motor pump based on submerged motor fuel pumps used in the aerospace industry. After long and exhaustive testing, the very first application was realized in late 1961 when a submerged motor pump was installed in the LNG carrier Beauvais followed by the first commercial vessels, the "Methane Progress" and the "Methane Princess" in 1964. These units were found to operate successfully. Similar pumps were later installed in other LNG carriers.

Figure 16 shows the history and development of cryogenic pumps over the last 40 years.

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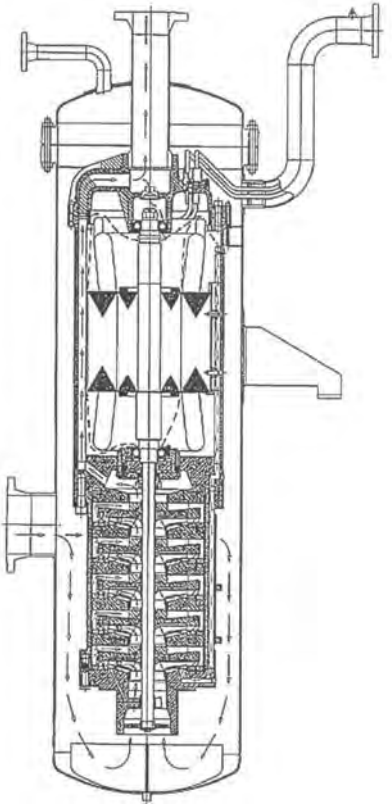


FIGURE 13 Typical suction vessel-mounted ex-tank submerged motor pump—multistage (Courtesy of Carter Cryogenics)



FIGURE 14 Typical multistage ex-tank submerged motor pump—with a maximum of 25 stages and over 8 m tall, 1000 m^3/h, and 5000 HP (Courtesy of Carter Cryogenics)

DESIGN CONSIDERATIONS FOR CRYOGENIC PUMPS

Cryogenic pumps such as LNG pumps incorporate many special features that make them different from other pumps in industrial and process plant services. Some of these features and requirements are as follows:

- Submerged electric motors
- Suction-vessel mounting with suction and discharge connections
- Retractable in-tank mounting for installation within tanks
- Special materials of construction
- Testing at very low temperature and at the high-power levels expected in service
- Cooldown and start-up considerations
- Warm up for teardown inspection
- Handling considerations for pump size and weight
- Required operation at low NPSH and often two-phase flow conditions which need high performance inducers and high-efficiency impellers

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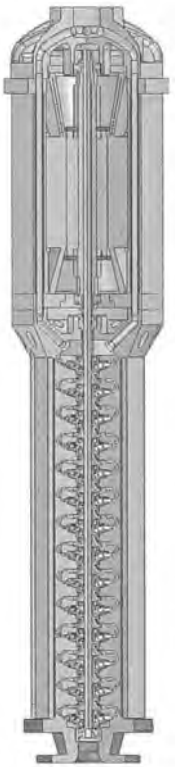


FIGURE 15 Typical multistage ex-tank submerged motor pump (Courtesy of Carter Cryogenics)

Removable In-tank Pump System When the removable pump system is installed in accordance with U.S. National Electric Code, it is suitable for use in Class I Division 1 locations, with leak-free pump maintenance. Figure 17 depicts the components usually supplied as part of the system.

The key to a safe, removable pump system is the foot valve. A patented foot valve ensures that no flammable gas or liquid enters the column during pump installation or removal. The foot valve is attached using a standard pipe flange at the bottom of the pump column. During pump installation, the pump weight opens a spring-loaded valve poppet, admitting liquefied gas into the column. During operation, pump discharge is isolated from pump suction by a conical seal between the pump housing and the foot valve. The tapered seal-type system is self-centering and requires no additional seals, such as O-rings. When the pump is removed, the foot valve poppet closes, isolating the liquefied gas from the column. The pump can be safely removed from the tank with no emission to the atmosphere. For applications where no leakage of liquefied gas into the column can be tolerated after a pump is removed, a redundant purged seal foot valve can be used. During installation and removal, pump facilities should support and protect electrical power cables. These consist of an electrical-power assembly and two antirotation stainless steel wire ropes (support cable and lift cable). The support cable ensures that electrical cables are never exposed to any load. The lift cable facilitates pump removal and is attached to the electrical-cable assembly by means of a one-touch clamp that requires no special tools. This clamp precludes the possibility of anything falling into the pump column during pump installation or removal. Spreader bars center the removal system and prevent excessive movement.

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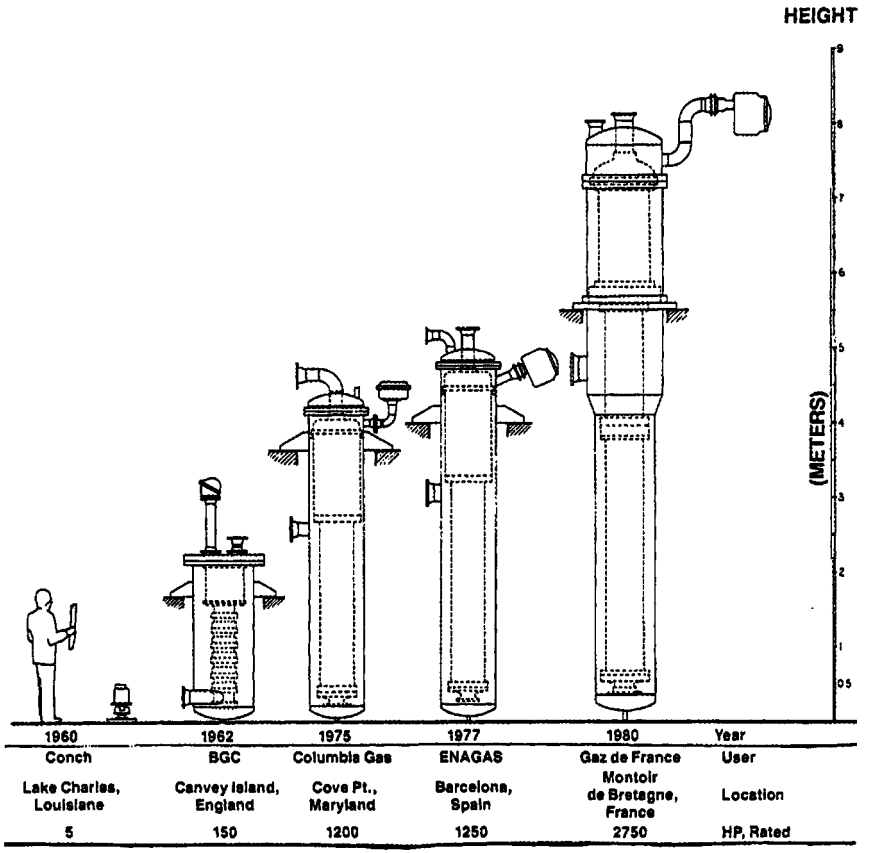


FIGURE 16 Typical history and progress of cryogenic pump development (Courtesy of Carter Cryogenics)

Materials of Construction Head plates, lift-shaft covers, and terminal headers are designed and fabricated in compliance with ASME Pressure Vessel Code. Impeller and pump casings are fabricated from ASTM A356-T6 aluminum alloy. Impellers are hard-coat anodized to provide an erosion-resistant integral hard surface. The strength, light weight, and heat-transfer properties of this material are well suited to cryogenic applications. Inducers are typically machined from AMS 7075-T6 solid aluminum billets. Five-axis machining provides better control of surface finishing and contour, as required in advanced designs to meet low NPSH and possibly higher-gas-volume-fraction two-phase-flow pump inlet conditions. This also enhances strength. After machining, the surface is hard-coat anodized, for cavitation resistance and durability. Some sample cryogenic pump materials are listed in Table 1.

Submerged Electric Motors A submerged cryogenic motor must be highly efficient, because the pump operates in –162°C cryogenic liquid and any heat generated due to the inefficiency of the motor will be the direct source of BOG. Therefore, a small amount of pumped fluid is actually routed through the motor to wet the rotor and stator as a coolant and as well as to serve as a lubricant for the bearings. In normal operation, the motor temperature rarely rises more than 5°C. Like many other hydrocarbon liquids, LNG is not corrosive or electrically conductive. Submerging the motor in LNG totally isolates the motor

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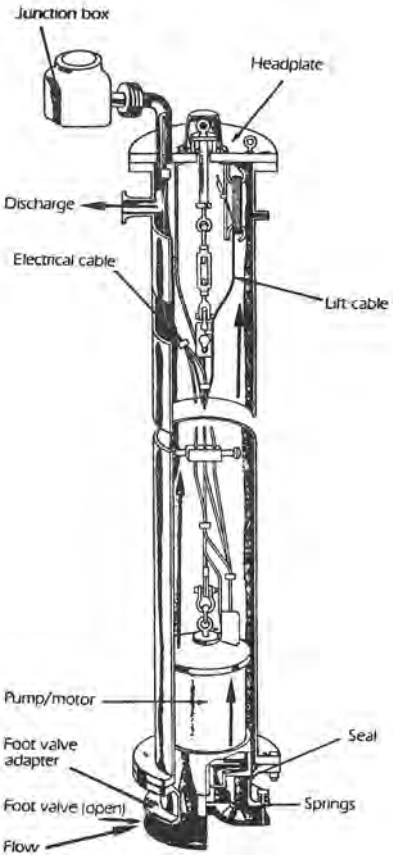


FIGURE 17 Typical removable in-tank pump system (Courtesy of Carter Cryogenics)

TABLE 1 Sample materials for cryogenic pump components (Courtesy of Carter Cryogenics)

Part	Equivalent Material	Specification	ASTM
Head plate	Stainless steel	304	A240/182/312
Lift-shaft cover	Stainless steel	304	A240/182/312
Terminal header	Stainless steel	304	A276
Impeller	Aluminum, A356-T6	AMS 4218	B26
Impeller wear ring	Hard-coat anodize	MIL-A-8625	Type III
Pump casing/diffuser	Aluminum, A356-T6	AMS 4218	B26
Casing wear ring	Bronze	SAE 660	B148
Motor housing	Aluminum, A356-T6	AMS 4038	B209-65
Inducer	Aluminum, 7075-T6	AMS 4038	B209-65
Suction casing	Aluminum, A356-T6	AMS 4218	B26
Bearings	Stainless steel	AISI 440C	A276
Rotor shaft	Stainless steel	17-4PH	A461

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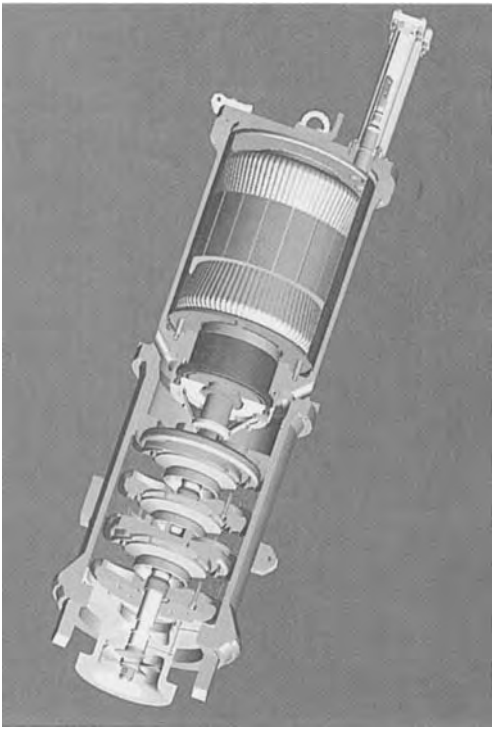


FIGURE 18 Typical submerged pump/motor assembly (Courtesy of Carter Cryogenics)

and its connections from any air or oxygen, which eliminates the possible danger of fire or explosion. Figures 18, 19, and 20 show the typical cryogenic submerged pump/motor assembly, motor rotor, and motor performance, respectively.[2,3]

Motor Stator Design for Stability and Moisture Resistance For the stability of motors above 200 kW, a form-wound stator system has been developed. Smaller motors are often random or "mush" wound for economy. All stators are vacuum pressure impregnated (VPI) to provide a moisture-impervious assembly. This keeps the insulation free of moisture even if the pump is disassembled when cold.

Rotor Design for Strength and Stability Rotors are made of standard electrical silicon-steel laminations with cast or fabricated aluminum bars and end rings. The rotor is mounted to the shaft with a controlled-interface fit. The assembly is quenched in liquid nitrogen (LN_2) for stability and then dynamically balanced in two planes to ISO standard 1940/1 Grade 2.5 to minimize vibration.

Hydraulic Design In recent years, hydraulic components such as the inducer, interstage guide vanes (IGV), impeller, and crossover diffusers are designed in most cases by commercially available computer programs and in-house-developed programs. The ability of computational fluid dynamics (CFD) to make reliable predictions has advanced rapidly and can enable avoidance of expensive design and testing practices. As discussed in Section 2.1.2, commercial CFD codes are available to use as an aid in the design process.

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FIGURE 19 Typical rotor of cryogenic pump motor (Courtesy of Carter Cryogenics)

Design procedures and integrated elements of systems are as follows:

- Kickoff meeting to review design specification
- Preliminary design review
- Detail (or critical) design review
- Sophisticated analysis tools are applied to both design and design verification such as
  - Rapid CFD for hydraulic design
  - Finite element analysis (FEA) for stress analysis
  - Rotor dynamics for vibration analysis

This system has specialized five-axis machining software, design for assembly (DFA) and design for manufacture (DFM) tools, and a smooth transfer of data to computer aided design (CAD) packages.

Inducer Designs for NPSH Performance Most cryogenic pumps require inducers to meet low pump inlet NPSH requirements over the wide flow ranges required. There are several types of inducers designed to meet the requirements at the pump inlet, such as fan inducers, Archimedes-type or spiral inducers, and the high performance inducer (HPI) illustrated in Figure 21a (side view) and Figure 21b (frontal view).

The suction specific speed N_{ss} of these inducers often reaches 80,000 to 100,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$). Even these values have been exceeded in tests of the HPI, which pumped down to 0.1 m of NPSH at the intake of an LPG pump at a flow rate of 1000 m^3/h.[1]

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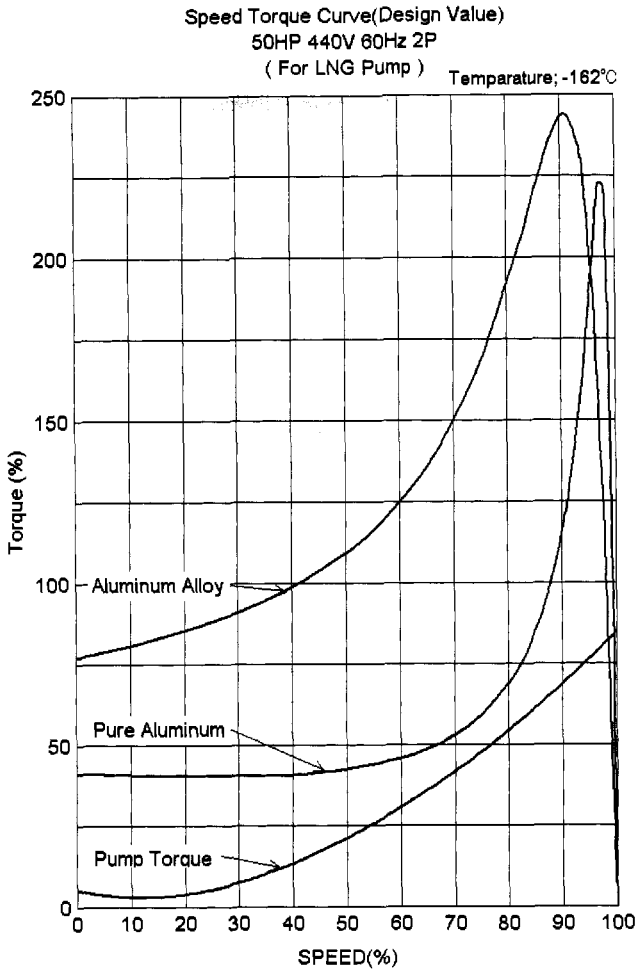


FIGURE 20 Typical cryogenic pump submerged motor performance (Courtesy of Carter Cryogenics)

The HPI type of inducer has also demonstrated the capability of ingesting two-phase flow having a gas- (or vapor-) to-liquid volume ratio (V/L) of up to 0.5 in normal cases and sometimes up to 1.0 in extreme aerospace applications, particularly in modern large commercial jet engine main fuel pumping systems (see Section 12.18.1).

Impeller Design Most cryogenic pump impellers are single-suction, radial-flow, and closed-type impellers with one-piece construction and serrated wear rings. Single-stage pumps can be fitted with back wear rings to assist in thrust balancing. A typical 3-D solid model of a radial-flow impeller is shown in Figure 22a (without the front shroud in place) and Figure 22b (with the shroud in place).

A typical performance map with different sizes and sets of impellers is shown in Figure 23. At a given specification requirement, head rise versus operating flow rate at a chosen pump speed can be easily found from performance charts of this kind.

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FIGURE 21A Typical High Performance Inducer (HPI) (Courtesy of Carter Cryogenics)



FIGURE 21B Typical high performance inducer (HPI) (Courtesy of Carter Cryogenics)

Diffuser Design Multistage pumps are equipped with several types of diffusers. When combined with radial-flow impellers and radial diffusers, this design does not generate the unbalanced hydraulic forces that are common in axial diffusers. The inherent stability prevents premature wear of guide bushings and wear rings. The radial diffuser stacks together with other diffusers in a way that ensures concentricity with the rotating elements. The design permits closer running clearances to enhance efficiency. A radial diffuser is shown in Figure 24.

The continuous crossover diffuser, shown in Figure 25, has been incorporated in the design of a recent multistage pump. CFD tools were utilized in the design process, which enabled designers to maximize overall pump performance and reduce the total length of the shaft, resulting in minimized total height and weight of the pump.

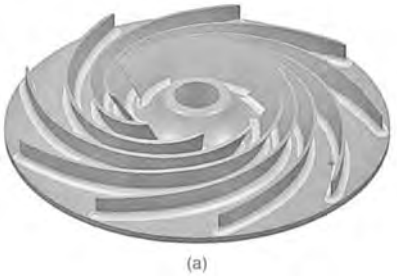


FIGURE 22A Typical 3-D solid model of radial-flow impeller (Courtesy of Carter Cryogenics)

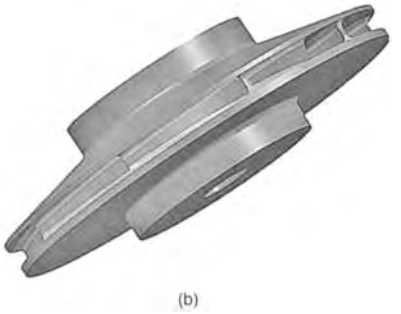


FIGURE 22B Typical 3-D solid model of radial-flow impeller (Courtesy of Carter Cryogenics)

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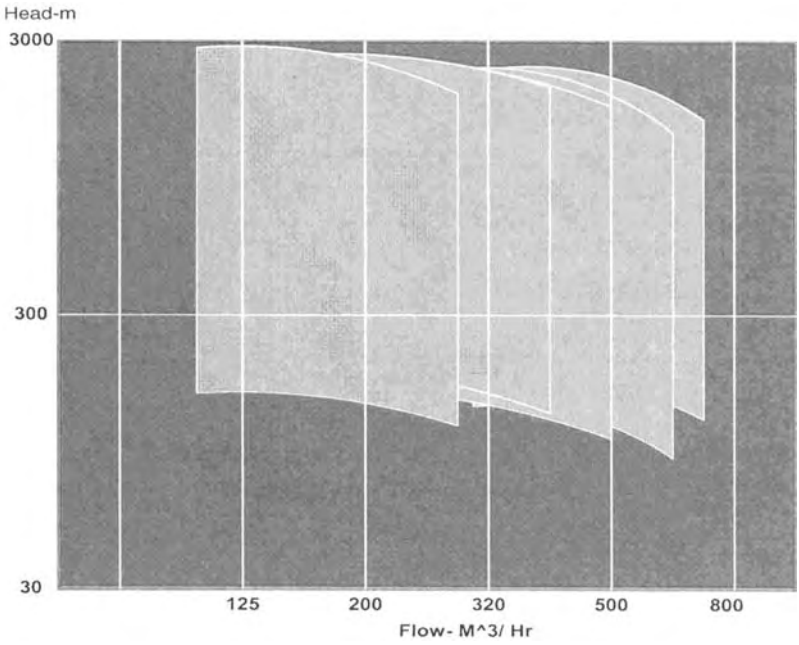


FIGURE 23 Typical envelope of pump performances obtained with different sets of impellers and pump speeds (Courtesy of Carter Cryogenics)

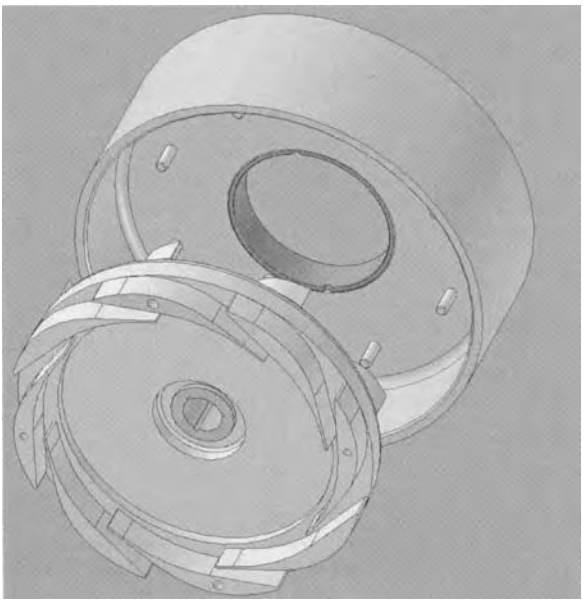


FIGURE 24 Typical radial-flow diffuser (Courtesy of Carter Cryogenics)

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FIGURE 25 Typical continuous crossover multistage diffuser (Courtesy of Carter Cryogenics)

Rolling-Element Bearings Cryogenic fluids have poor lubricating qualities, precluding the use of commercially available bearings. This has led to the development of durable rolling-element bearings that can accept thrust in either direction to handle spurious load reversals.

Pumps coupled to variable frequency drives (VFDs) can be supplied with ceramic ball bearings to prevent arcing due to eddy currents. The absence of heat in the bearings, combined with a thrust-equalizing balance drum, gives the bearings exceptional durability.

Thrust-Equalizing Balance Drum The thrust-equalizing balance drum balances thrust across the entire operating flow range, removing all axial bearing loads. The system is modular, and can be located either upstream or down stream of the motor. It keeps the motor at discharge pressure, preventing vapor from forming in the motor housing. This suppresses any hotspots and the possibility of liquid boiling, which could compromise cooling.

Terminal Header Submerged motor cryogenic pumps are provided with a terminal-header assembly to supply electrical power safely to the pump. The terminal-header assembly employs two hermetic seals, in series. Each seal is rated for maximum system pressure and minimum process temperature.

Extensive lab testing assures seal-assembly reliability and integrity. Testing includes:

- Seismic shock
- 24-h thermal exposure
- Cycling test: 100 thermal and pressure cycles over the full operating range
- 200-bar hydrostatic proof-pressure test with zero leakage tolerance
- Dielectric tests

Each terminal header is subject to acceptance tests for electrical-insulation resistance and proof pressure. Each seal is tested for integrity with a mass spectrometer to ensure that leakage is less than 1×10^{-6} sccm (standard cubic centimeters per minute). End-user connections are provided for purging, pressure sensing, or gas detection. Figure 26 shows the location of terminal headers for the pot-mounted and removable in-tank-mounted pumps.

Flexible Cryogenic Electrical Power Cables Electrical-power cables are typically configured as flexible nonasbestos insulated cables designed for use in cryogenic fluids at temperatures as low as –196°C. Each power cable is protected with a stainless steel braid sheath over its entire length and has both low (800 V) and medium voltage (6000 V) ratings. Figures 27 and 28 show samples of cryogenic cable.

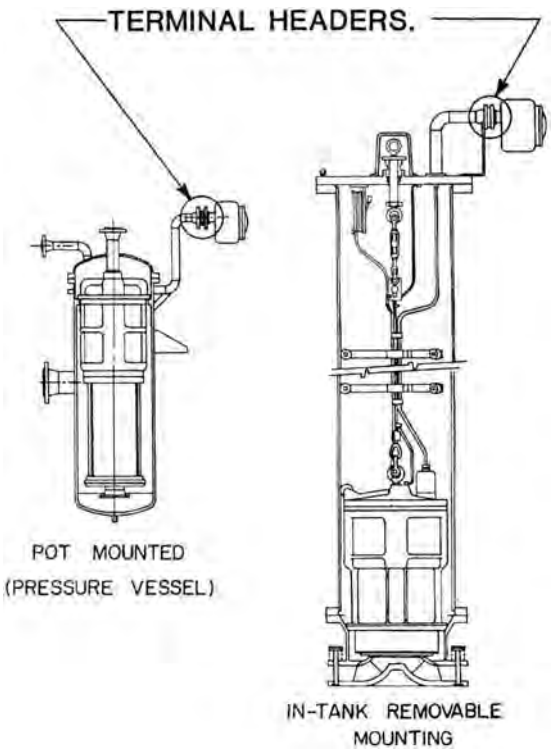


FIGURE 26 Terminal-header applications (Courtesy of Carter Cryogenics)

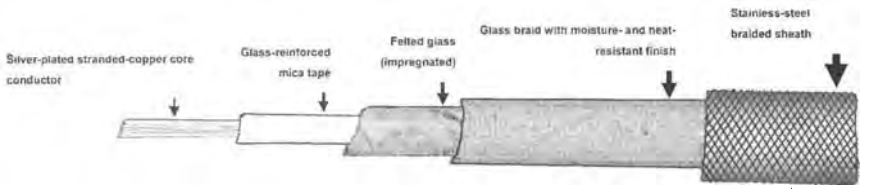


FIGURE 27 Typical layers of the cryogenic cable (Courtesy of Carter Cryogenics)



FIGURE 28 Typical stranded copper and the first three layers of insulation (Courtesy of Carter Cryogenics)

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CRYOGENIC TESTING

The manufacturer should provide full-scale testing of submerged motor cryogenic pumps. A typical facility for this purpose is illustrated in Figures 29 through 32. It has storage capacity for LNG, LPG, LN_2, and other fluids. During performance testing, all measurements are monitored by a high-speed data-acquisition system. This system graphically displays both raw and reduced data against predicted performance in real time. All instrumentation is tractable to recognized standards, with an overall measurement accuracy of ± 0.25%.

Figure 29 shows an ex-tank pump being lowered to the underground tank by crane for test. The cryogenic test loop during test is shown in Figure 30. After testing, the pump is moved by crane to the disassembly area for visual inspection as depicted in Figure 31. As seen in Figure 32, upon arriving at the disassembly area, the pump is heated up to the room temperature for teardown and visual inspection of any signs of abnormality.

Because of thermal contraction of the pump material in severe cryogenic conditions, the pump tested should be carefully examined after disassembly for any sign of rubbing by rotating parts such as inducer and impeller against nonrotating parts such as the inducer shroud or crossover discharge vane housings. Clearances of any mating parts at room temperature are not the same as those in the test tank at –162°C, but an unnecessarily larger gap will contribute to poor pump performance. Therefore, in the design stage, special care must be taken to specify the correct clearances for the mating parts of the pumps, which have many variations in size, flow rate, speed, and head-rise requirements.



FIGURE 29 A secondary cryogenic pump being lowered to the underground tank for testing (Courtesy of Carter Cryogenics)

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FIGURE 30 Cryogenic test loop during cryogenic pump test (Courtesy of Carter Cryogenics)

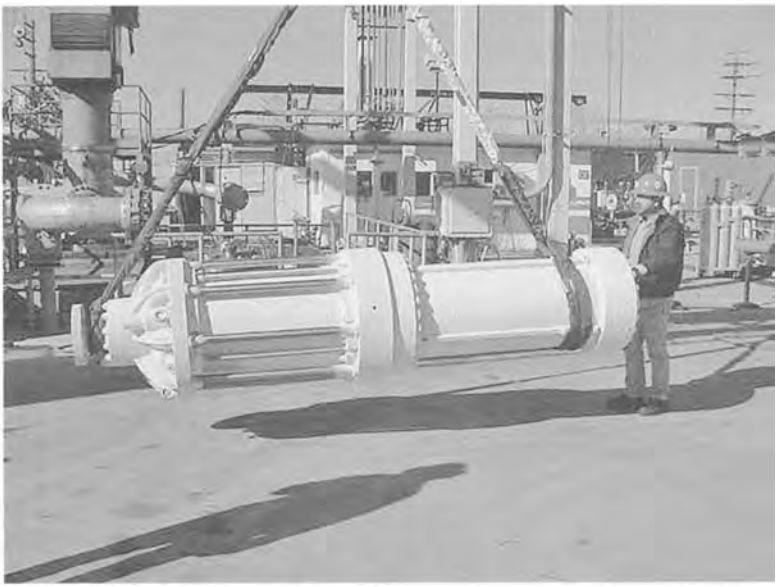


FIGURE 31 After testing, the cryogenic pump is transferred to undergo teardown inspection (Courtesy of Carter Cryogenics)

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FIGURE 32 After testing, the pump is warmed to room temperature for teardown inspection (Courtesy of Carter Cryogenics)

OTHER TECHNICAL CONSIDERATIONS

Methodology for Seismic Qualification of Pumps

REQUIREMENTS

1. Equipment should keep its physical integrity, and earthquakes should not produce any unrecoverable deformation that could prevent proper operation. This requirement extends to the mounting brackets and the anchorage or fastenings to the support structure.
2. Earthquakes should not produce a loss of the pressure boundary that could lead to spillage of the LNG.

SEISMIC ASSESSMENT The seismic response of each pump should be combined with the effects of normal operating loads and compared with the allowable loads, stresses, and deformations.

FREQUENCY ANALYSIS Since the seismic load depends on natural frequencies, the first step of the methodology is to compute natural frequencies. In all cases, fluid-added masses should be taken into account. (See Section 2.1.4.)

SEISMIC RESPONSE Once the natural frequencies and mode shapes are known, the seismic response of the pumps should be computed. Three response spectra (two horizontal and one vertical) should be introduced for each analysis.

The response spectrum method gives the expected maximum values of the response during the earthquakes. Results include maximum section forces and moments, maximum deflections, maximum values of reaction forces, and maximum forces at connections or kinematical constraints.

The response spectrum method does not take into account nozzle loads. The effects of nozzle loads are added to the results of the response spectrum analyses by means of conventional static analysis.

In the case of the in-tank pumps, the response spectrum method does not consider the hydrodynamic forces on the column derived from the seismically induced movement of the whole LNG tank. These forces should be computed from maximum hydrodynamic pressures

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within the tank and should be introduced as static forces over the outer surface of the column.

RESPONSE TO NORMAL OPERATING LOADS The effect of normal operating loads should be added to the seismic response in order to assess the structural integrity and operability of the equipment. For computing the response to normal operating loads, the same finite element methods already used for frequency and response spectrum analysis should be employed. The response should be obtained by conventional static analyses.

Methodology With respect to the seismic qualification of submerged motor cryogenic pumps it should be noted that these pumps are mechanical devices that are inherently rugged from a seismic point of view. Operational loads are commonly more severe than seismic loads and, as a consequence, it is very rare that a well-designed pump will suffer from seismic damage if it is correctly attached to the supporting structure. Documented seismic experience of actual pumps in real earthquakes shows that only in a very few cases pumps were taken out of operation by the earthquake; and, in these cases, the failure was associated with the failure of the supporting pad or a rupture in the anchorage pipes.

With this background in mind, it seems reasonable to focus the qualification effort "outside" the pump-motor assembly, looking at the path of the inertial loads from the pump-motor assembly all the way to the anchorage to the supporting structure.

Inside the pump-motor assembly it should be sufficient to look at the deformation of the shaft due to earthquake inertial loads and at the additional forces on the bearings caused by the quake. However, in all probability, the results of the analysis will show that shaft deformation and bearing forces caused by the specified earthquakes are negligible.

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SECTION 12.17
PORTABLE TRANSFER OF HAZARDOUS LIQUIDS

P. A. NOLTE

Transfer systems for hazardous chemicals must not only take into account the best means to transport the chemical based on the chemicals fluid characteristics, but must also pay particular attention to both safety and environmental issues. The chemical can be something as common as gasoline, or a very poisonous restricted-use pesticide such as Paraquat. When the requirement of portability is added to the chemical handling system, design parameters must now take into account variables such as power source, environmental conditions, and chemical restraints. Several power sources have been incorporated through out the years for the portable transferring of hazardous chemicals. One of the earliest is simple hand power. Others consist of gasoline powered, ground-driven, compressed air, and battery operated pumping systems.

Major markets that require the portable transfer of chemicals include petroleum, agricultural, construction, pest control, and lawn care. Systems must be adaptable to changing environmental regulations, chemical container designs, and application requirements.

AGRICULTURAL REQUIREMENTS

The need for portable chemical transfer systems in the agricultural market arises from farmers working fields with large equipment, usually several miles away from local chemical dealerships and storage facilities. It is usually more economical to bring the chemical to the piece of application equipment while it is in the field, rather than transporting the application equipment back and forth from the field. Chemical is delivered to the field in several different ways, depending on the volume required. Tanks can vary in size from 120 gallon (450 liter) herbicide tanks that can be transported by pickup trucks, to large 1000 gallon (3800 liter) tanks that are delivered by trailers or large custom trucks. With the growth of the custom applicator business, the sophisticated chemical transfer rigs have

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focused on speed of transfer, flexibility to supply the chemical required, and the ability to cover large areas. Some large aerial applicator rigs have the ability to allow spray helicopters to land directly on the rig, thus allowing quick, portable, and convenient transfer of spray solution from tanks into the helicopter.

The majority of hazardous chemical transferring in the agricultural market deals with the usage of either fertilizers or pesticides. A pesticide is a group of chemicals that consist of insecticides, herbicides, or fungicides. Insecticides focus on the control of insects, herbicides are chemicals used to control of grasses and weeds, and fungicides are chemicals used to control crop damaging bacteria and/or fungus. During the last 20 years, the usage of chemicals in the agricultural market has been greatly affected by growing environmental concerns and worker safety. Environmental concerns have caused chemical formulations to use less aromatic solvents. As an alternative, many chemicals are now viscous water-based suspensions that generate pumping challenges. Chemical systems have also seen new regulations that control items such as chemical spillage from fittings, tank sizes, and design and testing parameters. This, in turn, has affected the needs and requirements of the chemical handling systems.

Fertilizers present a transferring challenge due to the large volumes that are used, and the corrosive nature of most solutions. Two types of power modes are commonly used in the market for the transferring of fertilizers: gasoline-powered centrifugal pumps and ground-driven positive displacement pumps. Gasoline-powered centrifugal pumps used are usually lightweight aluminum or plastic units with 2 in (50 mm) ports. The pumps are coupled to a 3 to 5 hp (2 to 4 kW) gasoline engine and are capable of transferring rates up to 150 gpm (568 l/min). These high flowrate gasoline powered pumps are used to transfer fertilizer and water from large 1000 to 2000 gallon (3785 to 7570 liter) transport tanks into the application equipment's onboard storage tanks, 300 to 500 gallons (1135 to 1890 liters) in size. Ground driven pumps are mounted on the implement equipment such as planters and cultivators. The pumps are used to transfer and meter the fertilizer through a network of tubing that runs to the backside of cutting blades. This allows the knifing of the fertilizer into the ground. Ground driven pumps are popular for this type of application because the volume dispensed is directly proportional to the ground speed of the implement. Popular pumps that have been adapted for ground driven applications are positive displacement piston, diaphragm or multi-tube peristaltic pumps.

Pesticide transfer is now rigorously regulated because of environmental concerns and an increased emphasis on decreasing worker exposure to the hazardous formulation of a large percentage of these chemicals. The Environmental Protection Agency (EPA) has guided the agricultural chemical industry away from using 2.5 gallon (9.5 liter) throwaway containers to larger, returnable, reusable, chemical-specific containers. These "Mini Bulk" tanks range in size from 15 to 200 gallons (50 to 750 liters), depending on the chemical usage rate. This change in the way farm chemicals are packaged to the end user has generated new requirements for portable pumps. Current chemical delivery systems have evolved around the usage of 12-volt power systems. Due to regulator restrictions that forbid the cross contamination of agricultural chemicals, the delivery systems are usually designed for a particular chemical or container. An adequate pumping rate for most 12-volt portable system is a rate of 5 to 10 gpm (18 to 38 l/min). Agricultural chemicals come in a wide range of viscosities, varying from 1 to 500 centipoise. Some are true liquids, whereas most are a suspension or mix. Typically, positive displacement pumps of the diaphragm, piston, or gear type are used for portable chemical transferring. Gear pumps are usual limited to the low viscosity chemicals with no suspended solids. Piston and diaphragm pumps are best suited to handle the higher viscous chemicals, many of which have suspended solids. Chemicals with solids usually will require tank circulation, and this must be designed into the transfer system.

Figure 1 shows a pumping system on a 120 gallon (450 liter) tank that consists of a pump, 12-volt motor, a digital meter system, and no-drip hose coupling. The pump is permanently mounted to the shipping container at the time of tank manufacture. The motor, meter, and hose are removable items that are normally supplied by the chemical dealer. This allows the drive unit and metering device to be transferred from tank to tank, after a tank is emptied. The pump is a 6-chambered plastic diaphragm pump and is mounted to the tank by means of a bolting flange. The mounting flange also contains a built-in vent to

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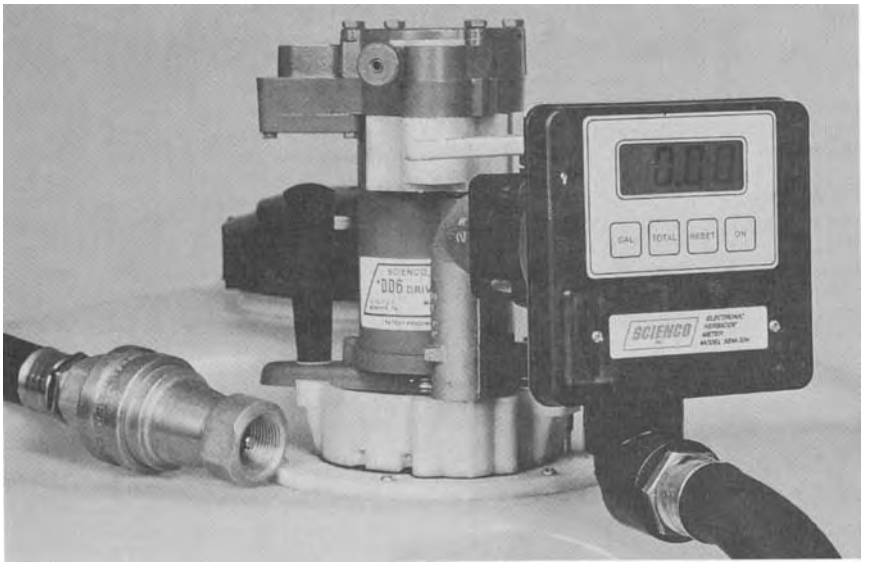


FIGURE 1 Portable 12-volt chemical pump (Flowserve Corporation)

allow air in during fluid pumping. An internal bypass valve in the pump is connected to the inside of the chemical tank, allowing chemical circulation within the tank during bypassing. This pump also allows manual opening of the bypass valve to allow drainage of chemical from the meter and hose before removal. When the pump is mounted to the tank during shipping, the complete system must meet and pass specific Department of Transportation (DOT) tests. These tests are specified in the United States by DOT's HM181 specifications. Depending on the size of the container and the hazardousness of the chemical, these tests can require drop tests of several feet at 0°F (minus 17.8°C), vibration, stacking, and pressure tests.

Figure 2 shows a typical 120 gallon (450 liter) chemical "Mini-Bulk" tank with the pump connected to the tank by a coupler (not permanently attached). The coupler system provides a means for attaching and detaching the pump and the tank, a means for venting, and a dip tube assembly allowing suction from the bottom of the tank. The pump assembly is coupled through the tank coupling for chemical dispensing and is then removed for transporting. The pump shown is a stainless steel, 12-volt diaphragm pump, with a stainless steel flow meter. The chemical dealer fills the mini-bulk tank with chemical. The grower purchases the chemical, uses the pump to dispense chemical into an application rig, and then returns the tank empty to the dealer.

PETROLEUM AND CONSTRUCTION REQUIREMENTS

One of the original areas requiring the use of portable pumps is the transfer of gasoline and diesel fuel. The need to refuel and maintain equipment in remote locations has generated a large need for portable transfer equipment. Road and building construction, mining, and farming all involve heavy equipment that must be refueled and maintained on a regular schedule. One of the earliest and simplest methods of portable transferring was with hand pumps. Hand pumps are positive displacement pumps of piston, vane, or diaphragm-type design. Of course, the power supply is very limited with hand pumps, and they are typically used only on thin liquids and small jobs. On large pieces of equipment,

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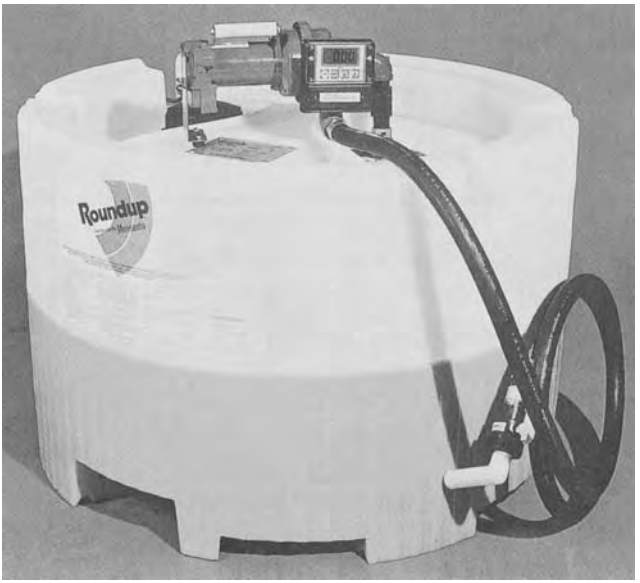


FIGURE 2 120 gallon (454 l) "Mini-Bulk" Container and pump (Flowserve Corporation)

the need is to transfer large volumes of fluids, such as water, antifreeze, diesel, gear lube, grease, and transmission fluid.

Figure 3 shows a portable maintenance vehicle with air powered piston pumps. In the construction industry, compressed air is a common power source for transferring product. A typical truck consists of a gasoline-powered air compressor, several tanks of petroleum products plus water and antifreeze. The size and type of pump will normally depend on the size of the equipment requiring servicing and the fluid being pumped. For thin fluids, such as diesel oil or gasoline, high volume, low pressure ratio pumps (approaching 1:1) will be used. Occasionally, on units that have the option of a power take-off (PTO), a high volume centrifugal-type pump will be used. For the oils and lubricants, higher pressure ratio pumps (3:1 to 5:1) are required because of higher transfer pressure requirements. To transfer grease, high pressure ratio pumps—to 20:1—are required. The pumps are normally coupled with a hose reel, 50ft (15 m) of hose, and a control nozzle.

LAWN AND PEST CONTROL

Lawn and pest control companies require a portable means of transferring chemicals in order to provide treatments. Liquids are used to treat yard infestations; in addition, the majority of homes and buildings are treated for termites during construction. It is common for lawn care companies to use PTO-driven centrifugal pumps capable of delivering 10 to 15 gpm (38 to 56 l/min) through a 100 ft (30 m) hose. Specialty requirements, such as tree spraying, will require a higher pressure system, normally a gasoline-powered piston pump. For insecticide treatment, the power source is commonly either electricity or gasoline. Typical pumps are roller pumps, capable of approximately 4 gpm (15 l/min) at 50 lb/in^2 (3.45 bar).

These industries have taken major steps to limit chemical exposure to the general public and to their operators. Efforts have been made by many to eliminate the need of transporting large 100-plus gallon (378 liter) tanks of mixed chemicals. One method that allows

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FIGURE 3 Flatbed service truck (Aro Fluid Products)

the tanks to hold and transport only clean water is to inject the chemical into the discharge only at time of use. The chemical is kept in small, 1 to 5 gallon (3.5 to 20 liter) containers with 12-volt metering pumps attached, which allow direct injection into the clean water line. Other systems use small proportioning pumps. These units are two-stage piston pumps powered by clean water. The smaller piston injects a set volume of chemical into the water supply for each stroke of the larger water piston.

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SECTION 12.18
AEROSPACE

12.18.1
AIRCRAFT FUEL PUMPS

J. E. CYGNOR
Y. J. CHU
D. PALOMBO

The operating characteristics and reliability of aircraft fuel pumps are absolutely critical with respect to the performance and the safety of flight of today's gas turbine-powered commercial and military aircraft. With a large commercial airliner carrying upwards of 50,000 gal (189,000 L) of fuel on a long range flight, making fuel the single heaviest load the aircraft must handle, the importance of intelligent and safe processing of this fuel load is clearly evident.

The aircraft fuel pump system is divided into two separate but related systems: the airframe fuel pump system and the engine fuel pump system. Two systems are required because of the significantly different functional requirements of each system. The airframe system is a low pressure system that operates continuously at low values of inlet NPSH and utilizes booster pumps located throughout the airframe fuel system to supply pressurized vapor free fuel to the engine fuel system. The engine fuel system is a high-pressure system that further pressurizes the fuel for delivery to the engine combustor fuel nozzles and other engine systems. The requirements of these systems and the pumps in them are covered by a wide range of government and industry specifications. Minimally, these define the performance, fuel types, operating and ambient conditions, drive source, reliability, life, weight, installation, quality, materials, and test requirements to which the fuel pump must conform.

The detailed nature of these specifications further emphasizes the critical nature of aircraft fuel pumps. The fuel pump designer's task is to provide fuel pumps that meet these specifications with the lowest weight, best efficiency, and at a competitive cost. Both centrifugal and positive displacement types of pumps are used in performing the necessary fuel pumping functions on essentially all large gas turbine-powered aircraft.

Because of the specific installation and performance requirements for airframe and engine fuel pumps, except the small piston-powered engine in general aviation, each pump is an individual and custom design for each specific application. As such, there are no catalogue type standard pump configurations or designs for medium or large aircraft applications.

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OVERALL AIRCRAFT FUEL SYSTEM

Figure 1 presents a basic schematic of the fuel pump components in the overall aircraft fuel system. The division between the airframe and engine systems is shown.

The primary function of the airframe fuel system is to provide a pressurized fuel feed to the engine fuel pump system under all operating conditions. In the event of the emergency conditions of nonoperating or failed boost pumps, a bypass located either in the airframe plumbing or the boost pump allows the engine pump system to receive sufficient fuel to provide takeoff and climb thrust. Other functions required of the airframe fuel pump system are as follows:

- Provides fuel transfer between various fuel tanks to maintain continuous engine feed and to adjust the center of gravity of the aircraft.

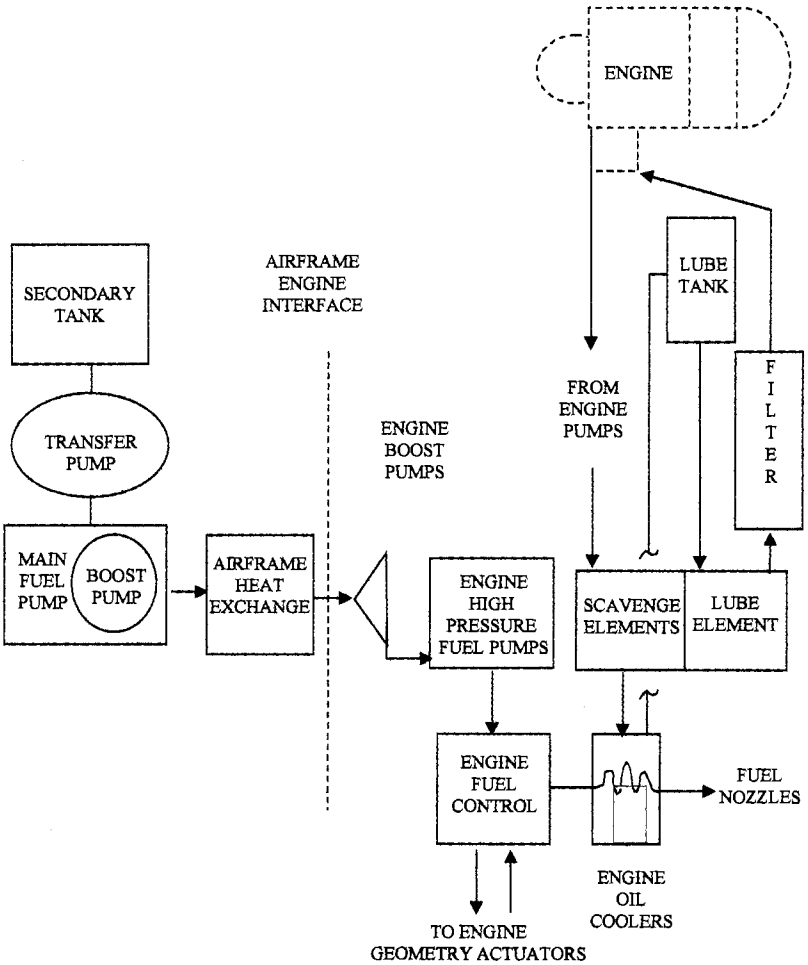


FIGURE 1 Basic schematic of fuel system

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- Provides fuel jettison in flight in the event it is necessary to quickly reduce the aircraft weight.
- Assists in the defueling of the aircraft on the ground.

The primary function of the engine fuel system is to receive the fuel from the airframe fuel system, under normally pressurized conditions or emergency conditions, and provide pressurized fuel to the engine. This fuel is used for burning in the combustors and for powering actuators that control the engine's variable geometry.

Because fuel is the only consumable fluid carried by the aircraft, it represents the sole heat sink on board the aircraft. (For extreme conditions, some aircraft have carried an additional disposable heat sink fluid.) The on-board fuel and the air ingested by the engines or scoops provide all of the cooling necessary for the proper functioning of the various airframe and engine systems. The impact of this on the fuel pumps is a significant increase in the temperature of the fuel on which the pumps must operate.

GENERAL DISCUSSION OF AIRFRAME FUEL PUMPS

The primary type of pump used for providing the fuel boost and fuel transfer functions on an airframe is a centrifugal pump element driven directly by a fuel-flooded electric motor. Positive displacement elements are rarely used and alternate drive means such as hydraulic motors and air or fuel driven turbines have only been used on a selected few military aircraft. On small general aviation aircraft, jet pumps with the motive flow provided by the engine fuel pump are popular.

Helicopters commonly incorporate a special type of fuel system called a "suction feed" fuel system. Because a helicopter is altitude-limited, it is not necessary to provide a pressurized engine fuel inlet condition. In fact, for safety considerations, it is desirable not to pressurize the fuel lines between the fuel tanks, located in the lower sections of the airframe, and the engines, which are located in the upper sections of the airframe. For these applications, the engine fuel pumps are designed to continuously operate with a fuel inlet pressure lower than the fuel tank pressure by the height of the fuel lift between the fuel tank and the engine and the incurred fuel line pressure losses.

AIRFRAME FUEL PUMP DESIGN

The design of an airframe fuel pump is generally characterized by the configuration in which it is mounted in the airplane. The two common airframe pump types are classified in this way; namely,

- In-tank mounted
- Spar mounted

Each type of pump has advantages as well as disadvantages. Although they are all motor driven, each fits different applications and different types of aircraft. To minimize weight, all are made of aluminum except for highly stressed parts such as shafts and fasteners.

The in-tank type of pump is one of the most widely used pumps in the aerospace industry and is normally a cartridge canister pump as shown in Figure 2. Since the pump is installed at the bottom of the tank, it may take advantage of the fuel head inside the fuel tank to help its performance at altitude. (See performance discussion further on.) Therefore, it is often used as a boost pump to feed the engine or in-tank transfer pump for center of gravity (CG) balance. The flow rate or capacity of the in-tank-mounted pump varies greatly from tens of gallons per minute (or m^3/h) for boost or transfer to an order of magnitude more for jettison application. Pressure rises range from 10 to 50 psi (10.69 to 3.45 bar). Because it is structurally simple and does not require a priming element, it is a

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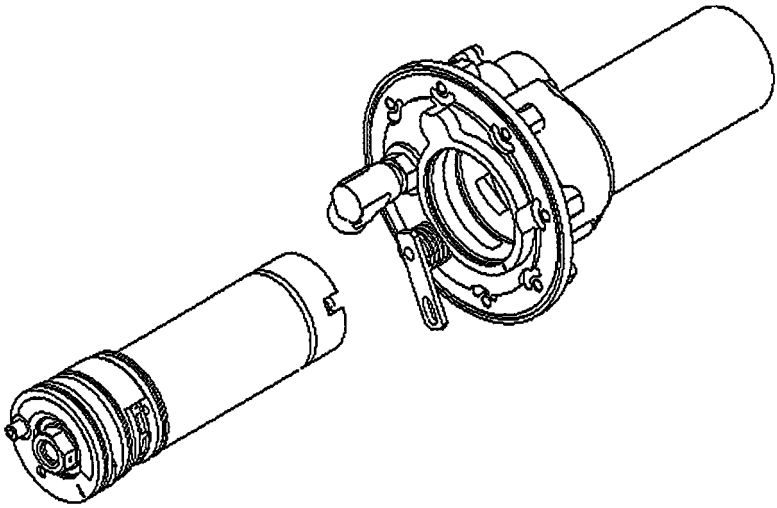


FIGURE 2 Typical in-tank pump (Courtesy of Parker Hannifin Corporation)

preferred choice for the fuel system engineer. However, there may be three distinctive drawbacks of this type of pump:

- It may weaken the wing structure, as most of the in-tank type of pumps are of the cartridge-canister style, and an access hole is required for the pump cartridge at the bottom of the wing.
- It may not achieve the pump-down level required, because the cartridge-canister pump requires some floor clearance to allow the fuel to flow into the impeller at the pump inlet, therefore, the fuel quantity below the pump impeller eye may become the "unpumpable volume." To minimize this volume, an ejector or jet pump (Chap. 7) may be deployed at the bottom of the tank.
- The design phase requires close scrutiny to ensure safety, because the entire pump including the motor is inside the fuel tank and therefore must be explosion-proof. (See discussion below on safety requirements.) This concern is less challenging if the pump is fully submerged in the fuel, because the fuel serves to cool the pump and to quench any possible ignition source.

The spar-mounted pump has flow rates and pressure rises that are similar to those of the in-tank pump, and it offers some attractive features which may compensate for the shortcomings of the in-tank pump. As shown in Figure 3, it consists of three elements: the cartridge, mounting plate, and snorkel. The electric motor, which is included in the pump cartridge, is located outside the fuel tank in a dry area; it is inherently better than the in-tank pump in terms of safety. The spar itself is not subjected to the wing deflection. Furthermore, the inlet snorkel of the pump can be deployed to the lowest spot of the fuel tank to achieve the optimum pump-down level. Since the fuel must be lifted to the main impeller, the altitude performance of this type of pump may not be as good as the in-tank type.

MOTOR DRIVE OF AIRFRAME FUEL PUMP

The drive motor of the pump deserves some special attention in this section, because of the rapid change in motor technology as applied to aircraft fuel pumps. As shown in Figure 4, motors for aircraft fuel pumps have evolved into a wide range of types.

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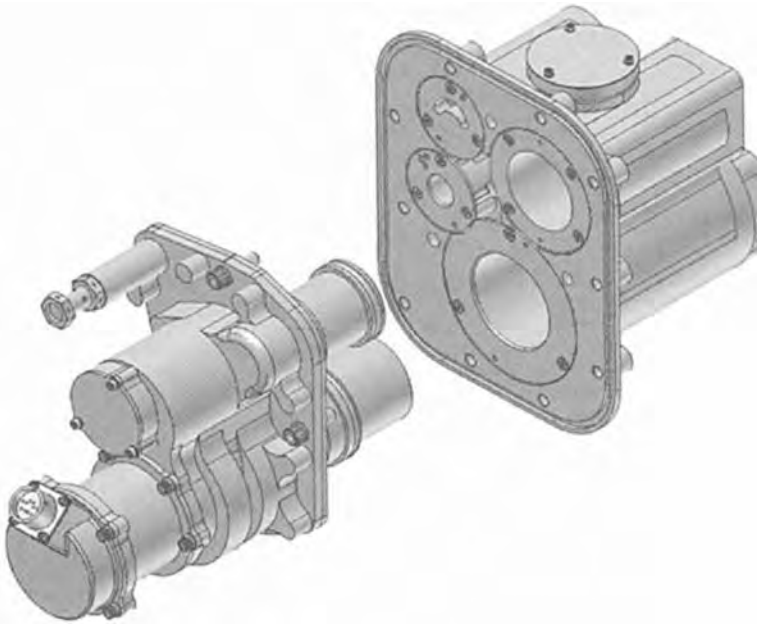


FIGURE 3 Spar-mounted pump with preferred snorkel shown

Prior to 1980, almost all motor-driven pumps in aircraft fuel applications were either the low voltage brush-type motor or constant-frequency ac induction motor. However, the variable-frequency (VF) pump started to become popular in 1990, as did also the electronic-drive motor-driven pump. Although the 28-V brush-type dc pump and 400-Hz ac induction motor-type pump still dominate the field, the growing number of electronic-drive

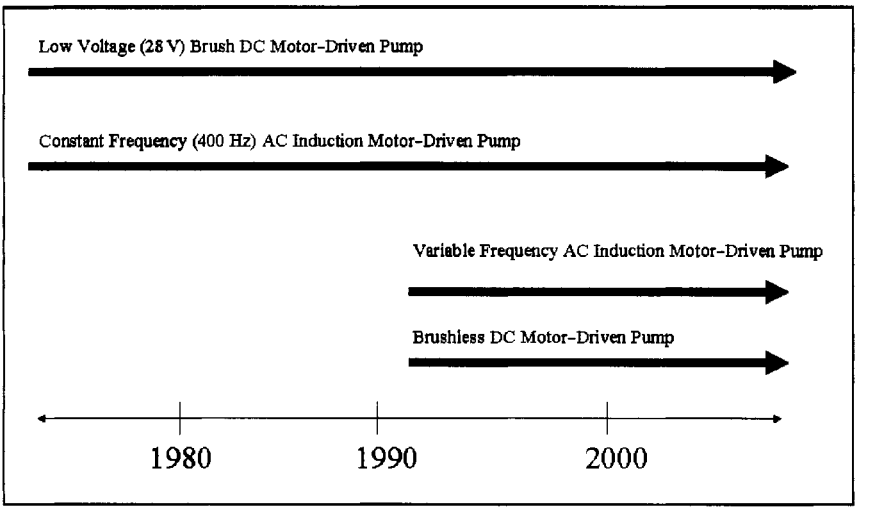


FIGURE 4 Evolution of airframe motor-driven fuel pump

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motor-driven pumps (brushless dc motor-driven pump) will make it a major player in the future. In fact, the brushless dc motor-driven pump has been used in almost every airplane since 2005. The description of each type of motor drive for airframe fuel pumps is shown below:

Low-Voltage Brush-Type Motor-Driven Pump This is one of the most popular types of fuel pump used in the aircraft industry. Most of these pumps are used in the piston engine-powered aircraft of General Aviation. More than half a million pumps of this type have been produced in the industry. These pumps are powered by either battery or low-voltage power bus. In most of the applications, there is no speed control on the motor; therefore, the pump speed varies greatly with the load of the pump, and so do the flow rate and pressure rise of the pump. In order to extend the life of the motor, the motors used in this type of pump are fuel flooded. This not only eliminates the life-limited dynamic seal between motor and pump but also provides constant cooling and lubrication flow for motor and bearings. Due to weight and power constraints, this type of pump is seldom used at a power level above 1 hp (746 W). For smaller aircraft, it is widely used for the boost or feed pumps. The strengths and weaknesses of this pump are as follows:

- Strengths
  - Reliable
  - Inexpensive
  - Simple power interface
- Weaknesses
  - Suitable only for low power application
  - Limited life expectancy
  - Limited dry run capability
  - High in-rush or starting current
  - Performance varies due to the load change

Although this type of pump is quite simple, its performance is very sensitive to load variation. As shown in Figure 5, a small change in the pump load (expressed here as torque) may cause some change in the pump speed; consequently, the pump performance may vary. Since it contains brushes for commutation; it is not recommended for prolonged dry run. In some applications, additional resettable thermal protection devices may be required to accommodate this requirement. Because there is a limited life expectancy for brushes in the motor, the pump needs to be overhauled after a limited number of hours of operation. The starting current of the pump may be undesirable for electrical systems. It is not uncommon to see more than 600% of normal running current at pump start-up for a very short period of time. Similarly, if the pump encounters a locked-rotor condition, the high current could destroy the motor unless the thermal-protection device of the motor is tripped.

Constant Frequency (CF) AC Induction Motor-Driven Pump This type of pump is primarily driven by a 400-Hz, 115-V ac (line to neutral), or 200-V ac (line to line) motor. Both three-phase and single-phase motors of this category are widely used in aircraft applications. However, the three-phase one is very popular, being used in almost every large transport—both commercial and military. The advantages of using a three-phase motor are compactness, high power-to-weight ratio, and superior reliability. A properly designed three-phase motor and mating impeller can pump up to hundreds of gpm in a package that is a few pounds in addition to overall motor weight. (m^3/h = 0.227 gpm.)

Unlike the brush-type motor, there is no contact between the stator and rotor; hence, the life of the induction type motor is extremely high and commonly reaches hundreds of thousands of hours before overhaul. As shown in Figure 6, when the load (in terms of torque) varies from T_1 to T_2, the speed variation is small. Therefore, the performance of the pump is very predictable. The features of this type of motor are summarized as follows:

- *Simple and robust.* No extra parts are required besides the rotor and stator.
- *Stable operating speed over wide torque range.* Load of the pump has little effect on the pump speed.

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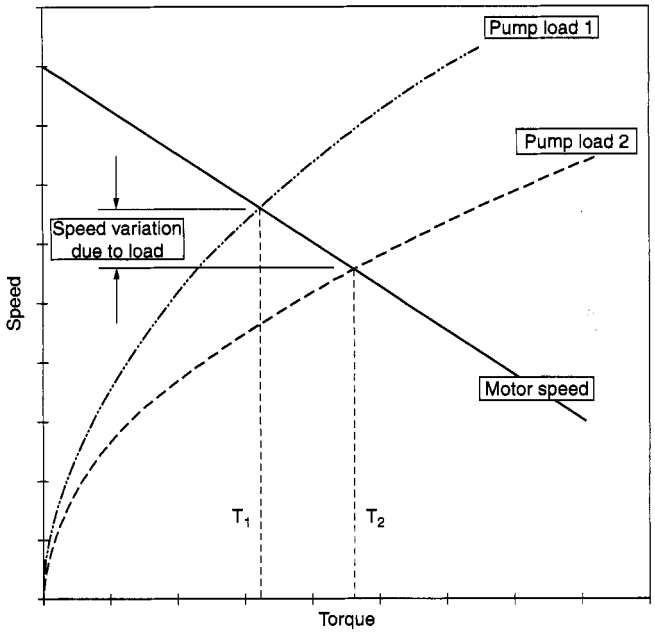


FIGURE 5 DC motor-driven pump load

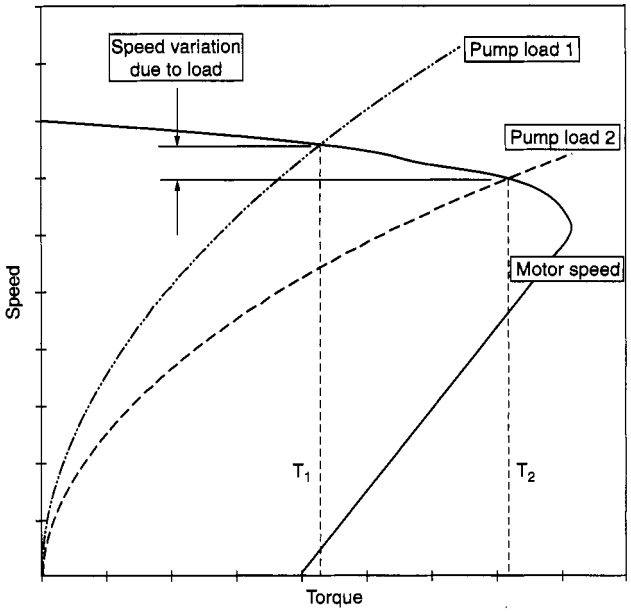


FIGURE 6 Constant frequency ac induction motor-driven pump load

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- *Dry runnable.* The properly designed ac induction motor may be capable of running dry for an extended period without overheating of the motor.
- *High efficiency.* Peak efficiency is around 80 to 85%; and pumps are commonly designed to operate at both this peak and at peak power factor, thereby consuming minimum energy and current for maximum hydraulic output.
- *Low starting or in-rush current.* Unlike the brush-type motor, the starting current of the ac induction motor is seldom over 350% of its steady-state operating current.

Variable Frequency (VF) AC Induction Motor-Driven Pump The increasingly popular VF motor-driven pump is basically the same as the CF-type pump; however, it allows the elimination of the constant-speed drive from the aircraft electrical system. In the VF power system, the pump power supply is directly connected to the aircraft generator without the constant speed drive in between. The frequency of the power supply to the pump motor is directly proportional to the engine speed. Normally, the VF motor-driven pump operates with the power frequency from 300 to 800 or 900 Hz. However, since the pump power consumption is proportional to the cube of the speed, such a large range of motor frequency results in a larger range of the pump load (in terms of power.) Consequently, an oversized motor may be required to cope with the large change of the load if the speed of the motor is not allowed to slip substantially below synchronous speed at high input frequency. As a result, there are two types of VF-powered pump in the aircraft industry:

- The high-slip ac induction motor pump
- The electronic-drive motor-driven pump (brushless dc motor-driven pump with power factor correction module)

Balancing the pump load (torque) without significantly increasing motor weight and the pump power at high frequency becomes an interesting trade-off. As shown in Figure 7, for a given family of performance curves of VF motors, the pump speed can vary greatly depending upon how and where both the pump load curves intersect with each other. One way to deal this wide frequency range in the VF motor without penalizing the motor weight is to use a "weak" motor and allow the motor speed to slip substantially from its synchronous speed. By doing so, the pump load will be allowed to increase only to a limited extent. It is also not unusual to see the speed of the motor slip in the range of 30 to 40% when it is

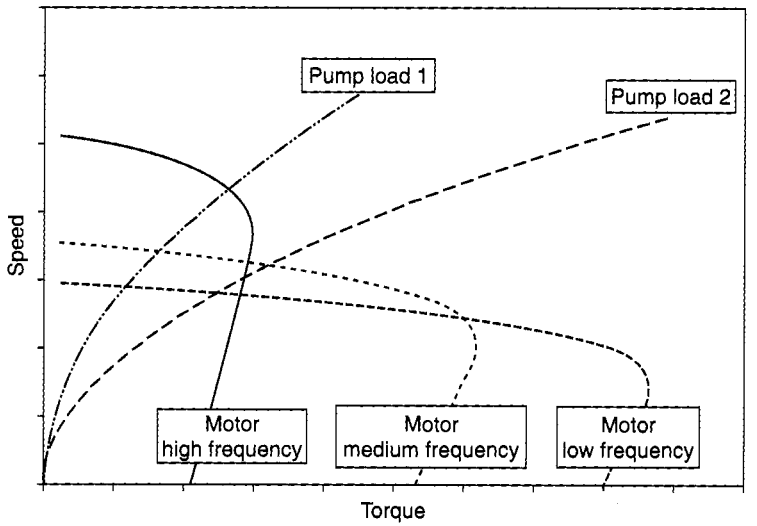


FIGURE 7 Variable frequency ac induction motor-driven pump load

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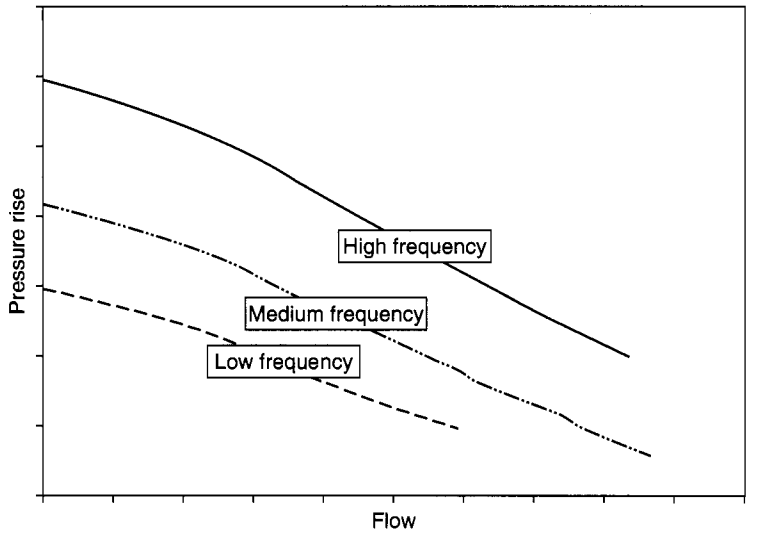


FIGURE 8 Performance curves of variable frequency ac induction motor-driven pump load 1

operated at high end of frequency range (800 to 900 Hz). If the pump load is either small enough or the motor is robust enough at high frequency, then the pump performance will be as shown in Figure 8. However, if the motor is weak and the slip in motor speed is excessive at high frequency—such as illustrated by the curve *Pump Load 2* in Figure 7–then pump performance will behave as shown in Figure 9. From this discussion emerge the inherent weaknesses of the VF motor-driven pump with a high-slip induction motor:

- *Poor power factor and motor efficiency through the entire frequency range.* This situation becomes extreme when the pump operates at the low end of the frequency range; for

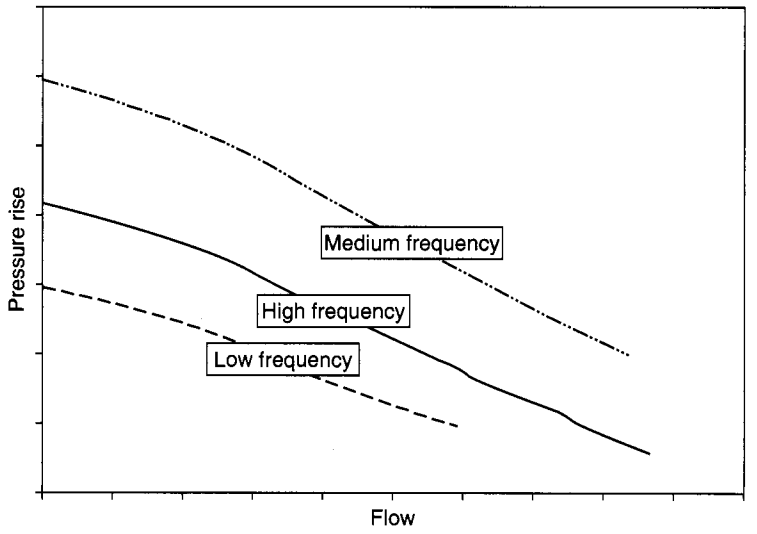


FIGURE 9 Performance curves of variable frequency ac induction motor-driven pump load 1

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example, power factor can be as low as 30 to 40%. Sometimes, it only reaches 30 to 40% in power factor at 360 Hz if the full frequency range is from 360 to 900 Hz.

- *Overheating of dry run at low frequency.* When operated at low frequency, a VF motor is saturated; so that, regardless of the load on the motor, it consumes the same amount of current even under dry run conditions. Consequently, if the pump encounters a dry run condition at low frequency, the motor may easily become overheated.
- *High locked-rotor current at low frequency.* The locked-rotor current of a VF motor is inversely proportional to the square of the motor frequency. Therefore, the attendant wide range of locked-rotor current from low frequency to high frequency becomes critical for the electrical system.
- *Low starting torque at high frequency.* The starting torque of a VF motor is inversely proportional to its frequency. This may become an issue if the pump has a high static load. If the starting torque of the motor is not capable of overcoming the static load of the pump, the motor may stall.

Brushless DC Motor-Driven Pump The brushless dc motor has been widely used in other industries for several decades; and, due to advancement in the power electronics of the motor controller as well as a sharp decline in controller cost, it has more recently gained momentum in application to aircraft fuel pumps. It is distinguished from the other pump types by the following unique features:

- It can vary the pump speed while remaining efficient; hence, the output of the pump can be "on demand" to precisely match the aircraft mission requirement. Therefore, it may reduce the power consumption of the pump.
- The brushless dc motor is approximately 5 to 15% more efficient than either the constant frequency ac induction motor or the brush-type dc motor. Moreover, when the pump runs dry, it consumes only a fraction of power consumed by the brush dc motor or ac induction motor.
- It can incorporate some intelligence in the motor controller such as over temperature protection, over current protection, failure reporting/isolation, health diagnosis and prognostic functions. Therefore, it may save some maintenance cost.
- It can provide a "soft start" of the pump and so reduce the in-rush current at pump start-up, or it may offer faster response to the control signal.
- It can interface with other components in the fuel system such as pressure switches or pressure transducers to supplement or replace the functions of those components and so simplify the fuel system.

On the other hand, although the stator and rotor of the brushless dc motor may be as robust as those of the ac motor, there is a weakness of the brushless dc motor-driven pump. This is the loss of reliability of the electronic components and solder joints on the circuit board of the motor controller if these are exposed to high temperature. Most industrial grade electronic components are rated at 40 to 85°C. The cost associated with enhancing the reliability and ruling out infant mortality of the electronic components remains an issue in the application of brushless dc motor-driven aircraft fuel pumps.

CONTROL OF BRUSHLESS DC MOTOR-DRIVEN FUEL PUMPS

Due to its unique features and the importance of the brushless dc motor in aerospace application, the electronic motor controller deserves some special attention. Generally, the electronics can be split into two sections, the input power filter and the motor drive.

The diagram in Figure 10 depicts the major components in the drive electronics. The input filter would be everything to the left of the dc link capacitor. (Current research may lead to the elimination of the dc link—in an integrated ac-to-ac converter or "matrix converter.")

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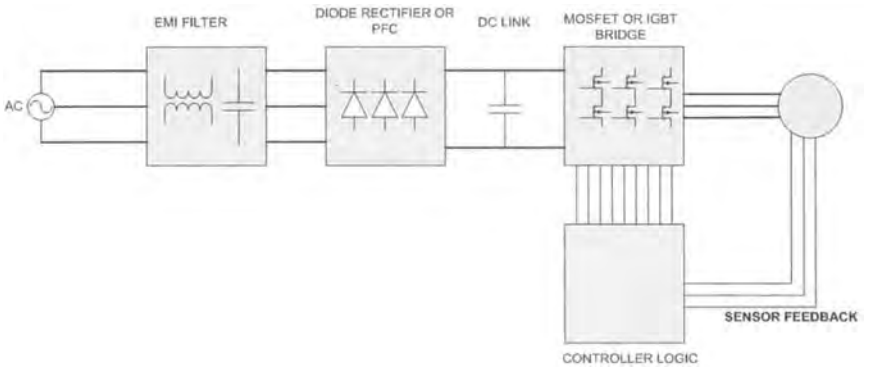


FIGURE 10 Diagram of a basic brushless motor drive

Input Power Filter At a minimum, the input power filter must prevent conducted energy from the motor drive from polluting the input power source. The two sources of emissions are called common mode (CM) and differential mode (DM). Motor controllers usually have transistors that switch at very high frequencies, which implies that the voltage V across them have very fast rise and fall times t. CM noise originates from the stray capacitance between the power devices and the pump chassis driven by the dV/dt across the power transistors. The source of the DM noise is from the ripple current created by the high frequency switching of the motor controller. The filter for these CM and DM noise sources is usually a collection of inductors and capacitors which create a high impedance to these high frequency noise sources.

If the input to the pump is ac, then the input filter, at a minimum, must convert the ac into dc but may also have the task of power factor correction (PFC). Power factor is the ratio of real power to apparent power (distortion PF) multiplied by the cosine of the angle between voltage and current (displacement PF).

Consider the circuit in Figure 11 and the waveforms in Figure 12. The diode bridge and capacitor form a type of peak detector, with R1 representing the load. The diodes become forward biased (conducting current) only when the voltage from the anode to the cathode exceeds the voltage drop of the diode plus the voltage that is stored in the capacitor. When the input ac voltage falls below this point, the diode stops conducting and the current in the ac line falls to zero. The resulting current waveform, although perfectly in phase with the input voltage, has high peaks containing many higher order harmonics. These harmonics create reactive currents in the line which contribute to RMS heating of the power lines to the controller, necessitating over sizing of the wire feeding the pump. If the input to the pump is dc, then this section is omitted.

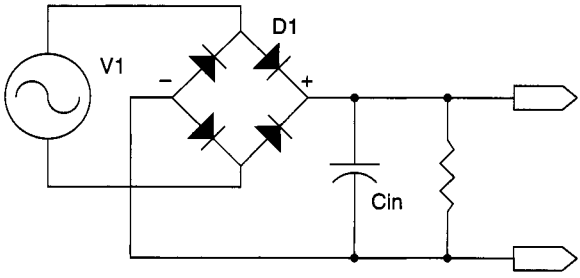


FIGURE 11 Simple diode bridge and capacitor bank

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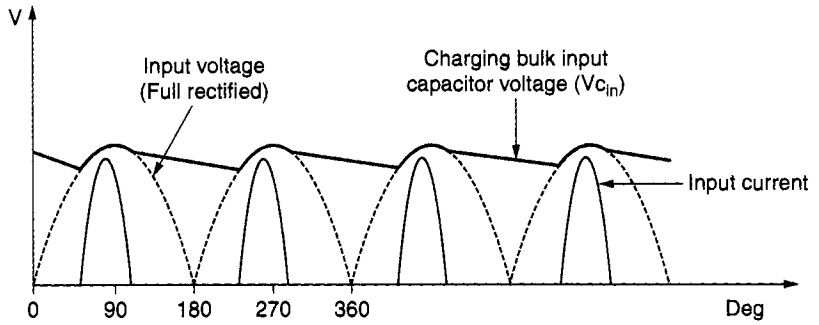


FIGURE 12 Waveforms of circuit in Figure 11

Power factor correction can be accomplished with pure passive components (capacitors, inductors, and transformers) or with a combination of active components (high frequency power electronics) and passive components. The use of active components running at high frequency may allow smaller passive components, but their advantage is somewhat mitigated by the fact that additional CM and DM noise sources are created, thereby complicating the design of the input power filter.

Motor Driver The function of the motor driver for a brushless dc motor is to replace the function of a mechanical brush and commutator. As the permanent magnet rotor rotates with respect to the stator, the controller must continually reverse the current flow in the individual coils to always keep torque in the same direction. Commutation of the motor is just the beginning. The complexity of the drive is a compromise between its desired functionality and the complexity of development in an aerospace environment. At one end of the spectrum, a brushless motor-driven pump can appear like a dc brush motor pump, simply spinning at a speed proportional to the input voltage as soon as power is applied. At the other end of the spectrum, the control electronics can adopt sophisticated control algorithms to vary the speed of the pump based on local sensors and commands from a digital communication bus.

A brushless motor controller must sense the rotor position with respect to the stator position in order to properly energize the stator coils. At the onset of a motor drive design for a brushless motor one must make a choice as to whether the controller will use embedded sensors in the motor for position feedback or use a "sensorless" scheme. A sensorless motor controller detects the rotor position by looking at the Back-EMF or current of the stator phases. With a sensorless design, no delicate electronic devices are mounted near or on the motor and only three phase wires connect the motor to the controller. On the surface, it may appear that the sensorless approach is the best option. But the complexities of developing in an aerospace environment, and almost always the necessity for either software or other programmable devices to realize a sensorless design can make the choice of adding sensors to the motor more economical over the life of the program.

There are instances that require motor position feedback as well. A high resolution position feedback device (resolver or encoder) may be required for very precise position or velocity control, operation at very slow speed or an extremely wide dynamic operating range. Digital signal processors (DSP) and control algorithms are becoming more and more sophisticated, which may obsolete the need for position feedback, even with the most challenging applications. If the environmental conditions permit, a simple hall-effect-type sensor can be integrated directly into the stator, providing a low-cost and reliable low resolution position sensor, suitable for commutation and velocity feedback.

However, if the design permits, adding microprocessors to a controller adds incredible flexibility. For instance, a pump can read a local pressure or flow sensor and vary the pump speed accordingly. Communication busses (CAN, RS-232, and so on) can be added to interrogate the pump's current status, health, operating time, historical running conditions, faults, and other information.

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Another consideration of a brushless motor controller is whether to run the motor as a "six-step" trapezoidal or with field-oriented control (FOC). A six-step drive is very simple in operation, but often can create torque ripple in the motor which may manifest itself as extra noise. FOC applies a sine wave to the motor phases, and with the proper motor (sine wave Back-EMF), a very smooth drive can be realized, with low vibration.

PERFORMANCE CRITERIA OF AIRFRAME BOOST PUMPS

The fact that aircraft boost pumps operate at very low net positive suction heads is evident in their operating environment. Fuel tanks are vented to ambient pressure and commercial airlines fly up to 45,000 ft altitudes where the standard atmospheric pressure is 2.14 lb/in^2 (.148 bar). Altitudes are even greater for supersonic aircraft. Tank pressurization above the ambient pressure is employed only on some specialized military aircraft.

The performance criteria that specifically define the design of the centrifugal pumping element in an airframe boost pump are presented in the following discussions.

MAXIMUM/MINIMUM PRESSURE RISE AND MOTOR SPEED

The pressure rise limits of the pump are defined by the pump specification and are determined by airframe and engine system needs.

The motor speed range is selected by the pump designer to be compatible with the overall requirements of the pump, in particular the altitude requirements. Because of the need to minimize size and weight, the maximum possible speed is employed. This has, in recent years, resulted in the almost exclusive use of axial inducers in airframe boost pumps. They are used either alone, as purely axial pumping elements, or in conjunction with mixed-flow sections depending upon the flow and pressure requirements. Axial inducers are a pumping element that have proven to provide the best suction performance because of their low blade loading and gradual pressure rise characteristic. The inducers on airframe boost pumps incorporate features such as blade sweep back, blended suction surfaces, and thin blade inlet edges. The inlet blade angle at the tip usually ranges from eight to ten degrees measured from a plane perpendicular to the axis of rotation. These pump elements have demonstrated the capability of operating at suction-specific speeds in excess of 40,000 measured in units of gpm, feet of fluid, and rpm (that is, the universal suction specific speed Ω_{ss} exceeds 14.6).

FUEL TYPE

Two types of fuels are in general use worldwide in aircraft gas turbine engines. They are referred to as "wide cut fuels" and "kerosene-based fuels."

Both types of fuel are composed of a complex mixture of a range of individual hydrocarbon compounds. As the name suggests, the wide cut fuels have a wider range of hydrocarbons than kerosene-based fuels. The composition of the fuel is controlled, rather loosely, by the fuel specifications through defining limits on such factors as distillation range, density, flash temperature, heat of combustion, vapor pressure, freezing point, additives, and limits on certain compounds. Wide cut fuels are characterized by relatively low density, low freezing point (−65°F/−53.9°C), high vapor pressure and low flash point (−45°F/−42.8°C). Kerosene-based fuels on the other hand are characterized by a relatively high density, higher freezing point (−40°F/−40°C), low vapor pressure, and high flash point (140°F/60°C). The U.S. designations for wide cut fuels for commercial and military uses are Jet B and JP-4, respectively. The designations for kerosene-based fuels are Jet A1 for commercial uses and JP-5 and JP-8 for military uses. Jet A1 predominates in production because of the huge demands of the airlines. Jet A1 is used in commercial service worldwide

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primarily for safety reasons because of its high flash point. Jet B is only used when its low freezing point is required; for example, in northern Canada in the winter. The U.S. Air Force previously used JP-4 as their standard fuel, but for availability and safety reasons have switched to JP-8, which is a military version of Jet A1. The navy uses JP-5 because of its high flash point for safety considerations on board aircraft carriers.

The knowledge of the true vapor pressure of the fuel used is necessary in the design and test evaluation of the centrifugal pump element. Because the fuels are a mixture of a range of hydrocarbons, the direct determination of the true vapor pressure is difficult. It is determined indirectly, for a desired temperature, through the Reid vapor pressure (RVP). The RVP is an average vapor pressure determined under a defined set of conditions, established by test at 100°F (38°C). The true vapor pressure (TVP) at this temperature is slightly higher than the RVP. Moreover, the TVP-versus-temperature relationship is determined by the RVP (see Reference 1).

Another key property of the fuel that directly influences the design and performance of airframe boost pumps is the solubility of air in the fuel. This is defined by Henry's law through a solubility coefficient. The ullage volume above the fuel in the aircraft tanks is generally vented to the ambient pressure. The maximum amount of dissolved air in the fuel will occur on the ground. The fuel will be in an air-saturated condition and can then contain as much as 14% dissolved air by volume.

Any reduction in the pressure of the fuel will result in the release of this air in accordance with Henry's law and expansion of this air to a volume in accordance with Dalton's law of partial pressures. In addition, as the pressure is reduced, the vapor pressure of the light end hydrocarbon constituents of the fuel is reached. They too will vaporize, adding to the volume of vapor evolved. Reference 1 provides a detailed review of all of the properties of aircraft gas turbine engine fuels.

ALTITUDE CLIMB PERFORMANCE

The altitude climb performance required of an airframe boost pump is specified in terms of altitude achieved versus time in minutes, flow rate required with respect to altitude, the fuel tank temperature with respect to altitude, and the minimum pump pressure rise required versus altitude. The temperature versus altitude variation is due to the cooling of the fuel through heat transfer to the cold ambient atmosphere and boiling of the fuel that is experienced in a climb event.

Immediately upon takeoff, as the aircraft gains altitude, the dissolved air in the fuel will begin to evolve. The rate of this air evolution will depend upon the climb rate of the aircraft. Presently, there is no accurate way to determine this rate of air evolution. This volume of evolved air is handled in the design of the pump by judicious oversizing of the inlet based upon experience and empirical design parameters. Further assistance in handling the volume of vapor is provided in tank bottom mount pumps through pump element inlet tip vapor vents and discharge hub vapor vents. Snorkel inlet pumps can receive vapor handling assistance through the liquid ring reprime element. Any remaining air and fuel vapor mixture is compressed as it passes through the increasing pressure within the inducer and is redissolved into the fuel. Therefore, the pressurized fuel delivered to the engine is free of vapor.

As the altitude increases, a tank pressure will be reached that will equal the vapor pressure of the fuel. At this point, especially with wide cut fuels, the vapor release will become quite violent (boiling). Here, the rate of vapor evolution will be determined by the capacity of the tank vent system. As the light ends leave the fuel, the vapor pressure of the fuel will decrease. This is referred to as "weathering" of the fuel. After the maximum cruise altitude is reached, the vapor pressure of the fuel will reach equilibrium with the altitude pressure.

The altitude climb test is the most difficult test to perform on airframe boost pumps because it must be a single fuel pass test without recirculation to accurately reproduce an actual climb condition. This requires approximately 1500 gal (5,680 L) of fuel for one test

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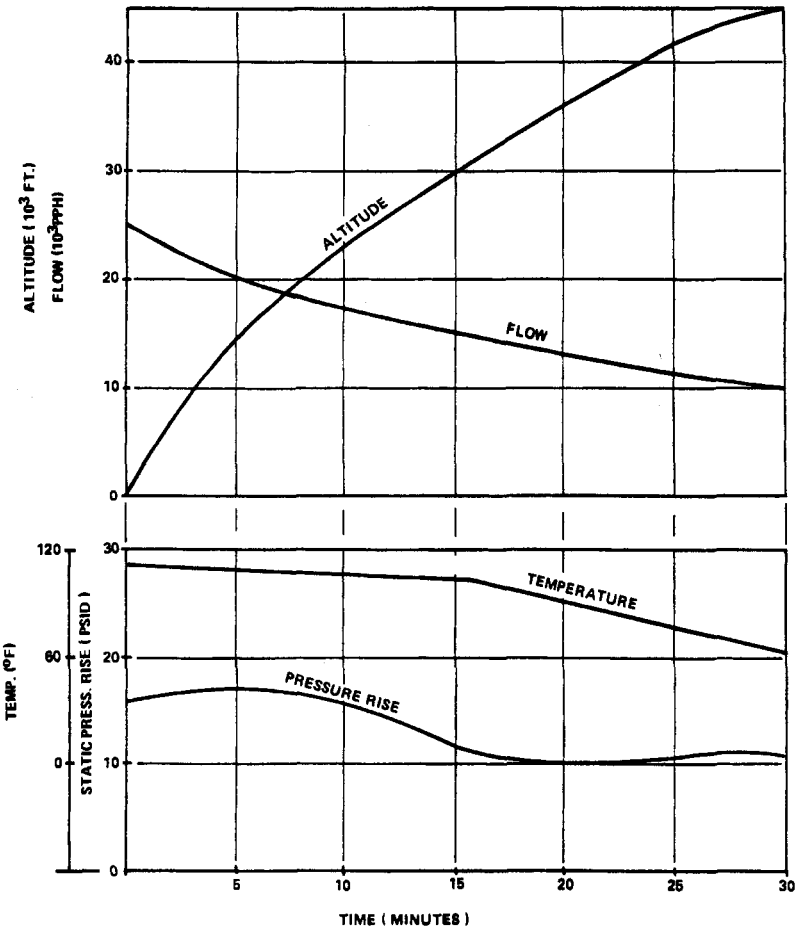


FIGURE 13 Main boost pump altitude climb (kg/h = PPH × 0.4536; m = ft × 0.3048; bar = psi × 0.06895; °C = (°F − 32) × 0.556)

on an average commercial airliner fuel boost pump element. Figure 13 presents the results of an altitude climb test on a typical main fuel boost pump.

CONSTANT ALTITUDE PERFORMANCE

This requirement for an airframe boost pump is specified in terms of a required fuel flow range at a given temperature and at a constant altitude. The minimum required pressure rise is also specified. The submergence of the inlet of the pump (for bottom mount types) or the snorkel inlet (for snorkel inlet types) is also defined and is measured in the range of a few inches. For altitude pressures lower than the initial fuel vapor pressure, the fuel is weathered to achieve equilibrium with the altitude pressure. The fuel is recirculated in this test. Figure 14 presents the constant altitude performance of the typical main fuel boost pump that has the climb performance of Figure 13.

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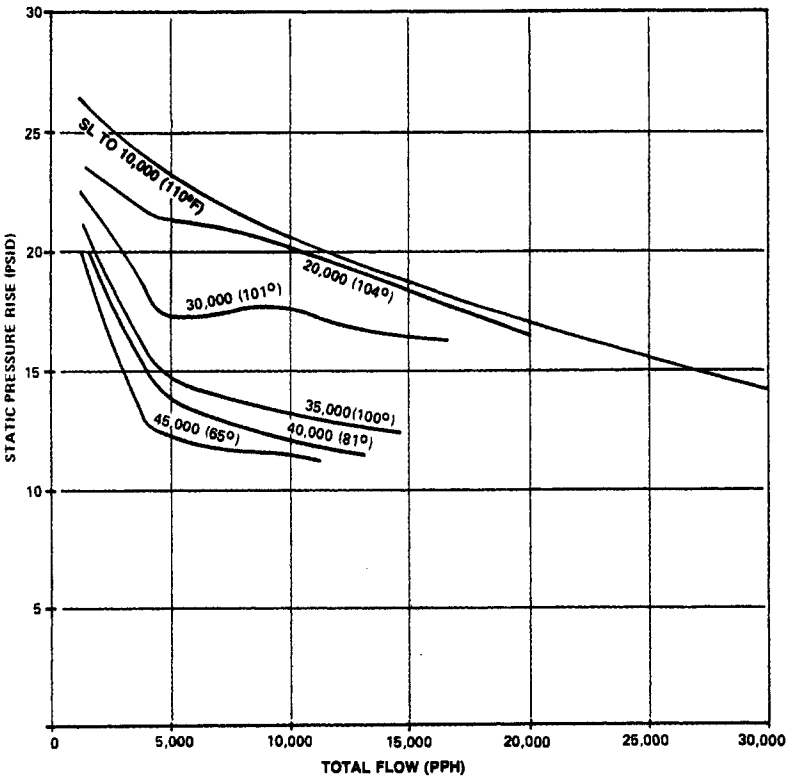


FIGURE 14 Main boost pump altitude performance (Altitudes are shown on the curves in ft; m = ft × 0.3048; kg/h = PPH × 0.4536; bar = psi × 0.06895; °C = (°F − 32) × 0.556)

SAFETY REQUIREMENTS FOR AIRFRAME FUEL PUMPS

Because they are one of the components which consume the most energy inside the fuel tank, airframe motor-driven fuel pumps have been closely scrutinized since the accident of TWA Flight 800 in 1996. Any possible ignition source in the fuel tank needs to be carefully examined and eliminated. Ignition may be caused by failure of either mechanical or electrical components within the fuel tank. The common failure modes of the airframe motor-driven fuel pumps in the aviation industry have been carefully documented in Reference 2, which requires that both the pump designers and users follow the guidelines for checking the pump safety features so as to avoid any malfunction, failure, or degradation.

Special attention in this regard is paid to the pump motor. All openings that communicate the pump electric motor cavity with the interior of the fuel tank must incorporate flame-arresting features. This includes communication through the pumping elements to the pump inlet. Also nonresettable thermal fuses or fusible links are incorporated in the motor end turns to ensure that the motor is disconnected before it can reach the minimum autoignition temperature of jet fuel (approximately 390°F/199°C) through various failure modes. The pump designer must follow the design and test guidelines in Reference 3, which includes attention not only to normal operation but also to the abnormal conditions such as dry run and locked rotor.

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GENERAL DISCUSSION OF ENGINE FUEL PUMPS

The primary fuel pump used for providing the boost and high-pressure fuel pumping function for an engine is called the "main fuel pump." It is engine gearbox-mounted and driven. In its most common form, it includes a positive displacement external spur gear high-pressure stage and an integral centrifugal boost stage. This type of pump is exclusively used on all modern commercial airliner engines and supplies all of the fuel flow requirements of the engine combustors. On commercial airliner engines, the main fuel pump is the sole engine fuel pump and therefore is a prime reliability piece of equipment. All engines, commercial and military, have a main fuel pump.

Positive displacement pumps have certain operating characteristics that make them very adaptable to aircraft gas turbine engine fuel systems. These characteristics include the ability to reprime from a completely dry condition and the ability to deliver useful fuel pressure over a wide speed range. Engine control systems utilize a portion of the main fuel pump discharge flow to power the actuation systems for fuel burn flow metering and engine variable geometry control for all operating conditions. Corner point operating conditions, such as altitude relight and ground starting, occur at 8 to 10% of rated operating speed. The pressure required for actuator muscle and response characteristics at these conditions is generally a 250 lb/in^2 (17.2 bar) pressure rise. At the maximum power takeoff condition at 100% speed, the pressure rise requirement ranges up to 1500 lb/in^2 (103 bar). Positive displacement pumps have proven capable of meeting these needs and the external spur gear pump type has received the widest acceptance from the industry.

The key attribute of the external spur gear main fuel pumps relates to the safety of flight and prime reliability requirements. In over 50 years of experience, this type of pump has demonstrated a graceful type of failure mode. It does not fail in a catastrophic manner in which it suddenly ceases to function. It slowly, over an extended period of time, degrades in performance until it no longer meets minimum requirements, but continues to function. In commercial service, these pumps routinely operate for over 10,000 flight hours without service or repair.

Table 1 presents the pertinent characteristics of representative external spur gear main fuel pumps.

TABLE 1 Main fuel pump parameter chart

Engine Application and Engine Power	Gear Stage Displacement in^3/rev (cm^3/rev)	Rated Input Speed RPM	Rated Output Flow GPM (m^3/h)	Rated Pressure Rise lb/in^2 (bar)	Unit Weight Pounds (kg)
• Commercial Airliner • 68,000 Pounds (302 kN) Of Thrust	3.487 (57.142)	6270	86 (19.53)	1250 (86.2)	34 (15.42)
• Fighter Aircraft • 28,000 Pounds (124 kN) Of Thrust	2.341 (38.36)	6075	55.2 (12.54)	1390 (95.8)	24.5 (11.11)
• Executive Jet • 5,000 Pounds (22 kN) Of Thrust	.631 (10.34)	6300	15.5 (3.52)	1200 (82.7)	12 (5.44)
• Helicopter • 650 H.P. (485 kW)	.233 (3.818)	4205	4 (.91)	1000 (69)	4.6 (2.09)

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Other types of positive displacement pumps, such as axial piston and sliding vane pumps, have been applied to engine main fuel pumps. These have not achieved the level of acceptance of external tooth gear pumps.

On highly specialized engines, such as military fighter aircraft engines, additional high-pressure fuel pump functions may be required. These include fuel pumps for supplying the thrust augmentation systems (afterburners) and, in some applications, fuel pressure for the actuation system for the engine's exhaust nozzle. Because augmentation fuel flows are very high and the systems are operational for only a small percentage of the mission time, high-speed centrifugal pumps are used in these situations. Centrifugal pumps offer a low unit weight and the ability to be run dry when the system is not operating, thereby conserving power. These centrifugal pumps operate at speeds up to 25,000 rpm and deliver up to 200 gpm (45.42 m^3/h at 1000 lb/in^2 (69 bar) pressure rise. The fuel actuation function has been provided by variable displacement piston pumps. The fuel boost function for the multiple high-pressure fuel pump engines is usually provided by a single centrifugal boost element either separately mounted on the gearbox or integrated with the main fuel pump.

An alternative drive means has been used for the centrifugal augmentor fuel pump in some applications. An air turbine utilizing engine compressor bleed has been used as a drive source. When the system is nonoperational, the pump may be stopped or idled by throttling the turbine inlet air supply, thereby conserving power.

For large engines, such as those used on commercial airlines, the boost stage is a conventional centrifugal element that generally incorporates an axial inducer. For smaller engines used on general aviation and helicopter applications, various specialized centrifugal elements and jet pumps have been used and have shown the capability of meeting the needs of suction feed fuel systems.

As discussed the engine boost pump must function with the airframe boost pump in both the operating and failed, or not operating, conditions. The engine fuel inlet pressure with the airframe boost pumps operational is generally specified over a range from 50 lb/in^2 (3.45 bar) to 5 lb/in^2 (.345 bar) plus TVP for all engine operating conditions. There are to be no vapor or air bubbles present in the fuel. For the condition of failed, or nonoperating, air frame boost pumps, the engine boost pump is required to operate with an inlet vapor-to-liquid volume ratio (V/L) of .45 at the maximum fuel tank temperature for takeoff conditions to at least a 10,000 ft (3048m) altitude (to cover high altitude airports) and appropriate V/L and fuel temperature conditions for maximum continuous thrust conditions up to 45,000 ft (13,716m). The V/L condition occurs because of the line pressure drop that occurs between the fuel tank and the engine boost pump inlet. This is depicted in Figure 15. The value of the V/L ratio is defined as the volume of vapor divided by the volume of fuel. A V/L of .45 represents a void volume ratio of approximately 30 percent.

Because the engine mounted boost pump is line-mounted and is external to the airframe fuel tank, the vapor entrained by the fuel cannot be ejected back to the tank as it is with the tank-mounted airframe pumps. All the vapor must be ingested by the boost pump and be compressed and redissolved back into the fuel. If the boost stage becomes overwhelmed by vapor, the vapor will build up at the pump inlet and the system will become vapor locked. This must be absolutely avoided.

The value of .45 for the takeoff V/L ratio has been established over the years as a value that will provide adequate vapor lock margin. See Reference 4 for a history of the V/L parameter. Airframe systems are analyzed to ensure the margin is maintained. The reduced usage of wide cut fuels in the future will somewhat relieve this requirement. Figure 16 presents the relationship between V/L and tank altitude. Note that for a constant value of V/L, as the tank altitude level increases, the pump inlet pressure, and therefore its NPSH, decreases.

In addition to the magnitude of the V/L ratio and the boost pump NPSH, the condition of the multiphase flow is important. To ensure continuous and stable operation of the system, the phases must be well mixed and have equal transport velocities. An operating limit is reached when the phases separate and transport velocities of the vapor phases are less than that of the liquid phase. Under these conditions, vapor will collect at a high point in the system. In time, the system will become unstable if some means of removing this collecting vapor is not provided. A rule of thumb for roughly estimating this transition point is a minimum velocity of 3 ft/sec (1 m/s) for the liquid phase occupying the full diameter of the pipe. For velocities below this level, the probability that phase separation will occur

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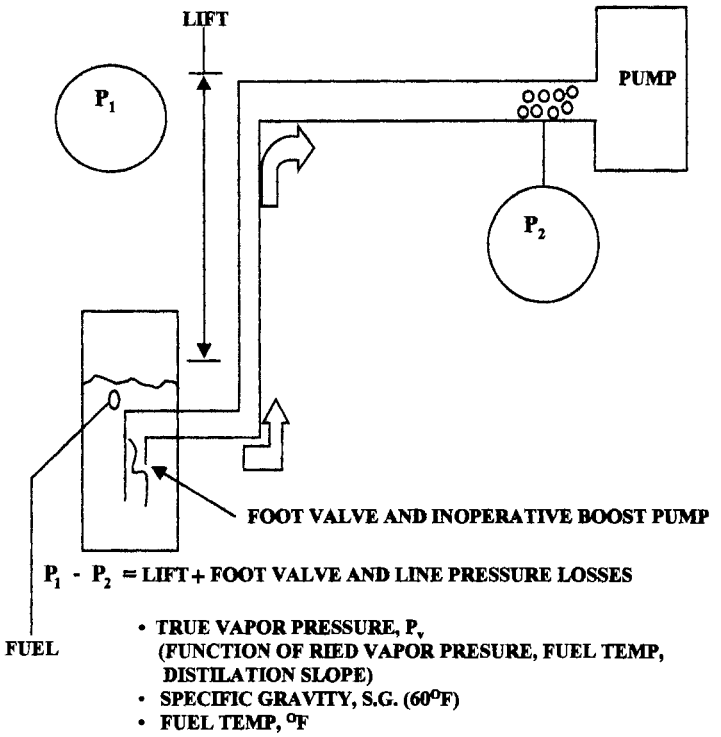


FIGURE 15 Factors that determine V/L ratio. $V/L = O \times (P_1 - P_2)/(P_2 - P_v)$, where the Ostwald coefficient O for typical fuels is in the range 0.1 to 0.2, depending upon these factors. Pressures P are absolute [°C = (°F − 32) × 0.556]

increases. In practice, an adequate mixing of the phases is maintained in pump tests that are run to prove the pump will meet the emergency conditions as previously defined.

For the suction-feed fuel systems employed on some helicopters, the uniform mixing of the vapor and liquid phases may not be maintained for all operating conditions. As previously discussed, these systems operate continuously under fuel-vapor-forming pump inlet conditions for all engine operating conditions; that is, from flight idle to maximum contingency power at all altitudes. Depending upon the engine fuel flow requirements, the fuel temperature and type and the fuel line configuration, separation of the liquid and vapor phases may occur. Because of the significant vertical pipe runs in these systems, this separated flow may exist in the form of alternating "slugs" of vapor and liquid flow in the line. When flow conditions of this type occur at the inlet to the engine fuel boost pump, a reserve volume of fuel must be provided within the pump. It is necessary to maintain a continuous flow of liquid fuel to the engine combustors for the time period when the vapor slug is being ingested by the pump and being compressed and redissolved into the fuel. Figure 17 is a pictorial presentation of the potential vapor and liquid flow conditions at the pump inlet. References 5 and 6 present a detailed review of these flow conditions.

DESCRIPTION OF THE DESIGN OF MAIN FUEL PUMPS

In this section, the design of a main fuel pump utilizing an external spur gear high-pressure element for application on engines powering modern airliners will be discussed.

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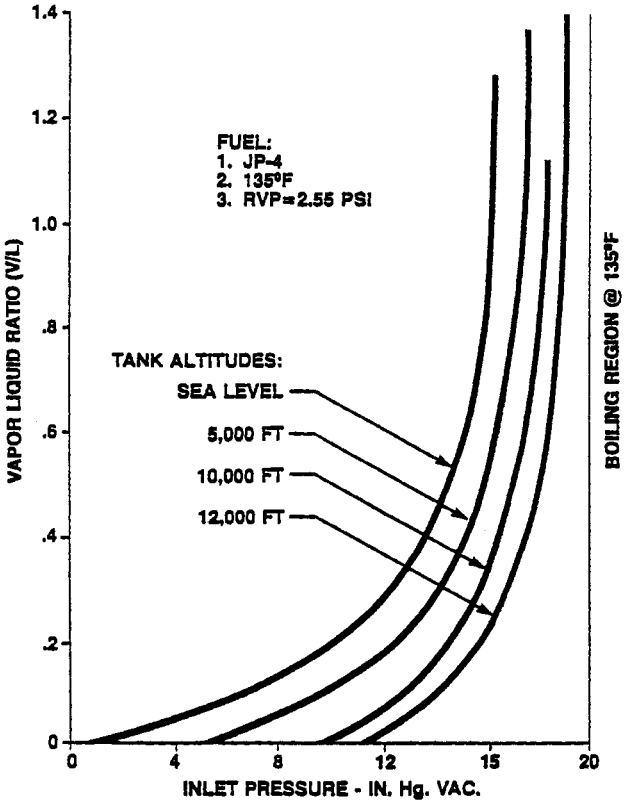


FIGURE 16 Vapor/liquid ratio versus inlet pressure (m = ft × 0.3048; bar = psi × 0.06895; °C = (°F − 32) × 0.556; mm Hg = in Hg × 25.4)

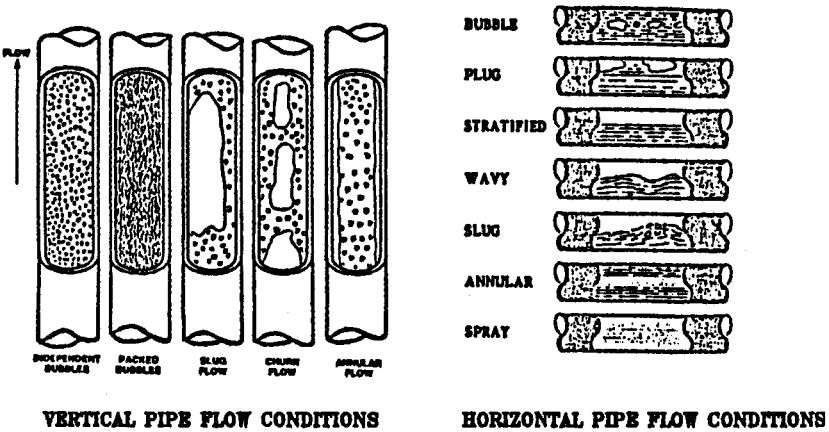


FIGURE 17 Possible multiphase flow conditions at pump inlet

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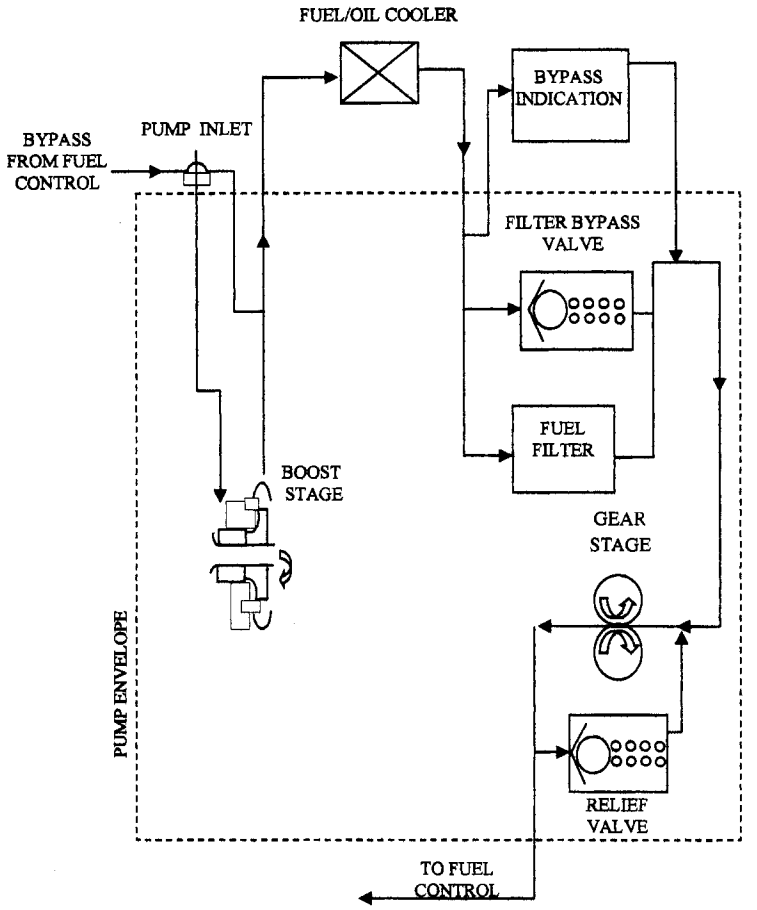


FIGURE 18 Main fuel pump system schematic

The designs of main fuel pumps for engines powering other classes of aircraft such as military fighters, executive aircraft, and helicopters are similar and differ mainly in the details of size and specific operating requirements. The flow and component schematic of a typical commercial engine main fuel pump is presented by Figure 18. The components encompassed by the dotted square are contained within the pump. As shown, the fuel from the airframe system enters the boost stage inlet. It then passes through the engine oil-to-fuel cooler where the fuel absorbs rejected engine heat, thereby fulfilling its function as a heat sink. This positioning of the oil-to-fuel cooler also provides the de-icing function for the next component in the system, the engine fuel filter. Using the engine oil-rejected heat for the de-icing function eliminates the need for fuel heaters using engine compressor bleed air as the heat source. These types of systems were used on previous generations of engines incurring weight and engine efficiency penalties.

Fuel filters are usually rated at 10 microns nominal and 40 microns absolute. The minimum filter surface area is usually determined by the worst icing condition. A filter bypass valve is provided to ensure continuous engine fuel flow in the event the filter becomes blocked. An indicator is provided to indicate impending filter bypass so maintenance actions can be initiated.

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The fuel then enters the high-pressure gear stage element and is delivered to the engine fuel control unit at a pressure determined by the downstream resistance characteristics. A high-pressure relief valve is utilized for burst protection for the high-pressure components of the fuel system in the event of a downstream blockage.

The control of the engine fuel flow is achieved in the fuel control system by bypassing the excess fuel back to the fuel pump upstream of the fuel oil cooler. This returned flow represents waste heat that is proportional to the quantity of the bypassed flow and the pressure differential between the gear stage and boost stage discharge pressures. This waste heat further increases the gear stage fuel inlet temperature.

The volumetric flow characteristics of positive displacement pumps closely match the fuel flow requirements of aircraft gas turbine engines. The ideal volumetric flow rate of a positive displacement pump is directly proportional to its rotational input speed. In actual practice, this is somewhat modified by the internal leakage flows that occur from the high-pressure discharge to the low-pressure inlet. These leakage flows are a combination of flow through the clearances between the pump parts and the necessary bearing lubrication and cooling flows. Also, leakage flows are proportional to the pressure rise across the pump and essentially constant over the operating speed range of the pump. Therefore, the leakage flows are a higher percentage of the ideal flow for lower percentages of rated pump speed. The percentage of the ideal flow that the delivered flow represents is the volumetric efficiency of the pump. Figure 19 depicts the basic relationship between the output characteristics of a positive displacement pump and the engine fuel flow requirements.

As previously discussed, the main fuel pump is required to provide both the engine burn flow requirements and the engine geometry actuation flow requirements. For a given class and type of engine (for example, commercial airline turbo fan engines), the engine geometry actuation flows are usually an essentially constant value for all engine speeds for a given number of actuator servos. Therefore, because of the volumetric characteristics of positive displacement pumps, the displacement sizing point of the high-pressure element for larger engines will tend to be the rated takeoff power high pump rotational speed condition. For smaller engines, it will tend to be the starting low pump rotational speed condition. The example in Figure 19 is sized at the starting condition. The disadvantage of this

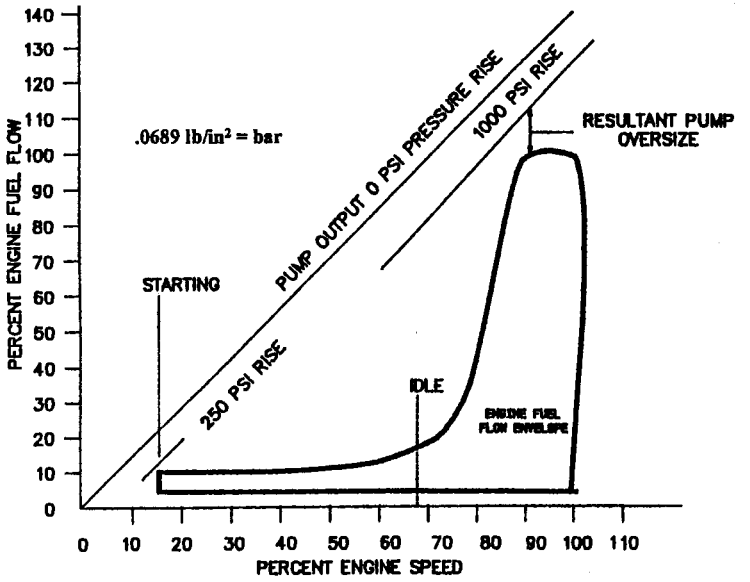


FIGURE 19 Positive displacement pump output characteristic versus engine flow requirements

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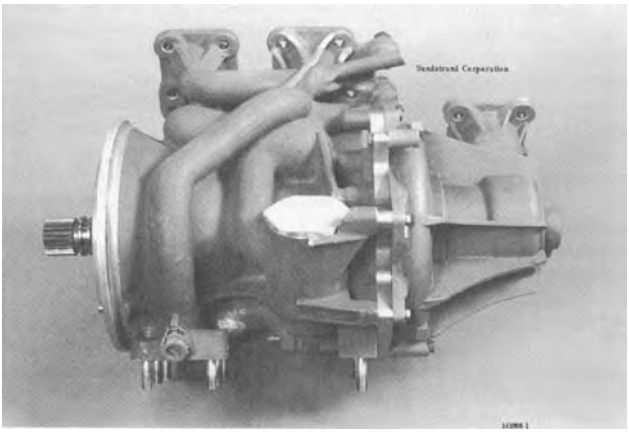


FIGURE 20 Main fuel pump assembly (Courtesy Hamilton Sundstrand)

type of pump concept, clearly shown in Figure 19, is the significant over-capacity of the high-pressure element for low engine power conditions such as the idle condition. This over-capacity represents the quantity of waste heat in the form of throttling loss in the bypass control loop that complicates the engine heat management system. The task is to avoid reaching the fuel thermal stability temperature limit (about 325°F/163°C) in the combustor fuel nozzles. Exceeding this limit will result in clogging the nozzles with fuel "coking" deposits. Significant effort has been expended on alleviating this problem through the application of variable displacement pumps. These efforts have not achieved general acceptance because of reliability, safety, and cost disadvantages.

Figures 20 and 21 present a photograph and a cross-section of a main fuel pump incorporating an external spur gear high-pressure stage and centrifugal boost stage. The pump is mounted directly to the engine gearbox and the rotational input power is transmitted directly to the spur gear stage by a spline coupling. The boost stage is driven by a secondary splined coupling. The drive priority is selected to reflect the power input order of

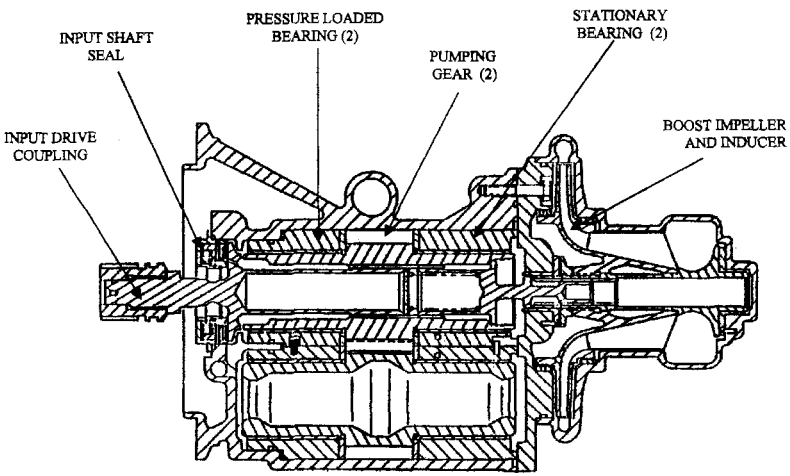


FIGURE 21 Main engine fuel pump cross section (Courtesy Hamilton Sundstrand)

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the elements and that the high-pressure element is the primary pumping element. The housings are aluminum castings that provide the minimum weight and also provide the necessary structural integrity and stiffness for all specified conditions.

A face-type dynamic mechanical shaft seal is provided at the drive end of the pump. A wide range of design configurations have been employed, but all modern pumps utilize the sealing faces as the primary seal and "O" rings for the secondary seal. These seals are required to operate with pressure differentials in both directions. Special design features must be employed to accommodate this requirement. The drive shaft seal is vented to a dry cavity that is connected to the engine overboard drain system. For reasons of safety, all fuel and oil passages separated by seals, static or dynamic, must have double seals with a drain port between them.

The type of external spur gear pump element used in main fuel pumps is the fully pressure-loaded type. Figure 22 depicts the basic details of the configuration used. The six parts shown in Figure 22 are assembled into a figure-eight bore arrangement in the pump housing. The four bearing blocks, which are a slip fit, are pressure-loaded toward the faces of the spur gears, and the assembly as a whole is pressure-loaded toward the inlet side of the housing bores. An initial axial sealing force is provided by springs. The axial pressure blow-off and pressure loading forces are closely balanced. This provides a pressure-balanced seal between the end faces of the spur gears and the bearings. This minimizes leakage over a wide range of delivery pressures on low viscosity jet fuels. The overall arrangement provides the necessary flexibility to maintain the minimum possible leakage clearances between the discharge and inlet over the complete range of operating conditions for many thousands of hours of operating life. The inherent flexibility of the design provides it with the unique capability of automatically compensating for the differing rates of thermal expansion of the materials used in the construction of the pump and the wear that will inevitably occur in usage.

The gears are manufactured from highly alloyed tool steels and surface hardened to resist wear in the harsh environment of low viscosity and low lubricity aircraft gas turbine fuels. The gear profile, lead error, and tooth spacing are held to very low limits. The roundness of the gear journals is also held to a very close limit and a mirror-like finish is applied. These journal dimensional characteristics are required because of the very low film thickness (6 to 30 micro inches or 0.15 to 0.76 microns) encountered. This is attributed to the low viscosity of hot aircraft gas turbine fuel (1 to .5 centipoise). The bearing surfaces, radial and thrust, are a highly leaded bearing bronze alloy generally with a solid film lubricant coating to assist in the initial "bedding" in.

The bearings are designed to provide full film lubrication for all operating conditions with the exception of the starting condition. On some pumps, a hybrid bearing design that incorporates a high-pressure pad to augment the bearings load carrying capacity is used. The bearing thrust faces contain gear trapping relief cuts that control the gear mesh flow dynamics. This includes avoiding any fluid trapping conditions, controlling gear stage inlet, and discharge pressure pulsations. Figure 23 is a photograph of a gear set with two of the bearings.

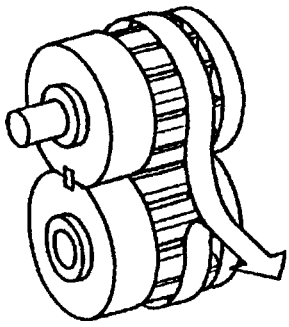


FIGURE 22 Pressure-loaded spur gear pump

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FIGURE 23 Spur gear pump gears and bearings (Courtesy Hamilton Sundstrand)

The design of the centrifugal boost stage is quite similar to that used for the airframe boost elements. The inlet must be sized to handle the total inlet flow rate consisting of the vapor phase and the liquid phase. The use of axial inducers has become common on all sizes of pumps because of the severity of the required inlet conditions. The selected overall pressure rise of the boost stage is primarily dependent upon the temperature and, therefore, vapor pressure of the fuel at the entrance to the high-pressure element and the various pressure losses between the boost stage discharge and the high-pressure element inlet. These pressure losses include filter pressure losses (clean filter and clogged filter), oil-to-fuel cooler pressure losses, and various coring and plumbing pressure losses. A pressure "stack" analysis between the boost stage inlet condition and the gear stage inlet pressure requirement must be made for all operating conditions across the pump input speed range—including the emergency conditions—to determine which operating condition is the critical sizing point for the pressure rise of the pump boost stage. A simplified example of a pump pressure stack is presented by Figure 24. Reference 7 discusses boost impeller sizing.

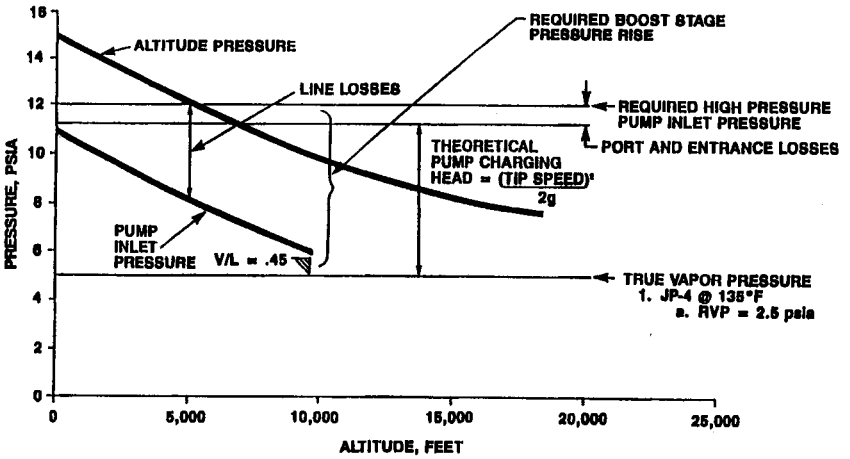


FIGURE 24 Typical pump inlet pressure stack (bar = 0.06895 × psi; °C = (°F − 32) × 0.556; m = ft × 0.3048)

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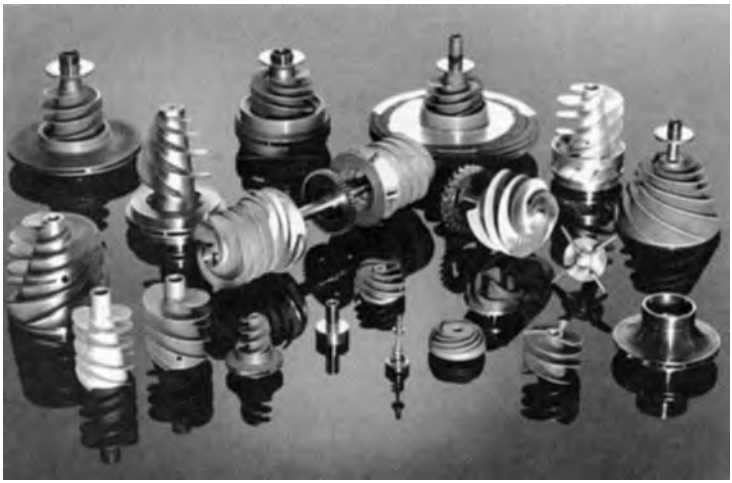


FIGURE 25 Various impellers and inducers (Courtesy Hamilton Sundstrand)

Figure 25 is a photograph of various impellers and inducers that have been used in airframe and engine fuel pumps. The specific speeds in units of feet of fluid, gpm, and rpm range from 500 to 6000 (universal specific speed Ω_s = 0.18 to 2.2) and suction-specific speeds in the same units range up to 40,000 (14.6).

With respect to boost stages, particular attention must be paid to minimize the conduction of heat from the hot high-pressure stage to the relatively cool boost stage. This is required to avoid the possibility of vapor lock through fuel "boiling" in low-pressure regions of the boost stage under conditions of low fuel burn flow rates and, therefore, low boost stage through flow rates. These conditions generally occur at engine idle and descent operating conditions.

The pump-splined drive couplings are key to the reliability and safety criteria of the pump. Similar to the pump gears, the splined couplings are fabricated from high alloy steels with a surface hardening treatment. The design of the involute splines in terms of profile wear must take into account the lubricant used for the splines and the misalignment imposed by the various drive line elements. A spline design that will satisfy the wear conditions at rated speed will generally meet all overload conditions, including the maximum shaft shear torque requirements. Spline lubricants that have been successfully used include engine oil, fuel, and specially blended greases. Specific spline design parameters must be applied for each lubricant.

PERFORMANCE CRITERIA OF ENGINE MAIN FUEL PUMPS

The key performance criterion for the high-pressure positive displacement stage of a main fuel pump is to be capable of delivering the volumetric fuel flow rate required by the engine for all operating conditions. Before it is released for flight usage, the test sequence with which the positive displacement stage must successfully comply is primarily directed at evaluating the durability of the stage under the most extreme conditions it is expected to encounter in service. There are success criteria for these tests. At the completion of each individual test, the stage must meet its specified volumetric flow service limits. Upon completion of all the tests, the unit is subjected to a teardown inspection and the component parts must not exhibit any unusual wear or distress and must be in a condition that is deemed acceptable for continued service.

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The key performance criterion of the boost stage is to provide adequate pressure to the inlet of the gear stage to suppress fuel vaporization and cavitation for all operating conditions; that is, with assistance from the airframe boost pumps; and the emergency conditions without assistance from the airframe boost pumps. The tests that confirm this capability are run in conjunction with the positive displacement stage and are subject to similar success criteria. The performance criteria that specifically determine the design of the boost and high-pressure main fuel pump are presented in the following discussions.

GEAR STAGE VOLUMETRIC PERFORMANCE

The engine fuel flow requirements are specified in mass units. These must be converted to volumetric flow units for the lowest density fuel specified for all operating conditions. For these volumetric flow rates, the displacement required for each operating condition is determined by applying factors for pump volumetric efficiency that encompass the temperature of the fuel, production variance, and the service life flow deterioration. When the critical displacement sizing operating condition has been established, the overall volumetric performance of the pump can be defined for the complete input speed and pressure rise operating range including the required input power. Figure 26 presents the overall performance characteristic of a pressure-loaded external spur gear high-pressure stage of an aircraft engine main fuel pump.

GEAR STAGE CYCLIC DURABILITY

The ability of the pump to accept the cyclic duty imposed by modern engines without distress has been an accurate predictor of the pump's capability of meeting its service life requirement. The test is based upon the real-time pressure, temperature, and speed transients for engine operating conditions such as starting, ground idle, takeoff and climb, cruise, descent, and thrust reverse.

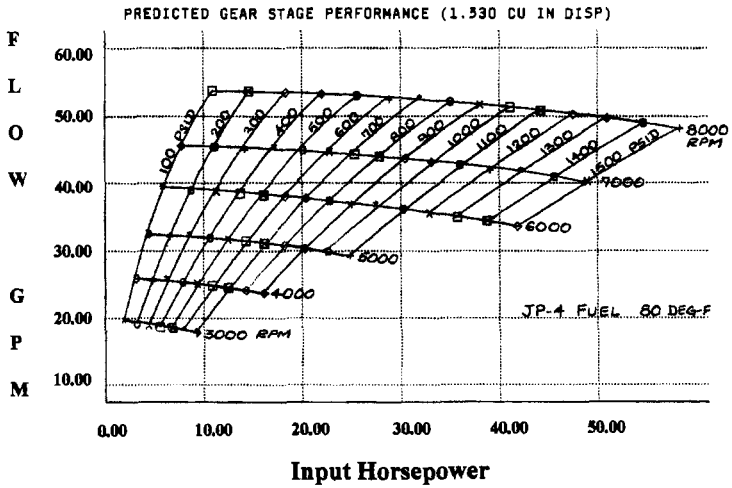


FIGURE 26 Gear stage performance ($m^3/h = 0.227 \times gpm$; $kW = 0.7457 \times hp$; $bar = 0.06895 \times psi$; $liter = 0.0164 \times in^3$; $°C = (°F - 32) \times 0.556$)

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The worst cases for the sequencing of the transients are selected to achieve the highest levels of stress in the gear and bearing system. The real transient times are halved and the steady state operating times minimized to accelerate the test. The number of test cycles and overall test time are based upon the expected service life of the pump.

LOW LUBRICITY FUEL

The use of the pumped fuel as the lubricant for the pump and all components within the fuel system is an established and required practice. Unfortunately, there is no requirement for lubricity in the fuel specifications. Therefore, some knowledge of the minimum lubricity to be expected in the field is necessary. The key factor to determine is the lubricity of the fuel under boundary lubricating conditions for evaluating the wear characteristics of the gear tooth profiles and the fuel lubricated spline teeth. An intensive cooperative industry and government evaluation effort has identified the Exxon ball and cylinder wear test machine as an accurate and reliable method of evaluating the lubricity characteristic of aircraft gas turbine fuels. The pump specification will contain the fuel lubricity level with which the pump will be required to demonstrate operation for its intended service. The pump designer must select gear materials and geometries that are compatible with the specified lubricity level and operating conditions. Wear experienced on low-lubricity fuel is a threshold type of surface failure. Therefore, this threshold limit must be avoided for all operating conditions. The test for compliance with this requirement is usually operation at takeoff and cruise conditions on the specified fluid at controlled lubricity conditions. The fluid lubricity level is checked periodically to ensure compliance is met. See Reference 8 for a detailed description of the test procedure.

COLD STARTS

Cold start tests demonstrate the capability of the pump to provide the performance necessary to start the engine under the most severe cold conditions expected and to demonstrate that the fits and clearances in the pump are compatible with the low temperatures. The cold start requirement ranges from −40°F (−40°C) to −65°F (−53.9°C), depending upon the fuel used and service expected.

CONTAMINATED FUEL

The large amount of fuel an engine burns in operation and the widely varying service conditions worldwide virtually ensure that some contaminants will be introduced into the engine's fuel system. In addition to these operational contaminants, there will be built-in contaminants because of the complexity of the airframe and engine fuel system that will be experienced in the initial operation of the pumps. Tests are specified that define operation of the pump at various operating conditions on fuel containing both liquid and solid contaminants. These included salt water, quartz crystals, sand, and iron oxide ranging in particle sizes from 1500 microns to less than 5 microns. These tests are more severe for military applications than commercial applications. Military applications usually require the contamination to be metered into the inlet of the pump in relation to the fuel flow rate and then removed from the system downstream of the pump by a filtration system. Hence, the contaminant is passed through the pump in a continuous single pass manner. In all modern fuel systems, the high-pressure positive displacement stage is protected by the engine-mounted fuel filter element. The low-pressure boost stage is not protected by the engine-mounted filter element. Although the centrifugal boost stage is inherently capable of operating on the specified contaminant, special design features are employed to minimize the abrasive wear effects of swirling contaminated fuel.

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V/L CAPABILITY

The reason for the need of the boost stage of an engine main fuel pump to have a V/L capability and some detail of the design requirements has been presented in previous discussions. The test set-up used for V/L testing is similar to the example shown in Figure 15. The testing is usually accomplished in two parts. Both parts are run with the maximum fuel RVP expected in service. The first part is to evaluate the V/L performance of the pump at all of the specified emergency conditions and prove all conditions can be met. The second part is an endurance test at high V/L conditions to ensure that no excessive cavitation erosion damage occurs that could limit the performance or life of the pump.

MINIMUM INLET PRESSURE

As long as the main fuel pump is being assisted by the airframe boost pumps, it must meet a minimum inlet pressure requirement. This inlet pressure is 5 lb/in^2 (0.345 bar) above the true vapor pressure (TVP) of the fuel. At this condition, the main fuel pump is required to supply the required engine fuel flow for all operating conditions. The design procedure and details for meeting this requirement were previously discussed.

The test set-up is arranged to ensure that vapor-free fuel is provided to the inlet of the fuel pump at 5 lb/in^2 (.345 bar) above TVP for all operating conditions. Heat exchangers are introduced into the system as required to establish the required boost and gear stage inlet temperatures. The full range of hot operating conditions, including the critical altitude idle conditions, are run to ensure vapor lock conditions do not occur. The maximum fuel RVP expected in service is used.

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12.18.2 LIQUID ROCKET PROPELLANT PUMPS

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Liquid-fueled rocket engines have been used on all the major launch vehicles, including the large, first-stage booster rockets as well as the upper stages of those same vehicles, and various smaller vehicles such as the lunar lander. The propellant pumps are major components of the engine. They pump a) the oxidizer, usually liquid oxygen, and b) the fuel, which is usually liquid hydrogen, up to the high pressures needed to feed the combustion processes. In the past, a widely used fuel was kerosene in a rocket propellant formulation specified as "RP1."

Of necessity, the pumps for these engines must have the minimum possible size and weight. They therefore run at extremely high speeds. Most have inducers or are fed by inducer pumps. These inducers have extremely high suction-specific-speed capability, N_{ss} exceeding 35,400 (Ω_{ss} = 13) and being as high as 70,000 (Ω_{ss} = 26) in liquid hydrogen[1]. This is so because the hydrogen is nearer its thermodynamic critical point and so generates less vapor volume when it cavitates than do other propellants. (See the discussion under "NPSH-Effects" in Section 2.1.1. All symbols in this section are defined in the nomenclature of that Section 2.1.1.) The main engine propellant pumps have higher heads per stage than any other centrifugal pumps in existence. As such, they are the world's highest-energy pumps, as indicated in Figure 32 and Table 13 of Section 2.1.1. Whereas the high-energy pump portion of that section is devoted largely to the challenges of designing such machines for long life—say 40,000 hours or more—the life of a rocket engine pump is measured in minutes; or, in the case of reusable rockets, not much more than a few hours. For good materials choices, such a short operating life means that these pumps are unlikely to suffer failure from cavitation erosion or other wear-related phenomena, despite the high inlet tip speeds of the inducers and impellers. Extreme attention to details of the mechanical design is required in order for these pumps to survive at design conditions—near which they generally operate. A few examples of rocket propellant pumps are given in this section, but many others also exist in various configurations[1,2].

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THE SATURN V BOOSTER ROCKET ENGINES

The Saturn V booster rocket was used in the Apollo program of the 1960s and '70s, which landed men on the moon. This rocket had a vertical height of more than 350 ft (197 m) at launch. The first or lowest stage was the largest and was propelled by five F-1 engines, each producing 1.5 million pounds (6.7 million N) of thrust. Propellants were liquid oxygen and RP1. Each engine was topped by a turbopump assembly that included both pumps on the same shaft and driven by the same hot-gas turbine. A cross-section of this assembly is shown in Figure 1, the two pumps being arranged in a back-to-back configuration and each having an inducer. The RP1 pump is the one next to the turbine, there being a considerably lower temperature difference between RP1 and the hot gas flowing through the turbine from the combustor than there would be for liquid oxygen. At the design speed of 5,490 rpm, the two pumps together consumed 52,700 hp (39 MW)—and considerably more at overspeed (approximately 6000 rpm). Table 13 of Section 2.1.1 contains performance figures. Essentially the same data are presented in the following descriptions of the pumps, which are summarized in Table 1. The table has been developed from various sources, including the references cited at the end of this section, as well as information supplied by the Boeing Company.

- *The oxygen pump* took in the liquid axially through a 15.75-in (400-mm) diameter inducer at the opposite end of the assembly from the turbine, as seen in Figure 1. Designed for a suction-specific speed N_{ss} of 35,400 ($\Omega_{ss} = 13$) meant that when ingesting 25,080 gpm (1.58 m^3/s) this machine could operate at an inlet static pressure of 22 lb/in^2 (0.15 MPa) above the vapor pressure p_v of the liquid oxygen. Because this liquid is cryogenic, p_v equals the pressure inside the fuel tank of the first stage of the rocket, which was not much above atmospheric pressure. Actually, the g-force and head of liquid in the tankage above the pump combined to provide additional available NPSH. This pump, which was designed for a specific speed N_s of about 2,100 ($\Omega_s = 0.77$), consumed

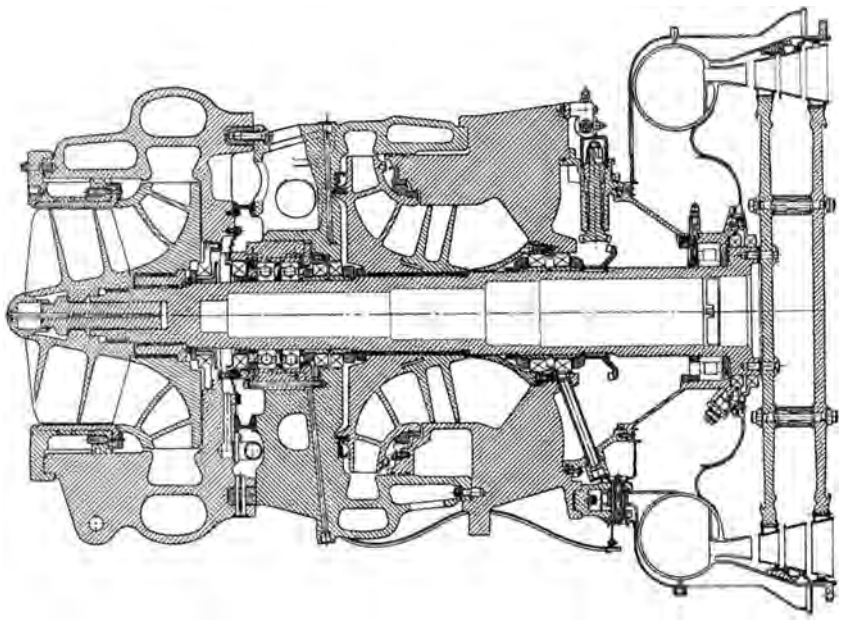


FIGURE 1 F-1 Turbopump assembly, used on Saturn V first stage rocket engine (Courtesy of The Boeing Company)

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TABLE 1 Data on liquid rocket propellant pumps*

Parameter	Units	F1 Engine		Space Shuttle Main Engine			
		Oxygen pump	RP1 pump	Oxygen pumps		Hydrogen pumps	
				LPOTP	HPOTP	LPFTP	HPFTP
Shaft power	hp (MW)	30,000 (22.4)	22,700 (19.9)	1,740 (1.3)	27,770 (20.7)	2,900 (2.2)	77,000 (57.4)
Rotative speed	rpm	5,490	5,490	5,450	31,100	15,700	37,400
Mass flow rate	lb/s (kg/s)	3,977 (1,804)	1,762 (799)	964 (437)	1,148 521	161 (73)	161 (73)
Inlet volume flow rate	gpm (m^3/s)	25,080 (1.58)	15,640 (0.99)	6,080 (0.38)	7,240 (0.46)	16,300 (1.03)	16,300 (1.03)
Pressure rise	Lb/in^2 (MPa)	1,530 (10.5)	1,810 (12.5)	332 (2.3)	4800** (33.1)	222 (1.5)	6,840 (47.2)
Head rise (approx.)	ft (m)	3,100 (945)	5,100 (1,555)	670 (204)	9,700 (2,960)	7,240 (2,200)	200,000 (61,000)
Specific speed (stage)	rpm,gpm,ft (universal)	2,095 (0.77)	1,130 (0.41)	3,230 (1.18)	2,700 (0.99)	2,550 (0.93)	1,150 (0.42)
Inducer diameter	in (mm)	15.75 (400)	16.00 (406)	11.725 (298)	4.7 (119)	12.014 (305)	
Impeller exit diameter	in (mm)	19.50 (495)	22.95 (583)		6.80 (173)		12.00 (305)
Overall diameter x length	in x in (mm x mm)	44 x 60 (1,118 x 1,524)		18 x 18 (457 x 457)	24 x 36 (610 x 914)	18 x 24 (457 x 610)	22 x 44 (559 x 1,118)
Turbopump weight	lb (kg)	3,080 1,397		276 (125)	622 (282)	177 (80)	775 (352)

*Compiled from information supplied by The Boeing Company and References 1–5.

**Main stage. Small preburner stage pumps 117 lb/sec (53 kg/s) to approximately 3300 psi (23 MPa) above main stage discharge.

30,000 hp (22 MW) or 57% of the total turbine shaft power, and it generated over 1,500 lb/in^2 (10 MPa) of pressure rise.

- *The fuel (RP1) pump*, being in the middle of the turbopump assembly, had to take in the 15,640 gpm (0.99 m^3/s) of RP1 through a side inlet piping configuration, which generally results in less N_{ss}-capability than axial inlet piping. Here the inducer is larger and the flow rate lower than for the liquid oxygen pump, creating the potential for higher N_{ss}; yet it ended up being lower, namely $N_{ss} = 34{,}200$ ($\Omega_{ss} = 12.5$). Nonetheless, this enabled the RP1 pump to operate at a static inlet pressure as low as 16 lb/in^2 (0.11 MPa) above the (negligible) vapor pressure of this liquid. With specific speed N_s just over 1,100 (0.40), this pump consumed the remaining turbine shaft power of 22,700 hp (17 MW) and generated a pressure rise of just over 1,800 lb/in^2 (12.4 MPa).
- Throughout the burn, these pumps deviated no more than about 8% from the design value of Q/N; that is, the flow coefficient was essentially constant. Thus, this turbopump did not

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suffer from the off-design low-flow conditions of most commercial, industrial pumps, as described in Section 2.1.1 including those of high energy level, which made it a little easier for this machine to operate at high energy levels.

THE SPACE SHUTTLE MAIN ENGINES

The space shuttle program, which began in the 1970s, dealt with a launch vehicle that was flanked by two large solid-fueled rockets. In the middle at the base were three liquid-fueled "main" engines, each having nearly 500,000 lb (2.2 million N) thrust and propelled by hydrogen fuel and oxygen at a mixture mass ratio of 6:1 (oxygen to hydrogen). Each of these space shuttle main engines (SSME's) has a dedicated pumping system, consisting of a low-pressure single-stage inducer-type pump for each propellant, which in turn feeds a high pressure pump. Three of these four pumps can be seen in the photograph of the engine in Figure 2. Located up above is the low-pressure hydrogen pump, with its large discharge line going off to the right and down to the high-pressure hydrogen pump below. Opposite, on the left, is the high-pressure oxygen pump, with its large inlet line also in view. Engineering of these high-pressure propellant pumps embodied a triple challenge in comparison to the F-1 pumps, because a) this engine was designed to be re-used, so the pumps had to have a life of 7.5 hours with 100 starts; b) the pumps for this engine have higher energy levels than the F-1 pumps, and, c) due to the mission profile, they have to be throttled back further from the design point. Each of these pumps is boosted by a low-pressure inducer-type pump to suppress cavitation enough to maintain performance[3].

The propellant flow system is illustrated in the schematic diagram of Figure 3. (The pump speeds, pressures, and flow rates shown in this figure are somewhat lower than the design conditions of Table 1, which for the SSME pertains to the maximum engine thrust level.) All four of the pumps are driven by turbines and so are called "turbopumps." The system picks itself up by its own bootstraps, so to speak; each low-pressure turbopump boosts the flow to the corresponding high-pressure pump and is driven by the same fluid that it pumps. This driving fluid comes back from each high-pressure pump, the low-pressure oxygen turbopump (LPOTP) being driven by recirculated liquid oxygen, and the low-pressure hydrogen or fuel turbopump (LPFTP) turbine being fed by gaseous hydrogen heated by the thrust chamber, as indicated in Figure 3.

The high-pressure pumps are driven by turbines fed by "preburners," which are combustors that burn hydrogen-rich. Some of the fuel entering these combustors is the gaseous hydrogen coming from the LPFTP turbine exhaust, which cools the turbine housings on the way to the preburners. But most of the fuel supplied to the preburners is the 80% of the liquid hydrogen discharging from the high-pressure fuel turbopump (HPFTP), which first flows through the cooling passages of the nozzle walls. [Eleven percent of the oxygen is also fed to the preburners by way of the preburner boost stage that is a part of the high-pressure oxygen turbopump (HPOTP) package. Finally, the partially burned fuel passes as a hot gas into the main combustion chamber, where more oxygen is added and the pressure is 3,000 lb/in^2 absolute (21 MPa)[3,4].] Most of the remaining 20% of the hydrogen is that which was already described as cooling the main combustion chamber and, along the way, becomes gaseous and powers the drive turbine of the LPFTP. It also cools the hot-gas manifold and injector and pressurizes (in a small amount) the fuel tank. Approximately 75% of the liquid from the high-pressure oxygen turbopump (HPOTP) goes directly to the main combustion chamber, 11% to the preburners (as already stated), about 13% to drive the turbine of the low-pressure oxygen turbopump (LPOTP), and a small amount is sent to pressurize the tank.[5] A brief description of each pump follows:

- *The low-pressure oxygen turbopump (LPOTP)* consumes 1,740 hp (1.30 MW) and runs at 5,450 rpm. It has a single-stage, axial-flow, inducer-type impeller that is 11.725 in (298 mm) in diameter and is driven by a six-stage liquid-oxygen hydraulic turbine. Pump head rise is 670 ft (204 m) so the HPOTP therefore operates without pressure

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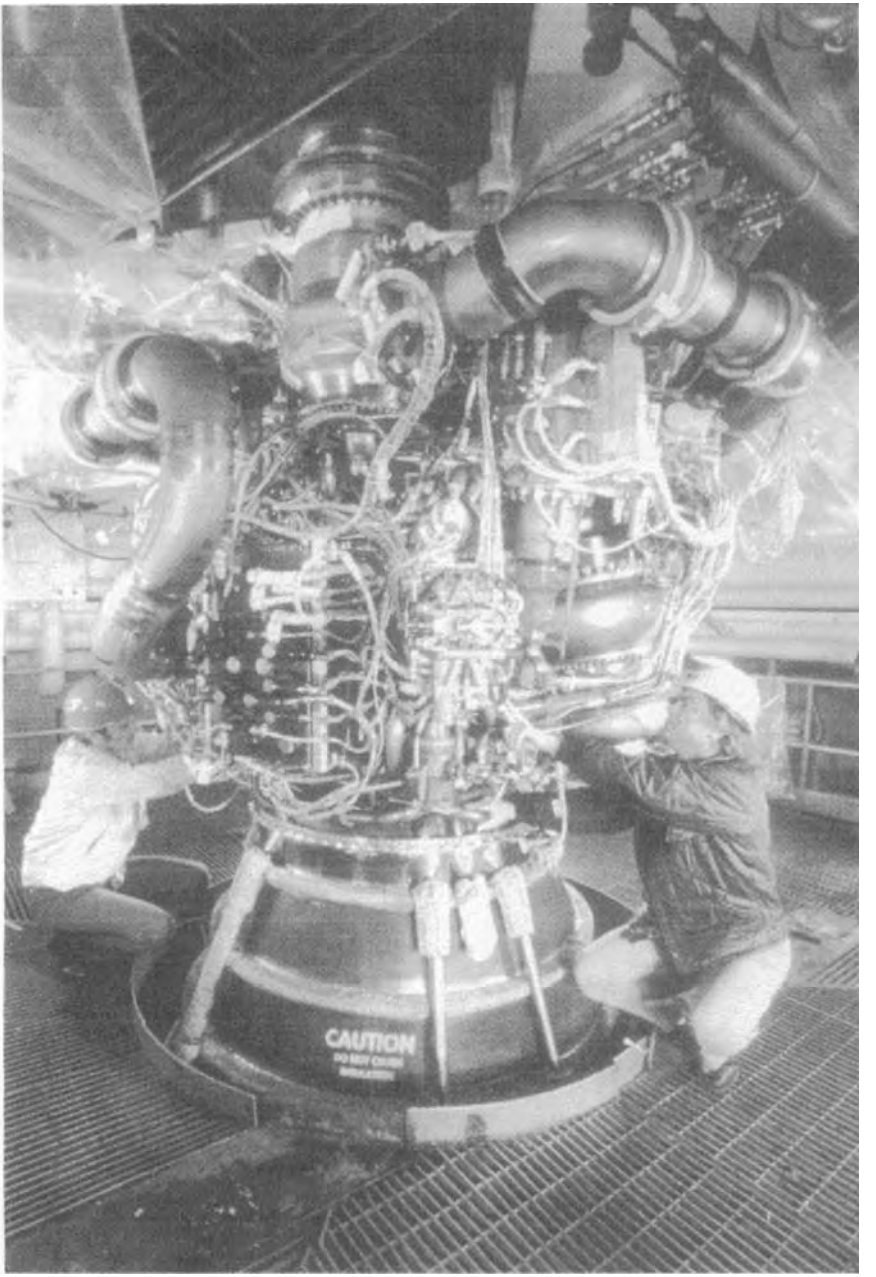


FIGURE 2 The space shuttle main engine (SSME) (National Geographic Magazine, March 1981: Jon Schneeberger/NGS Image Collection)

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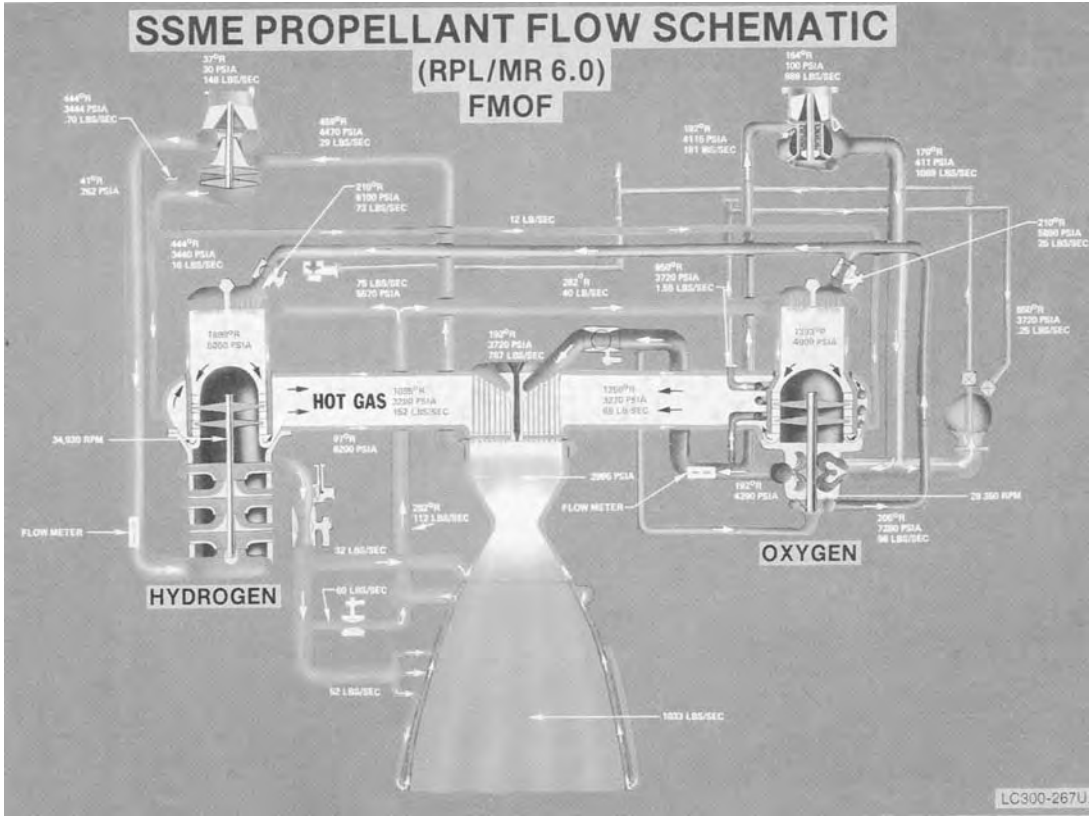


FIGURE 3 SSME liquid oxygen and hydrogen pumping arrangement (Courtesy of The Boeing company)

loss due to cavitation at all engine conditions. Specific speed N_s = 3,230 (1.18), which is low for an inducer, but this one performs a more substantial pumping task than do inducers running at high speeds on the same shaft as the high-pressure centrifugal stages they feed. The shaft has two ball bearings cooled by liquid oxygen, and the envelope of the package is 18 in (457 mm) in diameter × 18 in (457 mm) long.

- *The low-pressure fuel turbopump* (LPFTP) is similar in concept to the LPOTP. It is a 2,900 hp (2.2 MW) machine that runs at 15,700 rpm. The inducer-type impeller is 12.014 in (305 mm) in diameter and is driven by a two-stage gaseous-hydrogen turbine. It ingests liquid hydrogen to the extent of $\frac{1}{6}$ of the mass flow rate of oxygen. Due to the very low specific gravity of liquid hydrogen (= 0.0708 at atmospheric pressure), the head rise is more than 7,000 ft (2,100 m). N_s = 2,550 (Ω_s = 0.93). The shaft has three ball bearings cooled by liquid hydrogen, and three seals limit leakage between the pump and the turbine before engine start and during operation.
- Both low-pressure pumps exhibited suction-specific speeds N_{ss} in excess of 35,000 (Ω_{ss} = 13)[4]. They are shown schematically at the top of Figure 3, and the high-pressure pumps are below them.
- *The high-pressure oxygen turbopump* (HPOTP) is shown in cross-section in Figure 4 together with the associated preburner unit. Inducers are shown in the detailed cross-sectional drawing of Figure 5 feeding the double-suction main stage that generates nearly 5,000 lb/in^2 (34 MPa)—making this one of the world's highest-energy pumps (as defined in Section 2.1.1, Figure 32). And more than 3,000 lb/in^2 (21 MPa) in addition to this pressure is generated in the small booster impeller (on the outer end of the turbopump as seen in Figures 4 and 5) that feeds about 11% of the main flow to the preburners for driving both high-pressure pumps. Both shrouds of the main impeller act as orificed balancing disks to keep all but a small preload off the two angular-contact ball bearings, which are cooled by liquid oxygen. Radial loads are minimized by the use of a vaned diffuser. The compact design and high speed (over 31,000 rpm) of this 28,000 hp (21 MW) machine result in operation between the first and second critical speeds. Several seals are utilized,

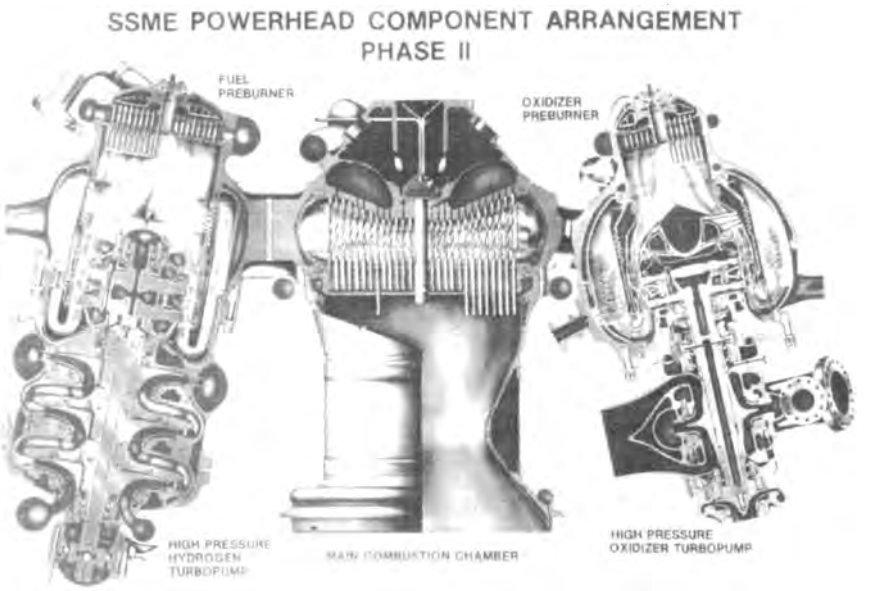


FIGURE 4 SSME high-pressure turbopumps (Courtesy of The Boeing company)

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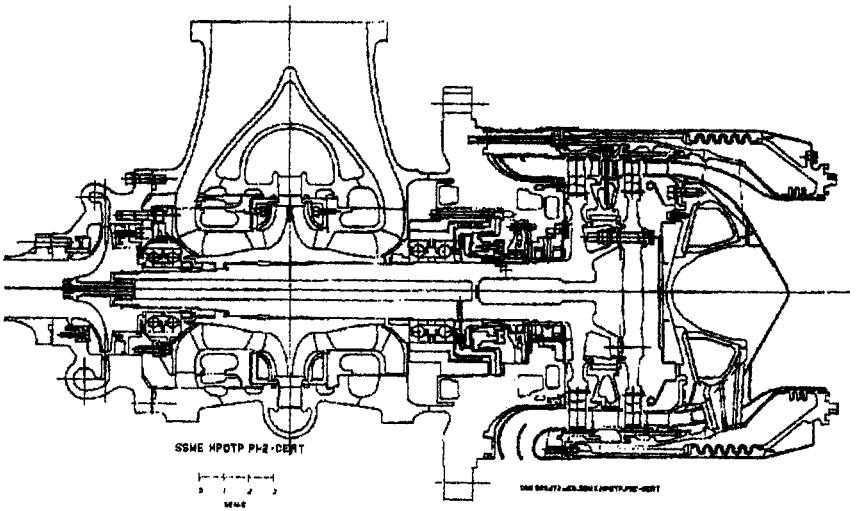


FIGURE 5 Cross-section of SSME high-pressure oxygen turbopump (Courtesy of The Boeing company)

including a helium-purged seal that isolates the hot gas of the turbine from the cold liquid oxygen. Special material combinations are used to prevent sparking, ignition and explosion due to rubbing in the strong oxidizing environment of the liquid oxygen.

- *The high-pressure fuel turbopump* (HPFTP) is also shown in Figure 4. Additional detail is afforded by the cross-sectional drawing of Figure 6. No inducer is needed due to the pressurization of the LPFTP and the more favorable vaporization characteristics of liquid hydrogen. The pump has three identical impellers and diffusers, except that the third-stage diffuser discharges into a scroll or volute, and the third-stage impeller also

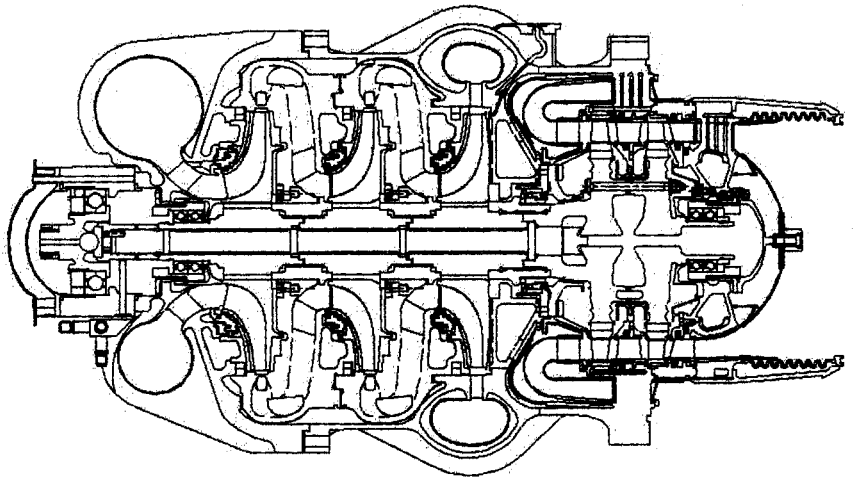


FIGURE 6 Cross-section of SSME high-pressure oxygen turbopump (Courtesy of The Boeing company)

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acts as an orificed balancing disk, which removes all axial thrust (except for a small spring-preload force) from the angular-contact, liquid hydrogen-cooled ball bearings located at each end of the pump-and-turbine package. The head per stage is over 65,000 ft (20,000 m) because of the high pressure rise and low density of liquid hydrogen. This leads to an impeller OD tip speed of nearly 2,000 ft/sec (600 m/s)—probably the highest in existence for a pump and successfully deployed in titanium, which has the benefit of a higher strength-to-weight ratio than the more commonly used steels. The high speed (over 37,000 rpm) keeps this 77,000 hp (57 MW) machine small and light and necessitates operation between the second and third critical speed.

THE RD-170 ENGINE

The RD-170 rocket engine was developed in the Soviet Union and is described in some detail by Sutton[6]. Propellants are liquid oxygen and an RP1 equivalent. Consisting of four nozzles, each of which produces about the same thrust as one SSME, this "four-engines-in-one" configuration is fed by a single propellant pump assembly—similar in concept to the F-1 turbopump. Referring to Table 13 of Section 2.1.1, the oxygen pump produces a pressure rise of about 8,500 lb/in^2 (59 MPa), which is undoubtedly the highest pressure rise of any centrifugal pump stage in the world, making it the highest-energy pump of Figure 32 in Section 2.1.1. Not far behind is the RP1 pump at 7,100 lb/in^2 (49 MPa). Assuming the pump efficiency to be 80% would yield a total shaft power for both pumps of about 230,000 hp (172 MW). At such high energy levels, as discussed in Section 2.1.1, the life of this pump must indeed be quite short. However, the design appears to have been successful. The only drawback is the environmental issue concerning the burning of a large quantity of hydrocarbon fuel, which is also an inhibition to further use of the F-1 engine.

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C • H • A • P • T • E • R • 13

INTAKES AND SUCTION PIPING

SECTION 13.1
INTAKES, SUCTION PIPING, AND STRAINERS

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The suction approach, or inlet, is the most critical part of a system involving pumps, whether in the form of piping or open pit. Improper design of the suction intake or piping can influence pump performance through efficiency and head loss, vibration, and cavitation. A centrifugal pump that lacks proper pressure or flow patterns at its inlet will not respond properly or perform to its maximum capability. The negative impact of improper flow at the pump inlet generally increases with the specific speed of the pump.

Flow control and the uniformity of flow to the point of pumped fluid contact with the impeller inlet vanes are the most important. Part of this may be controlled by proper pump design, but changes in flow direction, obstacles and changes in the geometry in the suction pit or suction piping are often main causes of improper flow quality at the pump inlet. In this respect the pit designer and suction piping designer have also definite responsibilities to achieve satisfactory pump operation.

In open suction pit (wet-well) designs, the fluid flow must be as uniform as possible right up to contact with the pump suction bell or suction pipe, preferably without a change in direction or velocity. Examples of dry-pit and wet-pit centrifugal pumps connected to open suction pits are shown in Figure 1.

In dry-pit pumping, the pipe leading to the pump suction flange should not include elbows close to the pump in any plane. Also, any other fittings that change flow direction and velocity and which may impart a spinning affect to the flow should be avoided within five to eight pipe diameters of the pump suction flange. Centrifugal pumps not designed for prerotation of the fluid entering the impeller, either dry or wet pit, may suffer loss of efficiency and an increase in noise if a spinning inlet flow occurs. Fluid rotation with the direction of impeller rotation can result in a decrease in pump-developed head. Fluid rotation against the direction of impeller rotation can result in an increase in pump-developed head and required power, possibly overloading the driver as well as drastically affecting the pump curve shape and performance in the system. If the total system is to operate efficiently and with minimum maintenance, close attention to the suction environment of the pumps is required.

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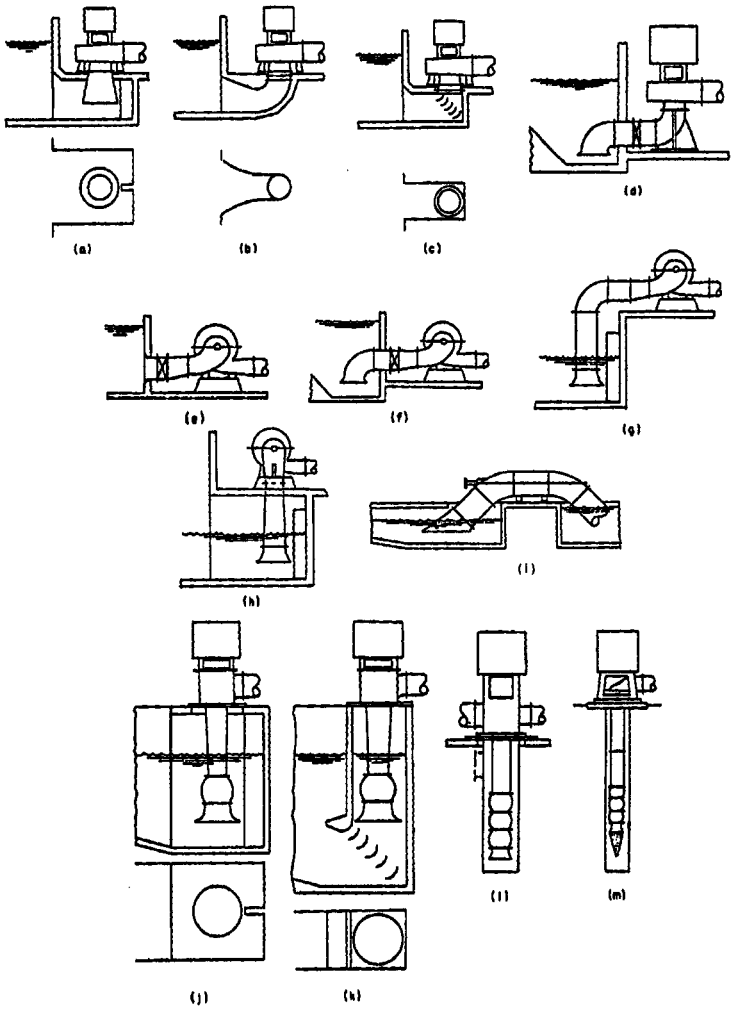


FIGURE 1 Pump suction connections: (a–d) vertical dry-pit pumps. (a) Suction bell diameter and distance from bottom, side, and back walls designed to provide radial inflow, equal velocity distribution, and minimum entrance loss. Proper spacing of bell and vertical baffle required to prevent underwater eddies and rotation at inlet. (b) To minimize excavation under pump, decreasing-type suction elbow, or formed suction intake can be formed in the foundation. (c) Width of suction inlet under pump minimized by use of turning vanes. (d) Smaller pumps provided with integral suction elbows and can be connected to wet well with commercial piping. Flared suction elbow is turned downward to obtain maximum submergence at inlet. Wet well sloped to suction inlet to prevent deposition of solids. (e–i) Horizontal dry-pit pumps. (e) Pump suction connected to larger pipe with eccentric reducer (double-suction pumps only).Wet-well connection should be bell mouth of a size to provide minimum required velocity. (f) Flared suction elbow turned downward for maximum submergence over inlet. Wet-well floor may be sloped toward inlet if solids are present. (g) Vertical suction pipe with bell inlet adequately submerged to prevent vortexing. Bell sized and spaced as in Figures 1a, h, and j. (h) Vertical suction pipe connected to bottom-suction pump. Bell sized and shaped as in Figures 1a, g, and j. (i) Design used for drainage over levees. Suction pipe designed to require minimum submergence and excavation. (j–m) Vertical wet-pit pumps. (j) Distance from bottom, side, and back walls designed to provide equal velocity distribution, with splitter to prevent vortexing. (k) Width of inlet minimized by use of turning vanes. (l) Canned suction pump with either above- or below-ground suction connection. Can length to suit NPSH required. Vertical installation minimizes excavation area. (m) Deep-well pump takes suction from underground water source.

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INTAKE STRUCTURES

Intake structures are used when the liquid to the pump comes directly from fluid bodies with free liquid surfaces. Their function is to provide a good flow quality, in terms of pressure and flow patterns, to the suction bell inlet of the pump.

The quality of the flow at the impeller inlet determines to which extend the maximum performance of the pump can be obtained. Improper design influences pump performance through efficiency and head loss, vibrations and/or the occurrence of cavitation. These effects can be caused by phenomena such as formation of subvortices, nonuniform velocity profiles at impeller inlet, and recirculation-and flow separation zones in the suction intake. In order to guarantee smooth pumping operation standard performance criteria have been defined to which a suction intake has to comply.

The inlet part of the impeller vanes are normally designed assuming a uniform velocity field that is free of rotation. In a well-designed suction intake the fluid enters the impeller inlet with variations of the velocity, that is, the flow distribution, less than 5% with respect to the averaged velocity. In this case also the time fluctuations of the velocities have to be limited.

The presence of swirl at any point in the flow just before the impeller inlet indicates that the fluid velocity consists of an axial component u and a tangential component u_t. Swirl angle α is defined as the angle between the resultant velocity and the axial direction. The amount of swirl at the impeller inlet should be limited, with a required swirl angle less than 4 degrees. The swirl angle can be measured with a neutrally pitched four bladed swirl- or vortimeter (Figure 6, Section 13.2). The allowable maximum number of revolutions per minute of this meter should be less than $1.7\ Q/D_1^3$ (see Reference 1, Hydraulic Institute Standards, Section 9.8.5.5).

Free-surface vortices (Figure 5, Section 13.2) are caused by improper intake design and/or insufficient submergence. Together with subsurface vortices, which may emanate from the floor, the ceiling and the back and side walls, they are only allowed when of low magnitude (see Section 13.2).

Insufficient submergence can also lead to failure at start up (when the impeller inlet is not filled with liquid), inadmissible amount of cavitation or the entrainment of air or gas bubbles in the pump. A pump may have adequate submergence from a pressure standpoint and still be lacking in sufficient depth of cover above the suction inlet to prevent surface air from being drawn in. Any wet-pit pump must have its suction inlet submerged at all times, and for continuous pumping every pump will have a fixed minimum submergence requirement. Because this relates to velocity, there are two basic parameters: the suction inlet diameter and the depth of water above the inlet lip. As pump size (and flow) increase, the inlet velocity may stay constant as the bell diameter increases, but at the same time, the impeller distance above the suction inlet becomes larger, so a fixed submergence value would lead to increased surface velocity, peripheral drawdown, and an increase in air intake.

Basically, submergence must of necessity increase with pump size. For the final determination, some balance must be struck between submergence and pit width to satisfy an average flow velocity of 1 to 1.25 ft/s (0.3 to 0.4 m/s) and maintain reasonable economic balance between excavation costs, concrete costs, and screen costs without neglecting ecological requirements and still fulfilling the primary need for circulating water in adequate quantities.

In order to obtain a maximum flow rate for a given difference in static head while reducing the available NPSH as little as possible, the head losses in the suction intake have to be confined to a minimum. The loss coefficient C_L of the suction intake is defined as:

$$C_L = h_L \cdot 2g/u^2$$

giving the relation between the head loss h_L and the dynamic head at the impeller entrance. Values of the loss coefficient of the suction intake alone (without trash racks and/or screens), may be expected to have values from 0.05 to 0.2.

The nonuniform velocity profile and/or swirl at the suction intake's outlet, and/or a high resistance factor of the (formed) suction intake can be caused from high velocities,

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improper flow guidance and/or high local decelerations of the fluid. These last two phenomena cause flow separation and the occurrence of secondary flows and/or dead corners.

Types of Intake Intake structures can be categorized as being for clear liquids or solids-bearing liquids. For clear liquids, intakes are further classified into rectangular, formed, circular, and trench-types, as well as suction tanks and cans. For solids-bearing liquids, trench-type and circular wet wells are usually considered. The different kind of intake structures will be briefly illustrated here; they are covered in detail in Reference 1.

RECTANGULAR INTAKE Probably the most common type of intake structure for vertical wet-pit pumps (or for dry-pit pumps having vertical suction pipe with a bell-mouth entrance) is the rectangular-shaped intake. For the design of such an intake, the recommendations found Reference 1 should be followed. The basic layout of the pumps and intake structure are shown in Figures 2 and 3.

The geometry is generally defined in terms of the pump inlet bell diameter, as shown. Once the number and size of pumps required is determined, a pump inlet bell diameter can be estimated. At this point, the bell diameter can be derived from the impeller inlet diameter or estimated based on an impeller axial inlet velocity of between 3 and 8 ft/s (0.9 and 2.4 m/s). The resulting impeller inlet diameter can then be converted to a corresponding bell diameter, approximately 1.5 to 2.0 times the impeller inlet diameter. With the bell diameter selected, the proportions of the inlet structure can be estimated from Figures 2 and 3. Table 1 gives recommended values for the dimensions. The minimum submergence S over the suction bell diameter D for vertical wet-pit pumps is a function of the Froude number F_D. Hydraulic Institute recommends using Hecker's formula:

$$S/D = 1.0 + 2.3\,F_D,$$

where F_D = Froude number = $V/(gD)^{0.5}$

D = suction bell diameter, ft (m)

S = submergence, ft (m)

V = suction bell inlet fitting velocity, ft/s (m/s)

g = acceleration due to gravity, ft/s^2 (m/s^2)

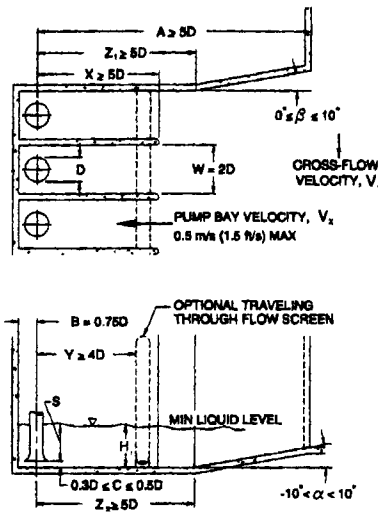


FIGURE 2 Recommended intake structure layout (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

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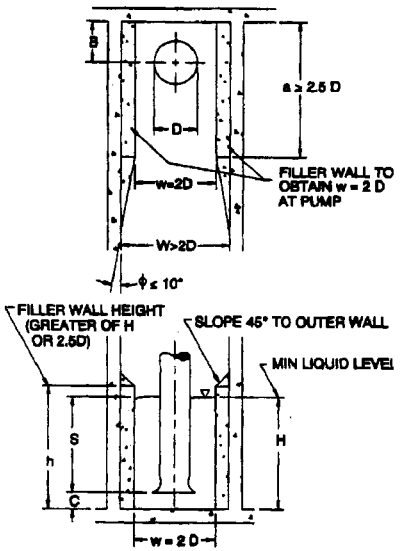


FIGURE 3 Filler wall details for proper bay width (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

For establishing velocities, the minimum submergence over the suction bell for vertical wet-pit pumps can be estimated from Figure 4, based on maximum expected flow rate. The graph is valid for suction pits that follow the standard dimensions indicated in Figure 2. Bottom clearance, back wall clearance, pit width, presence of antirotation baffles, and the velocity profile at the inlet of the suction intake are of critical importance in determining minimum required submergence.

The intake design process is intended to have an intake cross-sectional area average velocity of 1.5 ft/s (0.5 m/s) or less. It further assumed that the fluid approaching the intake with cross-flow velocities not exceeding 50% of the pump bay entrance velocity.

Any necessary change in inlet channel dimensions should be made gradually. Tapering walls should not diverge at more than a 14° included angle. If it is necessary to slope the floor, a maximum of 7° is recommended and, if possible, the floor should level off before reaching the pump area, as far back as possible. No sharp drops (waterfall effect) should be permitted.

If the inlet channel is a closed pipe with full-wetted perimeter, pipe velocities can be used up to a distance from the pump chamber where the tapered wall rule can be applied. This means that the pipe size would be increased in accordance with the rules for maximum taper rate to ensure that the velocity as it discharges into the pump chamber is not over 1.5 ft/s (0.46 m/s). Unless the inlet velocity into the pump chamber is kept below 1.5 ft/s (0.46 m/s), extensive baffling and additional space will be necessary, and this can only be effectively determined by analyzing the design modification with CFD and subsequently model testing of the pit.

TRENCH-TYPE INTAKES An effective way of decreasing the fluid velocity of a concentrated flow through a channel or pipe before the pump inlet is obtained with a trench-type intake. The trench-type intakes use a sudden transition in the vertical plane to guard the pump intake from the flow characteristics of the concentrated flow (when these changes would be in the horizontal plane often stilling chambers are used). The vertical transitions can be either in the wet-well intake structure or in front of the inlet of a suction intake. The maximum velocity in the approaching is limited depending on the direction of this flow relative to the direction of the well: 4 ft/s (1.2 m/s) when coaxial and 2 ft/s (0.6 m/s) when

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TABLE 1 Recommended dimensions for Figures 2 and 3 (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

Dimension Variable	Description	Recommended Value
A	Distance from the pump inlet bell centerline to the intake structure entrance	A = 5D minimum, assuming no significant cross-flow[a] at the entrance of the intake structure
a	Length of constricted bay section near the pump inlet	a = 2.5D minimum
B	Distance from the back wall to the pump inlet bell centerline	B = 0.75D
C	Distance between the inlet bell and floor	C = 0.3D to 0.5 D
D	Inlet bell design outside diameter	(see text)
H	Minimum liquid depth	H = S + C
h	Minimum height of constricted bay section near the pump inlet bell	h = (greater of H or 2.5D)
S	Minimum pump inlet bell submergence	$S = D(1.0 + 2.3 F_D)$
W	Pump inlet bay entrance width	W = 2D minimum
w	Constricted bay width near the pump inlet bell	w = 2D
X	Pump inlet bay length	X = 5D minimum, assuming no significant cross-flow at the entrance to the intake structure
Y	Distance from pump inlet bell centerline to the through-flow traveling screen	Y = 4D minimum. Dual-flow screens require a model study
Z_1	Distance from pump inlet bell centerline to diverging walls	Z_1 = 5D minimum, assuming no significant cross-flow[a] at the entrance to the intake structure
Z_2	Distance from inlet bell centerline to sloping floor	Z_2 = 5D minimum
α	Angle of floor scope	α = −10 to +10 degrees
β	Angle of wall convergence	β = 0 to +10 degrees (Negative values of β, if used, require flow distribution devices developed through a physical model study)
ϕ	Angle of convergence from constricted area to bay walls	ϕ = 10 degree maximum

[a]Cross-flow is considered significant when $V_C > 0.5\ V_X$ average

normal to each other. The width of the sump above the trench should be designed so that the velocity of the fluid above the trench has a maximum of 1 ft/s (0.3 m/s). The height of the trench should be twice or more than twice the bell diameters above the suction bell or twice the height of the suction intake to which it is connected.

For larger capacities of the individual pump (more than 40,000 gpm (2.5 m^3/s)) total intake structure (more than 100,000 gpm (6.3 m^3/s)) it is advised to conduct a model test of the intake structure.

FORMED SUCTION INTAKE For large circulation pumps it can be advantageous to use a dry-pit pump (or even wet-pit pump) with formed suction intake instead of a wet-pit pump with intake structure. From a fluid dynamics point of view the formed suction intake has certain advantages over the open pit intake. In cases where high demands are required to the pump material, as for instance with tropical seawater, it is often cheaper to switch

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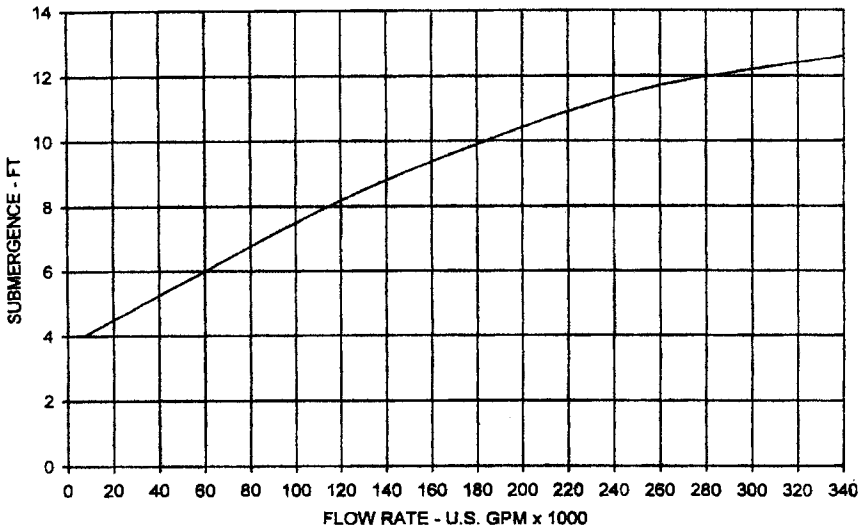


FIGURE 4 Flow rate versus submergence over suction bell for vertical wet-pit pumps (U.S. gpm × 0.227 = m^3/h, ft × 0.305 = m).

to dry-pit pumps instead of wet-pit pumps. The advantage is than that part of the high alloy metal in wet-pit pumps than be replaced by cheaper concrete (in the discharge piping and/or volute). Depending on the situation and demands of the pumping station, a choice will be made between an open-pit suction intake and a formed suction intake.

With respect to their function the formed suction intake has the advantage of the absence of dead water areas in which sand or silt can deposit. Further its performance is less sensitive to the direction and distribution of the flow at its inlet and it will not produce subsurface vortices. While during operation the minimum submergence required is often below the entrance level of the impeller, to get the pump priming at start-up, a water level that fills the inlet of the impeller is sufficient.

The most famous formed suction intake design is the TYPE 10 design developed by the U.S. Army Corps of Engineers of which the geometry is accurately described in Reference 1. Many of the major pump companies use their own developed formed suction intake designs. With the help of CFD it is nowadays very well possible to optimize the formed suction intake designs for certain design parameters, as there are minimum width, minimum length, minimum excavation, simplicity of shape, minimum hydraulic losses, or minimum submergence, without having penalties on their performance. As for all suction intake designs it is recommended to validate the performance of the designs by model testing.

An extreme case of optimization of a formed suction intake is described for a pumping station where the pumps and formed suction intake are situated in concrete columns that are part of a dam in a river. For the subject case a formed suction intake needed to be developed that is insensitive to debris and fluid mud and low water levels at the entrance. The spatial limitations were bounded by the dimensions of the column. As a solution a double entry formed suction intake has been designed. In Figure 5 a cross-sectional and three dimensional impression of the design are shown. Besides the fact that the design is very compact it also shows that it is not fully symmetrical, due to the fixed position of the pump in the column. The intake has been designed and validated by means of CFD and model tested in air as well on full scale on site, both to fully satisfaction. A minimum submergence of S/D=0.25 has been met.

A wet-pit intake style that closely approximates a suction pipe arrangement and uses what is essentially a dry-pit pump has been expanded by the U.S. Department of the Interior, Bureau of Reclamation to an elbow-type suction tube design. This incorporates a

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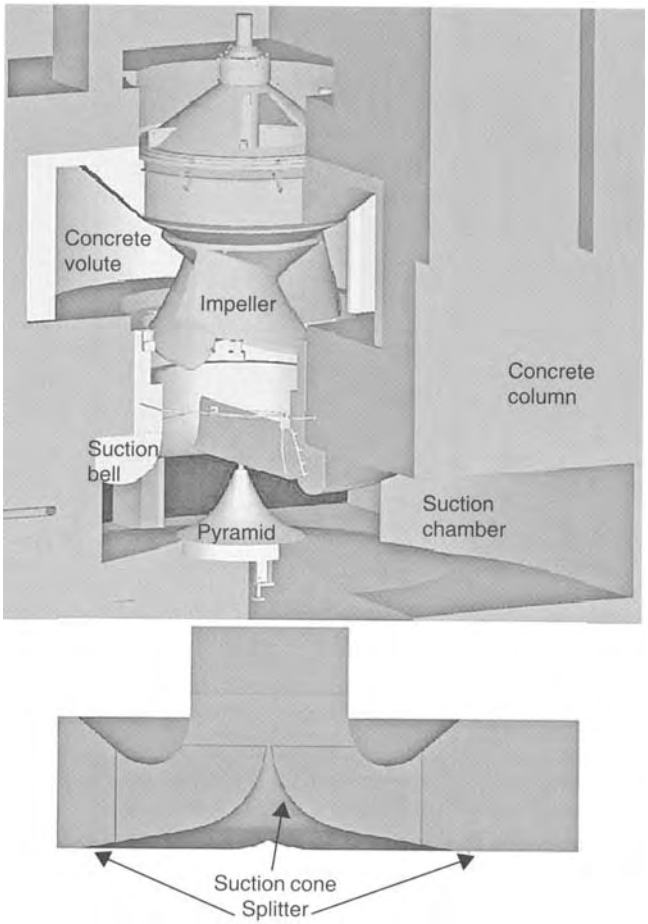


FIGURE 5 Design of an asymmetric double entry formed suction intake for pump inside concrete column which is situated in a river dam (Flowserve Corporation)

formed concrete suction inlet with a swing of 135° in the vertical plane and a gradual decrease in area to the suction eye of the pump (Figure 6). The resulting design saves considerable excavation in the wet-well area, reduces losses in the pump, and allows a smaller, higher-speed pump to be used. With higher velocities, debris dropout is reduced so silt buildup does not occur as readily and a smaller trash rack area remains effective longer. These inlets are independently self-sufficient and may be grouped into multiples as long as the forebay is designed to adhere to the basic design rules for wet-pit approach channels. Additional guidelines on the proportions of formed suction intakes are included in Reference 1.

SUCTION TANKS There are a variety of shapes and configurations for suction tanks. They may be vertical or horizontal, cylindrical, or rectangular. Outlets from the tanks may be vertically downward, horizontal, from either the side or bottom of the tank, or in the case of a pressurized tank, vertically upward. The outlet fitting of the discharge pipe of the suction tank can be straight, cone or bell shaped, either flush with or protruding through the

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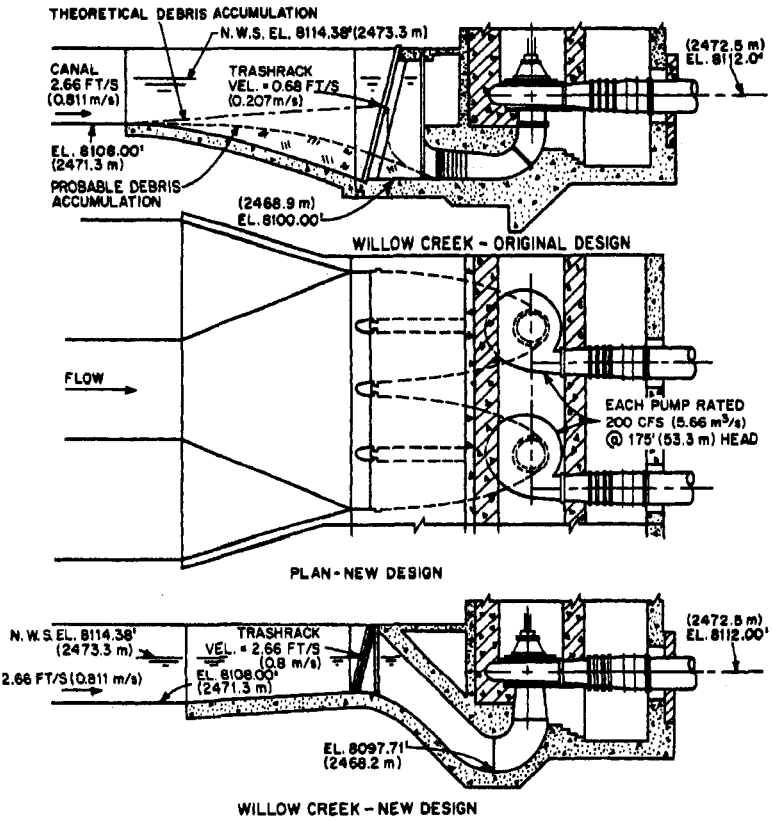


FIGURE 6 Improved 135° design of Bureau of Reclamation elbow suction tube inlet (U.S. Department of the Interior, Bureau of Reclamation)

tank interior surface. The formation of vortices in the tank can cause air or gas entrapment even when the tank outlet is fully submerged. This can cause pulsating pump flow, noisy operation, or a reduction in pump performance.

To determine minimum submergence of the outlet pipe in the tank, the Hydraulic Institute again recommends the following relationship:

$$S/D = 1.0 + 2.3\,F_D,$$

where F_D = Froude number = $V/(gD)^{0.5}$

D = outlet fitting diameter, ft (m)

S = submergence, ft (m)

V = outlet fitting velocity, ft/s (m/s)

g = acceleration of gravity, ft/s^2 (m/s^2)

The submergence S is the difference between the surface level and the centerline of a horizontal outlet or the level of the very beginning of a vertical oriented outlet pipe. The diameter D and the fluid velocity V are to be considered at the entrance plane of the straight, cone or bell-shaped outlet. If the tank has multiple inlets or outlets, the design should be such that flow interaction is not detrimental to overall performance.

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UNCONFINED INTAKE In case of unconfined intakes there is no structure around the wet-pit pump inlet to guide the flow. This is mostly seen with pumps placed on platforms in open water. Unconfined intakes give the best performance when unidirectional cross flow occurs at velocities less than 25% of the bell velocity. Obstructions to the pump suction that cause wakes should be located more than five times the cross sectional of the obstruction. The suction bell is best located 0.3 to 0.5 suction bell diameters above the bottom situated. In case of bottom debris this value can be increased.

CIRCULAR INTAKE The simplicity of shape, ease of construction, and compactness of a circular shape intake structure make this type of intake one of the most popular in the industry for both clear and solids-bearing liquids. Standard layouts of circular intakes are available for duplex and triplex sumps with pumps centerline or offset, dry or wet pit (see Reference 1 for further guidance). Intake designs with four or more pumps and/or flows exceeding 315 L/s (5000 gpm) require a model test.

A special type of circular intake is the Ranney well (Figure 7): a concrete silo 13 ft (4 m) in diameter, which becomes the collecting basin and pump well. It may be situated in or near a river and can be partly below and partly above water level with settings up to 100 ft (30 m). Small perforated pipes radiate horizontally from the base of the well and tap into porous strata, bringing small flows of water into the main well. This water can then be pumped from the well with a deep-well, wet-pit vertical pump. In the Ranney well, the

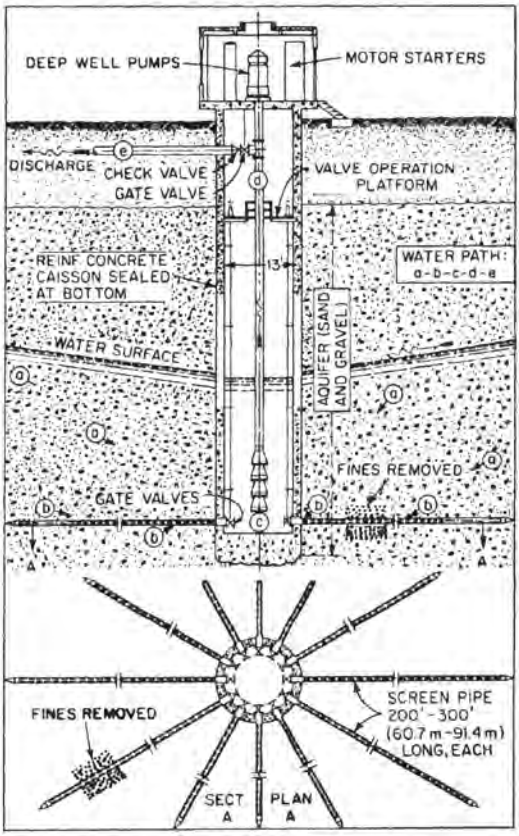


FIGURE 7 Ranney well collects into a pool from underground strata (Ranney Method Division, Pentron Industries)

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ratio of pump size (and flow) to pit size (and capacity) is such that very low velocity exists, as in a lake inlet. Water comes into the Ranney well all around the periphery. These conditions of direct flow and low velocity prevent vortexing.

Intakes for Solids-bearing Liquids Where most pump types must be prevented from entering debris and solids into the pump, there are also pump types that are made for pumping solids-bearing liquids like wastewater, polluted drainage water or discharges produced by industry. In case of wet-well intakes, care must be taken that the solids do not get too much accumulated, floating or settled, and that they can easily be removed.

In order to minimize the accumulation of solids in the intake structure the main idea is to minimize the presence of horizontal surfaces in the wet well. By positioning these horizontal surfaces only in the direct surroundings of the pump inlet, it is granted that the highest amount of solids will be sucked into the pump. The transition walls around the pump inlet area shall have steep angles (depending on the roughness of the wall material, larger than 45 to 60 degrees) to direct all solids in the direction of the pump inlet. Two types of intakes that perform well with solids-bearing liquids are trench-type and circular intakes, both modified with minimized horizontal surfaces at the pump inlet and sloping walls around.

Cooling Water Pump Intakes Cooling water circulating systems must have either a continuously renewable source, such as an ocean, lake, or river, or they must recirculate the same water from cooling ponds or cooling towers. Regardless of the type of pump selected (wet or dry pit), the suction water will come from an open pit of some sort or from a pressurized pipeline. Cooling water intake designs are categorized depending on the intake source: cooling pond, cooling tower, or ocean/lake/river source.

ONCE-THROUGH: OCEAN, LAKE, OR RIVER SOURCE—WET PIT Once-through intake structures are usually constructed of concrete and are arranged to gather the water into a localized area for pickup and to support the pumps. The optimum design will bring relatively clear water directly into the pump suction area at a low velocity.

For further guidance, Reference 1 provides intake design recommendations for both suction pipes and all types of wet pits that are a result of the combined efforts of sump designers, hydraulic researchers, pump manufacturers, and end users. It is intended to provide designers, owners, and users of pumping facilities a foundation upon which to develop functional and economical pumping facility designs. The choice of providing a trash rack or screen, or both, is based on the type and amount of debris likely to be encountered at the inlet.

An ocean inlet has additional requirements because of tidal action and variable direction currents that may exist. For these situations, it may be necessary to create a forebay inlet basin. Such a basin would be independent of the pump pit and fed by a submerged inlet tube extending out into the ocean for some distance (Figure 8). This tube may utilize normal pipe velocities, but the inlet must turn upward and the opening should be protected by a horizontal cap to allow the water to travel horizontally at a velocity high enough to scare fish away. This will prevent fish from being subject to a vertical velocity, which they are unable to notice, and will allow them to maintain a horizontal velocity that will direct them away from the inlet. The discharge into the forebay can be pipe velocity because the turbulence will be dissipated by lower outlet velocities [2 ft/s (0.6 m/s)] through trash racks and traveling screens. From this point onto the pump chamber, the velocity should be low and constant.

RECIRCULATING SYSTEM—COOLING TOWER For cooling tower systems, the pump pit is normally adjacent to the tower basin. Cooling tower basins are shallow, usually not more than 6 ft (1.8 m) water depth. Because pumps will require more submergence than this, a sloping ramp or sharp drop-off will be required if pumps are to be placed close to the basin. This will save initial costs for excavation and concrete work, but the resulting poor flow patterns conditions may result in endless pump problems.

Some successful small installations have been built with downslopes of 15° or more, but tower basin outlet velocity has been no more than 2 ft/s (0.6 m/s) and the

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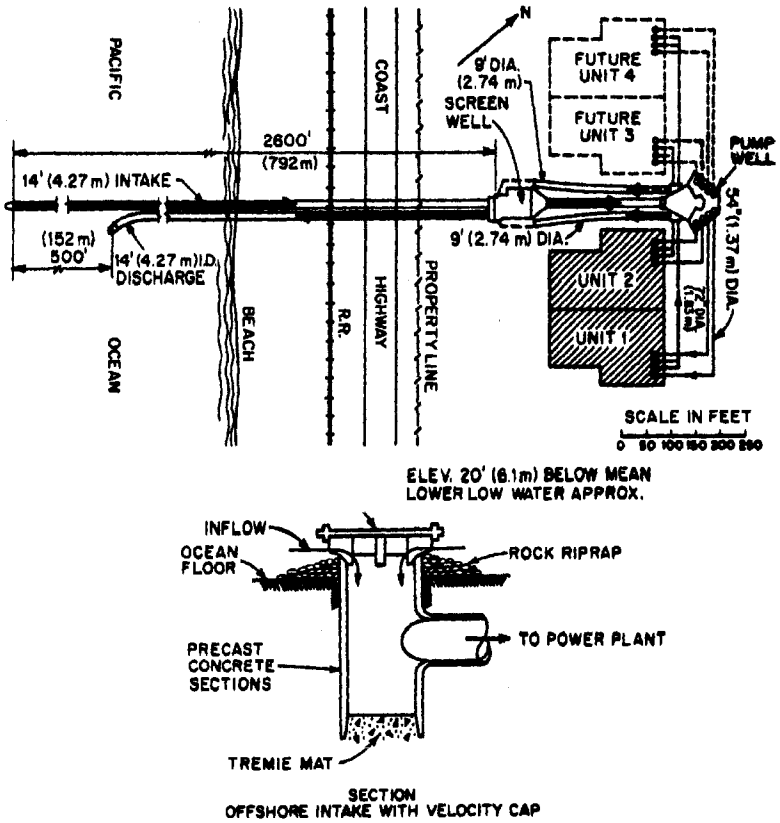


FIGURE 8 Ocean intake with velocity cap to minimize fish pickup

downslope widened as it approaches the pump pit. For low flows, low velocities, and fortunate arrangements of tower piers (lack of recirculation piping, and so on), such installations may be operated with reasonable success. Others of similar design, however, have proven to have disastrous effects on pump performance. Impellers and bearings may suffer rapid deterioration under such conditions.

The best solution, but obviously the most expensive solution, is to build the pump pit far enough from the tower basin to obtain a flat channel bottom. The channel should have an average channel velocity of 1 ft/s (0.3 m/s) or less for a distance at least equal to A in Figure 2, whether trash racks and screens are available. The slope from the basin floor to the channel bottom should not be more than 10°, preferably less than 7 degrees.

The basin exit width should be such as to make the exit velocity less than 2.5 ft/s (0.75 m/s). If several tower piers are in the exit path, this velocity limit should be reduced to 2 ft/s (0.6 m/s) or less. Upstream sides of the piers should be rounded off and the downstream sides should be tapered. Make the sidewalls of the downslope diverging so the velocity at the bottom is no more than 1 ft/s (0.3 m/s).

Velocities should be based on the runout flow of one pump. All dimensions and velocities mentioned are limits. If they are used based on maximum required flow rate, and the pump is selected for reasonable velocities, operation should be satisfactory. Obviously, if a pump with high velocities (suction bell inlet, impeller eye, rotational speed) is selected, it

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will require more submergence than a more conservatively designed machine. This will increase the cost by requiring a deeper pump pit located further away from the tower basin in order to perform satisfactorily.

A model test of the inlet structure is highly recommended if the best solution seems too expensive, or if topography will not allow the ideal arrangement. A model with a variable slope, variable sidewalls, and true representation of piers, pipes, and other obstructions should enable a satisfactory design to be achieved with a possible net reduction in overall cost. Actual pumps and valves should be used to create the flow in the pump pit, and extremes of flow and submergence should be included in the test program. Sometimes the cooling towers are on a hill, so a substantial drop in elevation provides pressure available nearer the flow area. If dry-pit pumps are used, this pressure can be utilized to reduce pump requirements. When this is done, the pump suction becomes a pipe, and the latter part of this section should be consulted.

RECIRCULATING SYSTEM—POND WET PIT The intake structure in a cooling pond should be located as far as possible from the inlet pipe to the pond to generate the maximum cooling effect. If spray surface equipment is used, it should be so arranged in relation to the intake building that a minimum surface disturbance is encountered. Prevailing winds should be considered, and the building should be located on the lee (upwind) side of the pond. If side and bottom areas might be easily disturbed to include silt in the flow, riprap—a wall or covering of stones thrown together randomly—should be applied to approach slopes as well as to bottom mats well beyond the inlet wing walls.

The handling of silt is usually not desirable in a pumping system. A high velocity through the pump will accelerate wear. At low velocity points in the system, silt will settle out and produce higher velocities, and more wear will occur as a result of the area blockage. If space is available, a silt-settling basin can be constructed ahead of the inlet basin. Cross baffles should be provided to slow the inlet flow to a velocity less than 1 ft/s (0.3 m/s). Most stream debris will settle out at this velocity, and the flow into the pump suction pit will be relatively clean, preventing the deposit of additional silt around the pump suction bell. If space is not available for silt beds in large-capacity installations, the main channel can be furnished with a weir across the flow path. The height of the weir should be selected to give an overflow depth above the weir of not more than one-third to one-fourth the water depth just preceding the weir. The velocity over the crest should not exceed the intake channel velocity. Weirs of this type are particularly effective when the intake channel is at right angles to the supply mainstream.

Multiple Pumps Some pump requirements are easily met with 100% capacity single pumps, but more reliability usually requires two 50% or three 33% pumps, or, if the service is sufficiently critical, three 50% or four 33% pumps, and so on. There are practical limits of size for various pump types, so large flow demands will undoubtedly call for a multiple pump arrangement.

The problem arising from multiple pumps arranged in a common pit is from the probable nonuse of some of the pumps while others are operating. This can cause variables in flow patterns that may lead to eddying and vortexing (Figure 9). Installation of separating walls in the common pit may introduce additional problems because the ends of the separating walls can create eddy currents in the corners at the unused pumps. A back vent in the dividing wall will relieve this situation, provided it vents at the water surface (Figure 10b left). If walls are extended past the screens and trash racks to a forebay, this problem will not occur, but the design has then become that of a single-pump basin.

The same velocity rules apply to multiple arrangements as to single-pump basins. Odd arrangements should be avoided even when they look invitingly symmetric—fan-shaped, round, radial, peripheral—all have directional problems that are not easily overcome. A basic pit design, consisting of a number of equally sized pumps in a common pit basin with flow entering parallel and straight in at 1 ft/s (0.3 m/s) or less, would not need to be model tested to assure reliability (Figure 10a left). If separating walls are required for structural support, and they are properly shaped and vented, no model test is recommended. The pumps should be located at the extreme rear of the pit so the whole approach assumes the characteristics of a suction pipe. Individual pump manufacturers may vary the location of

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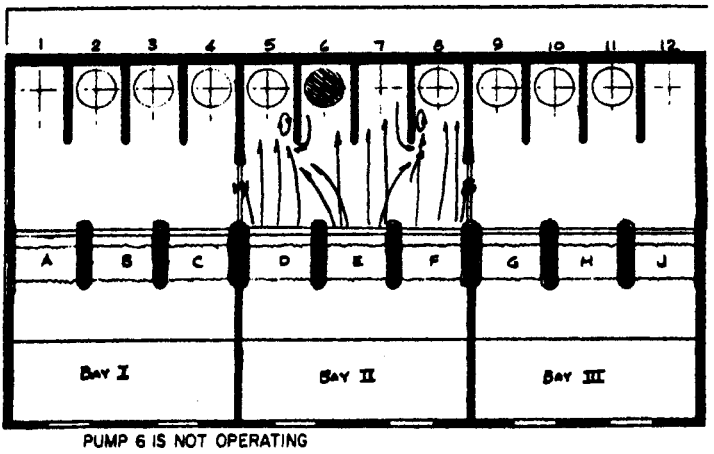
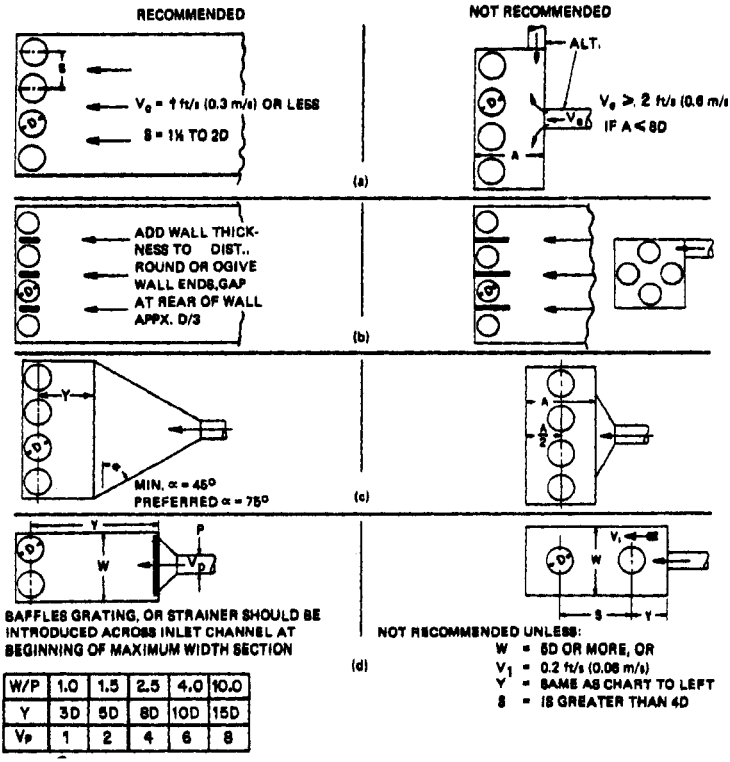


FIGURE 9 Empty spaces on nonoperating pumps cause vortexing at wall ends because of flow reversal



W/P	1.0	1.5	2.5	4.0	10.0
Y	3D	5D	8D	10D	15D
V_p	1	2	4	6	8

FIGURE 10 Using basics of good pit design precludes the need for model testing (Hydraulic Institute Standards, 13th Edition, 1975—out of print)

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the pump relative to the pit bottom, velocity of inlet spacing, and so on. It has been found that some of these variations require additional splitters or baffles below the pump, up the back wall behind the pump, or centered in the flow ahead of the pump. If so, a model test should be run and the additional pit cost weighed against other alternatives (changing the pit shape, pump location, pump size, and pump speed).

A dry-pit pump installation will have the pumps located either in a dry well at or below wet-well water level (Figures 1e and 1f) or directly above the wet well and using a suction lift (Figures 1g and 1h), which calls for priming equipment. The additional cost of priming equipment (vacuum pumps, and so on) may be partially offset by the additional space and valve requirement of the first option. In either case, the suction piping in the wet well should be treated in the same fashion as the pump suction bells in wet-pit installations as far as spacing, direction, and velocity of flow are concerned. When the pump is installed at an elevation that may be below the water level in the suction pit, a valve must be installed at the suction of the pump. The temptation to reduce the inlet size to use a smaller valve should be avoided. Pipe size at a pump inlet may decrease into the pump down to the pump suction size, but it would not be reduced below that size, and then have to increase again as it enters the pump. The pipe in the wet well should preferably have a bell end and project downward. The minimum water level above the lip of the bell can be estimated from Figure 4 for a recommended entrance velocity of 5.5 ft/s (1.7 m/s). The bell mouth should protect downward to assure uniform inlet flow and to attain maximum submergence.

Considerations for either single- or multiple-pump pit design relate primarily to even flow and low velocity into the pump. At the leading edge of the pump impeller (suction) vanes, this velocity will be increased and the direction of flow violently changed. To make the transition from pit flow to pump flow is the work of the pump designer. Some pump designs include ribs in the suction bell; others do not. It is obvious that the transition from parallel flow at 1 ft/s (0.3 m/s) to right-angle rotating flow at 15 to 20 ft/s (4.6 to 6.1 m/s) requires a high degree of skill in matching the suction bell to the impeller. If the pump is not designed to handle a wet-pit installation as described in previous paragraphs, a turning vane pit may be required.

Suction Pit Turning Vanes Figure 1c illustrates how turning vanes are used to guide the flow of water into the inlet of a vertical volute dry-pit pump. Figure 1k shows the same for the suction bell of a vertical diffuser wet-pit pump. If these pumps were connected to an inlet sump without turning vanes, as illustrated in Figures 1a and 1j, a wider and longer channel would be required to feed the suction bell from all directions. The pumps with turning vanes a narrower inlet channels may be desirable as multiple pumps can be spaced closer together providing screen width requirements do not dictate spacing.

Comparing depth of pit bottom below pumps with and without turning vanes, the following is to be noted. The excavation beneath a vertical volute dry-pit pump with turning vanes can be slightly less than the excavation beneath the same pump with no vanes but with a suction bell. The velocity approach into the closed portion of the channel beneath the pump can be as high as 3 ft/s (0.9 m/s) if turning vanes are used at the design flow; but it should be limited to 1.5 ft/s (0.46 m/s) with no turning vanes. Although the suction bell design requires a wider channel than a vaned inlet (approximately two-bell diameters), this is not wide enough and the channel must be made deeper to meet the lower velocity requirement for this type of inlet. When turning vanes are used with a vertical diffuser wet-pit pump, the pit must be excavated deeper than would be required if no vanes were used. This additional depth is required to form an elbow in the narrower channel and provide equal flow distribution to the impeller. Design velocity at the inlet vanes is 3 ft/s (0.9 m/s).

As an alternative to the use of turning vanes under a vertical volute dry-pit pump, the long-radius suction elbow inlet illustrated in Figure 1b offers some advantage in reducing the width and depth of excavation under the pump. The inlet velocity to the long-radius elbow, which is usually formed in concrete, is preferably no greater than 3 ft/s (0.9 m/s) at the design flow rate.

A decision to use turning vanes should not be based on a guarantee that there will be an increase in pump efficiency. The design of the vanes—their number and spacing—is still an art more than a science, and it is difficult to prove pump performance in the field.

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Turning vanes can be effective in eliminating underwater vortices, a problem sometimes associated with suction bells without turning vanes. It has been observed during model testing that the suction bell, as illustrated in Figure 1a, must be placed closer to the back wall than normally recommended for open channel inlets (similar to that in Figure 1j), to prevent underwater vortexing. The flow of water into a closed channel from an open pit containing water of considerably greater depth creates an unequal flow pattern in which the maximum velocity is along the floor. Also, there is little, if any, flow to the backside of the suction bell down from the top of the channel. Unless the bell lip is close to the back wall, flow along the floor and from the front only will overshoot the inlet and roll over, back, and up into the bell, forming an underwater vortex. Although turning vanes can prevent this, they may not prevent uneven flow distribution up into the pump impeller unless they are properly designed. They may even cause hydraulic and mechanical unbalance, which could result in noise, vibration, and accelerated wear of the pump bearings. For this reason, model testing of the turning vanes is recommended.

Environmental Considerations A pumping station has impact on different environmental parameters as it may:

- Change the water quality
  - Increase in temperature
  - Decreases the oxygen content
  - Change the clarity
  - Pollution by greases and/or lubrication oil
- Kill inhabitants living in the water
- Produce noise
- Change the hydrology and hydraulics

Off all these parameters, the suction pit design mainly influences the amount of inhabitants that travel though the pumping station with the chance of experiencing undue disturbance and destruction. The suction pit should be designed in such a way that it prevents as much fish as possible from entering the intake and the pumping system.

The most logical choice is to select a site for an intake not near feeding areas for large schools of fish (kelp beds, coral reefs, and similar attractive spots). Sheltered spots most suitable for intake flows may also be most attractive to fish. If the source is a river, the angle of the intake structure relative to the direction of flow is important in modifying the impact of these design requirements. Placing the screen house at a low angle to the river flow (instead of the usual 90° inlet) allows the river current to provide a swim-by attitude for fish while a low velocity screen approach is maintained. If the pump house is in line with the screen house, the pumps will feel minimal disturbance. A tendency for eddies to form (and create vortices) can be minimized by placing wing walls upstream and downstream to control velocity.

As an alternative to keep the fish away from the pump intake, remote intakes might be considered as an ocean outfall with velocity cap (Figure 8) or a Ranney well (Figure 7).

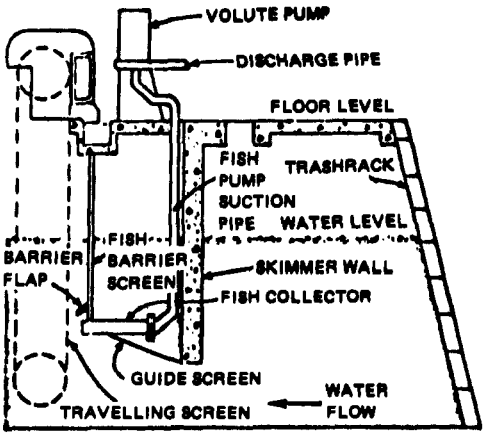
A design to accommodate fish limitations was mentioned briefly in a previous paragraph. Fish react to a horizontal velocity but are not aware of a pull in a vertical direction. Thus, to keep them from entering the inlet, a horizontal flow must be established at a velocity low enough to permit fish to escape. Inlet screen areas should be in small sections rather than one long face so a fish will not be trapped in the center and find it too far to swim to safety after it realizes its predicament. The maximum deterrent flow is about 10 ft/s (3 m/s), but this may be too much disturbance for the flow to the pumps. Smaller areas allow short-term limits for enticing fish away from the inlet, and survival will be much higher.

Fish congregating at an inlet or in a forebay pool can be crowded or herded to an outlet point by the use of vertical nets or horizontal screens. In a direct channel, horizontal moving screens can route fish past a sloping (relative to stream flow, say 35 degrees) moving screen. This results in directing fish to a narrow outlet at one side, leading to an outlet channel away from the main inlet.

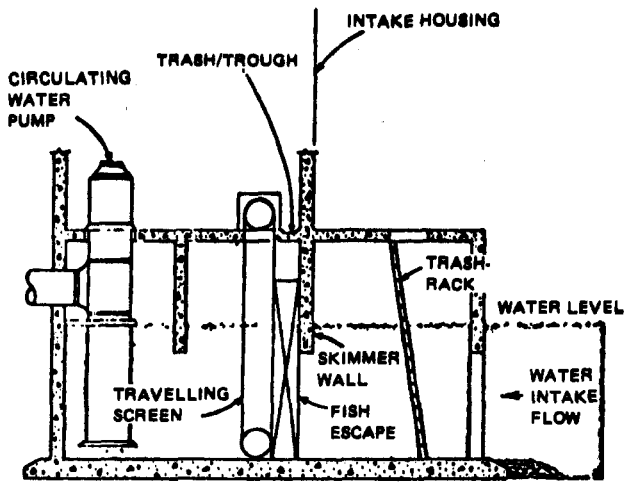
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The most difficult problem to overcome is related to small fish. Screen openings must be held to a minimum, and under velocity conditions, small fish have much less swimming-sustaining ability than larger fish, both in speed and in duration time. In a given steam flow (such as is generated by pumps with inlet water going through screens), a fish must have the ability to sustain a given speed against this flow for a certain length of time. When it weakens, it will fall into the current flow and will be impaled against the screen and destroyed. If the fish senses the velocity early enough, and has an alternate route, it can use darting speed to escape. Or it can follow another attraction (cross velocity flow into a separate chamber or a light attraction to the chamber) and be removed on an elevator or pumped out to a safety channel (Figure 11). Migrating fish need a continuation channel to restore their interrupted journey.

In designing an intake, it is necessary to keep the velocity below 0.5 ft/s (0.15 m/s) through the screen to avoid drawing fish into the screen. Alternatively, a cross flow can



(a)



(b)

FIGURE 11 (a) Fish pump prevents entrapment on traveling screen (Detroit-Edison). (b) Fish escape allows fish to bypass screen area.

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be created that will propel or attract the fish to one side of the inlet area. From there, they can be directed into a bypass pool and lifted back to their own living area, or they can be sent around the plant to a downstream location. If they are anatropous, they can be sent to an upstream rendezvous. Piers and screens should be kept flush across their inlet face to prevent attractive pockets where fish can hide and be drawn into the screens when they weaken.

Intakes that take their flow directly from a river may have a high velocity that would trap fish. Even if the velocities are lowered to reasonable screen levels—2 ft/s (0.6 m/s)—fish may still be drawn into the screen area and carried up to trash disposal. Obviously, trash and fish must be handled in separate areas, and screen wash pressures must be lowered to prevent harm to the fish.

Small slot-width wedgewire screens (Figure 12) are now being furnished in larger sizes, and, with their very low inlet velocity and backflushing capability, offer one good approach to reducing entrainment. Sand, chemical, and cloth filters may suit some situations, but backflushing and cleaning problems make them less attractive costwise.

Entrainment of marine organisms too small to be restrained by normal screens causes further problems in areas where such organisms normally develop. If the intake structure must be located in such an area, extensive information must be gathered at the site. Analyzing as much data as are available, in conjunction with plant flow and location requirements, may give the designer some idea of which equipment to select.

CFD Analysis and Model Testing of Intake Design When the basic rules for good pump suction pit design are adhered to, no model test will be required to ensure proper operation of the pumps and pumping system. The substance of these rules is to keep a straight-in approach at a constant low velocity from the water source to the pump chamber. The ANSI/Hydraulic Institute Standard dimensions and charts satisfy these criteria for the average pump in general application.

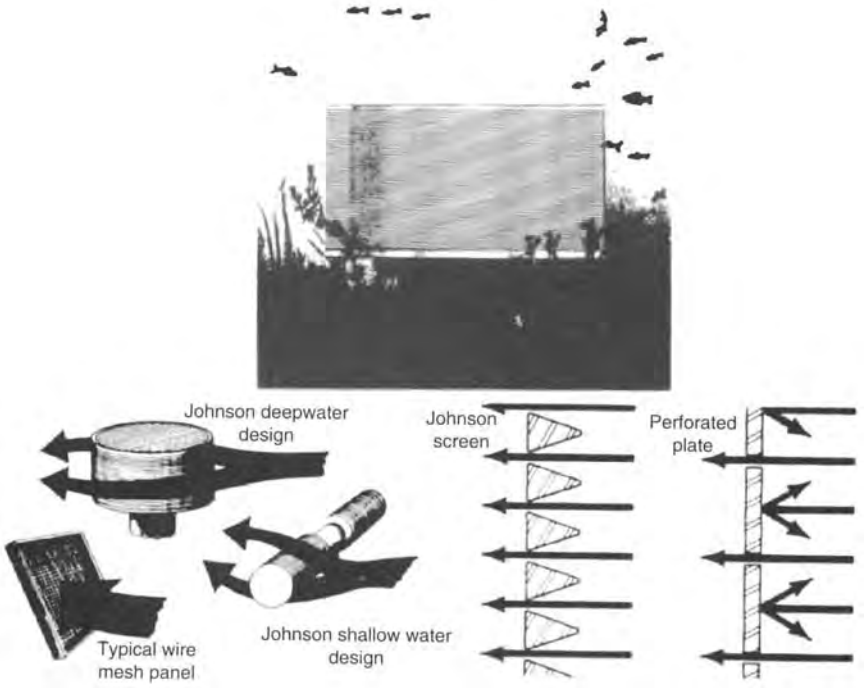


FIGURE 12 V-shaped slotted screen provides velocity control to avoid attracting fish and good air or water backflushing for debris cleanup (Johnson Division, UOP)

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Site layout problems may make the ideal solution impossible. Structural and environmental requirements may outweigh hydraulic requirements in some instances. When the ideal pit may not be possible or economically feasible, CFD analysis is very convenient to optimize the design within the given limitations. It is recommended that the outcome of such study be verified with a physical model test. It should be noted that the ideal dimensions are a composite covering not only a range of specific speeds but also a complex melding of pump design philosophies. Some variation from ideal dimensions should be expected from individual pump manufacturers.

Pump manufacturers are not in a position to guarantee the pump pit design. Differences of opinion between the structural and hydraulic pit design engineers and the pump design engineers may best be resolved by performing CFD analysis and/or a model pit test (Figure 13). Refer also to Section 13.2.

Vortexing The real problems resulting from improper pit design occur largely on the water surface in the form of vortices, or cones, produced by localized eddies on the surface of the water. If this disturbance continues, the flow of water will carry the underwater part of the vortex down toward the pump suction bell and ultimately into the pump (Figure 14). This introduces air into the impeller and will affect the mechanical radial balance of the impeller by interrupting the normal solid-liquid flow pattern. This type of disturbance will produce hydraulic pulsations in the pump flow and mechanical overloading of bearings and impeller guides.

Underwater vortexing sometimes occurs in round pits or in pits where the pump suction bell is at some distance from the rear wall. Flow past the suction bell strikes the rear wall and rolls back toward the bell, wall, and rolls back toward the bell, forming an eddy current that disturbs the normal flow into the pump. In a round pit, a cross baffle below the pump bell may reduce this effect. Where the pump is some distance from the back wall, a wall can be installed near the pump, or a horizontal baffle at suction bell level behind the pump will also reduce the disturbance. In all cases, the distance between the suction bell and the bottom of the pit should not be more than one-half the suction bell diameter, and one-third the diameter is preferable.

The use of the suction bell diameter as a basis for spacing should be carefully evaluated. It will be seen that there is nothing magical in this relationship, especially when several pump manufacturers all use different bell diameters. The real criterion is the allowable velocity at the suction bell. It has been found that very-low-head pumps are



FIGURE 13 A typical suction bay intake model test (University of Kaiserslautern)

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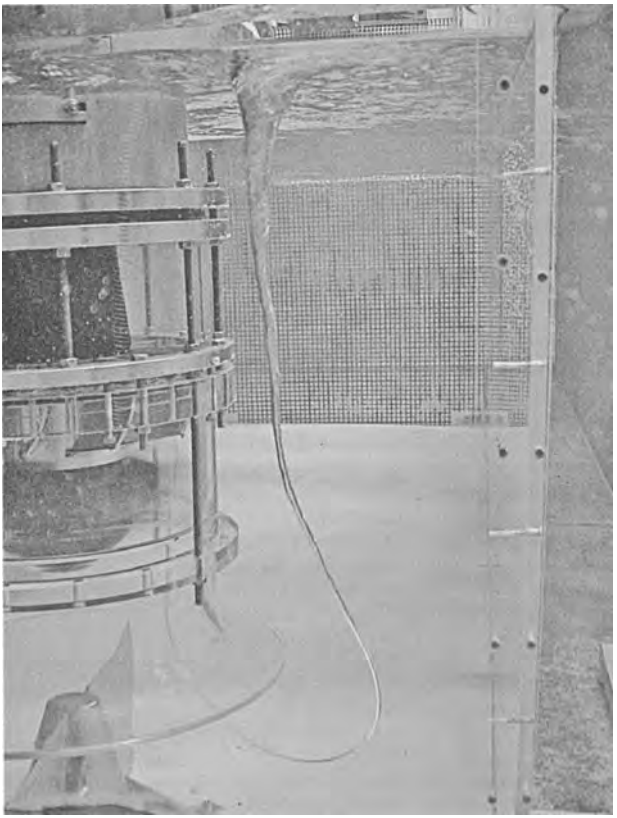


FIGURE 14 Surface vortex drawing air into pump suction bell (University of Kaiserslautern)

much more sensitive to bell velocity over 6 ft/s (1.8 m/s). For example, the velocity head loss at the bell inlet with a high velocity may be such a large percentage of total pump head that efficiency could drop as much as 10%. A good design rule for safe operation can be related to pump head. For pumps having up to a 15-ft (4.6-m) head, the suction bell velocity should be held to 2.5 ft/s (0.76 m/s); up to 50-ft (15-m) head, 4 ft/s (1.2 m/s); and above 50-ft (15-m) head, 5.5 ft/s (1.7 m/s). These values should be used for any substantial amount of pumping, but for occasional short-term pumping, they can be exceeded without destroying the pump.

Vortices may be broken up and effectually nullified by arrangements of baffles and vanes, or they may be prevented from occurring initially by a proper pit design. To determine what type of baffling should be used, CFD can be of help as it can predict the origin of vortices. To fully predict the effectiveness of the proposed baffling, model testing is needed. Methods of eliminating vortexing are discussed in Section 13.2.

Vortices are usually generated when the flow direction of the liquid to be pumped changes or when there is high velocity past an obstruction, such as a gate inlet corner, screen pier, or dividing wall. In combination, these two causes invariably generate vortexes. For this reason, the pump suction pit should be immediately preceded by a straight channel in which the velocity does not exceed 1.25 ft/s (0.4 m/s). Satisfactorily operating pump pits with higher velocities are rare and should not be put into operation without the assurance of a model study.

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An additional condition likely to generate vortexes is a multiple pump pit with individual cells in which only a portion of the pumps will operate simultaneously. The dead space behind the nonoperating pump will have flowing water tending to reverse direction and form eddies. Eddies and vortices can be avoided by eliminating or venting the walls at the rear of the pit. It also helps to position the pumps at the extreme rear of the pit. Alternatively, expensive modifications to the pit, such as splitter walls and baffles, may be required (Figure 9).

Round pits tend to generate vortices, especially when the pump is centered in the pit. These vortices usually occur around the pump column because of eccentric inlet flow. Special cases of the round pit are tolerable either when a Ranney well (Figure 7) is used or when there are booster pumps in the pipelines (Figure 15). In the Ranney well, the ratio of pump size (and flow) to pit size (and capacity) is such that very low velocity exists, as in a lake inlet. Water comes into the Ranney well all around the periphery. These conditions of direct flow and low velocity prevent vortexing. Booster pumps installed in a circular can (suction tank) must be centered in the can, and all inlet velocities to the can and flow in the can and into the pump must be uniform and high enough to provide fluid control. This velocity will vary from 4 to 6.5 ft/s (0.76 to 1.98 m/s).

Vortices are not generated by a pump or pump impeller and so do not fall into clockwise or counterclockwise rotation because of the pump rotation. Also, in a pump pit, vortices do not have a directional rotation induced by the rotation of the earth and therefore are not of opposite rotations above and below the equator, as is the case with "bathtub vortexing," which occurs in tanks being drained without pumps.



FIGURE 15 Booster pump suspended in a steel well (or can) that requires a minimum space for suction pit (Flowserve Corporation)

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CAN PUMP INTAKES

A can pump is defined as one that has a can, or barrel, surrounding the pumping unit. This can acts as a "sump" or intake structure for the pump suction impeller. The can may be either closed bottom, and contain the pump suction nozzle, or open bottom and connect directly to a piping header. The can design must provide uniform, stable flow distribution to the suction impeller inlet.

Vertical turbine pumps (Figure 15) require uniform inflow to the suction bell to avoid swirling and submerged vortices that may result in cavitation, vibration, and accelerated pump wear. When a pump with an open bottom can design is connected to a horizontal header (Figure 16a), the velocity in the header should be no more than 6 ft/s (1.8 m/s) to allow the liquid to turn and flow upward to the pump suction bell. The velocity in the can rising to the pump should not exceed 5 ft/s (1.5 m/s), and the suction bell inlet should be located at least three diameters above the top of the horizontal header. For velocities approaching the maximum recommended levels, vortex-suppressing vanes may be added to the suction bell area to break up swirling and nonsymmetrical flow patterns as they approach the impeller inlet.

If a 90° turning vane elbow on the can assembly surrounding the pumping element is used (Figure 16b), the velocity in the horizontal header can be as high as 8 ft/s (2.4 m/s). The turning vane elbow should be sized for a maximum velocity of 5 ft/s (1.5 m/s).

Most can pumps are of the closed bottom design (Figure 17). In this arrangement, the pump suction nozzle is located either in the can or in the pump nozzle head that contains both the suction and discharge nozzles. The pumping element (assembly) must be centered in the can to avoid nonuniform flow to the suction impeller. Flow straightening vanes are suggested for all can intakes. Because of the limited volume of liquid in the can, surging of the liquid level within the can—or barrel—cannot be tolerated; the can should be arranged so it is always full. The velocity of the flow between the pumping element and the inside of the can should not exceed 5 ft/s (1.5 m/s). For a suction nozzle velocity of 4 ft/s (1.2 m/s) maximum, the centerline of the nozzle should be at least two diameters above the suction bell inlet. To ensure uniform flow distribution into the can, the suction nozzle should be connected to at least five pipe diameters of straight pipe before elbows or other flow-disturbing fittings are installed.

Pumps with submersible well-type motors require flow around the motor for cooling. A shroud is typically used to direct flow across the motor as it goes into the pump. The top of such a cooling shroud is covered to restrict downward fluid flow and still allow venting

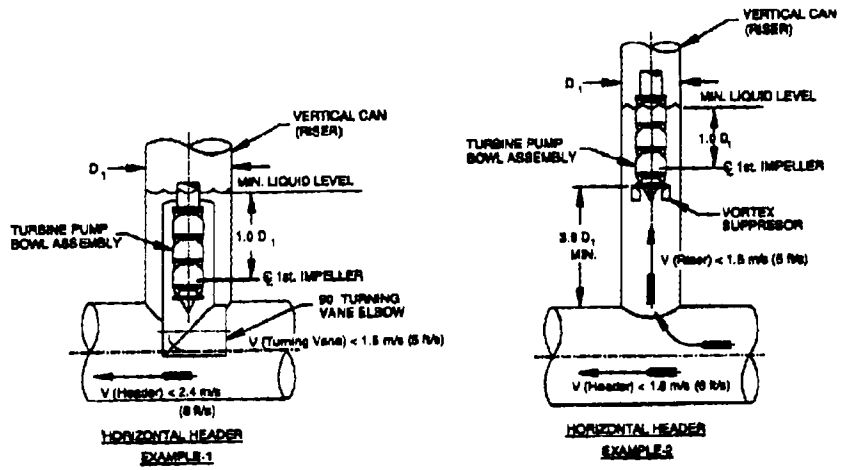


FIGURE 16 Open bottom can intakes for pumps less than 5000 gpm (315 L/s) (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

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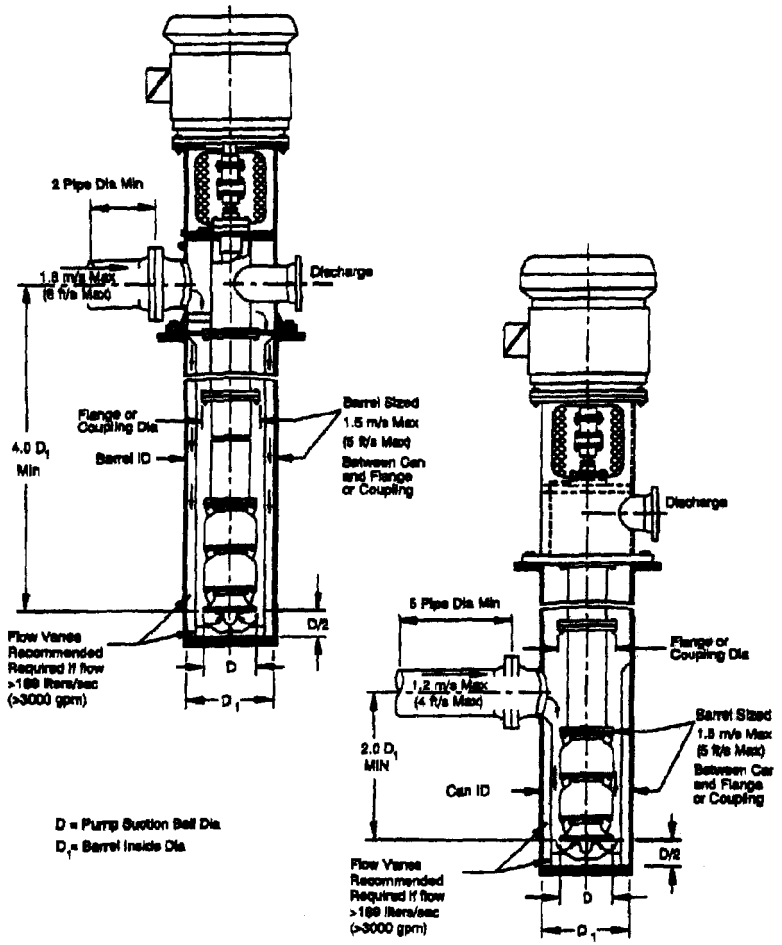


FIGURE 17 Closed bottom can (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

of air from the shroud. This arrangement is described in detail in Reference 1. It is recommended that the inlet piping be sized to limit draw-down of the liquid below the minimum required level during the start-up to a period of less than 3 s.

SUCTION PIPING

Single Pumps Piping to the suction of a dry-pit centrifugal pump (Figure 18) must be carefully planned to provide uniform, straight-line flow to the impeller, and adequate pressure and sealing against leakage, in or out. Air pockets just prior to entrance to the pump should be avoided, as well as down flow lines subject to sudden pressure changes. Air pockets can be prevented by proper elevations (Figure 18a). Pressure surges can be controlled by surge tanks, air tubes, and so on, which may require a system pulsation study to determine possible need and solution.

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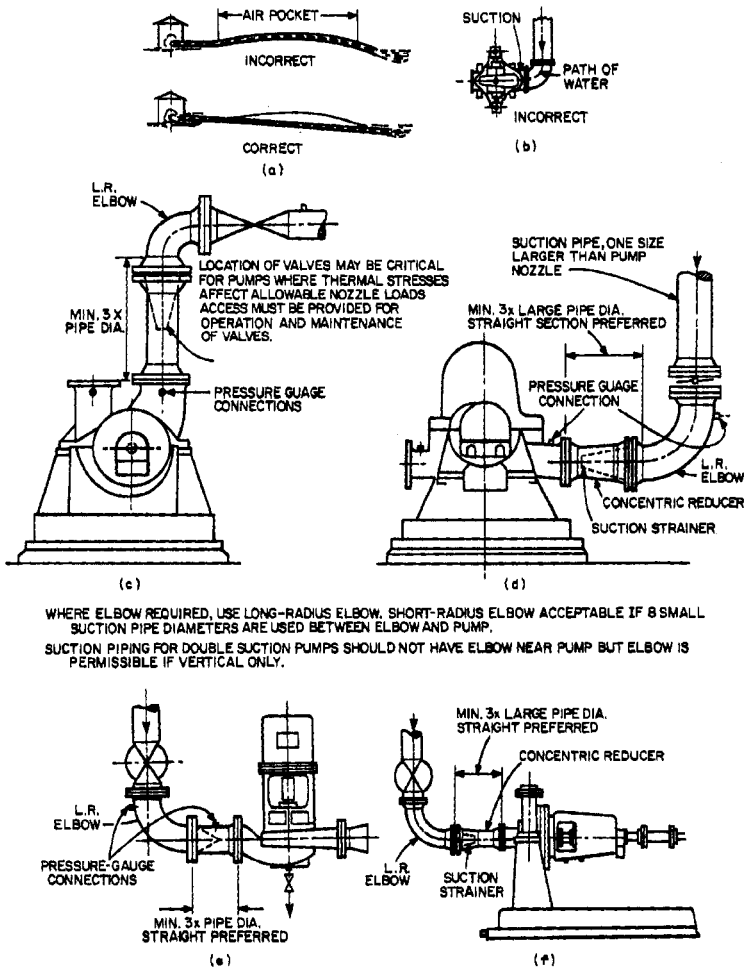


FIGURE 18 Suction piping faults for dry-pit centrifugal pump. (a) Air pockets should be avoided. (b) Suction elbow should not be in a plane parallel to pump shaft. (c) Valve location may be critical to pump nozzle loading. (d), (e), Suction elbow should be one size larger than pump's, long radius and three large pipe diameters distant. (d), (f) Concentric reducer should be installed distant from pump.

To provide an optimum flow pattern to avoid impeller disturbance, it may be necessary to have a straight run of pipe of as much as eight pipe diameters immediately prior to the pump suction (e.g., following a short radius elbow or tee). Following a long radius elbow or a concentric reducer, a straight run of at least three pipe diameters is recommended (Figure 18c to f). Eccentric reducers should not be used next to the pump suction nozzle. Although installing eccentric reducers with the flat side on top will eliminate a potential air pocket, large changes in diameter could result in a disturbed flow pattern to the impeller and cause vibration and rapid wear. Pipe venting, in conjunction with a concentric reducer, may be preferable to the use of an eccentric reducer.

The velocity of the fluid in the suction piping should only be exceeding 8 ft/s (2.4 m/s) when a uniform velocity profile and sufficient pressure at the pump inlet flange can be granted. The velocity may increase in the direction of the pump through the use of reducers.

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Ideally, a suction pipe should approach a double suction pump perpendicular to the shaft centerline. If there is an elbow in the suction piping upstream of the suction flange, it should be bringing flow from either overhead or below, not from the side of the pump. If there is a short radius elbow or other flow-disturbing device in the suction piping upstream of the suction flange, there should be at least five pipe diameters between the device and the suction flange. If a short radius elbow is in the same plane as the impeller shaft, there should be at least eight pipe diameters between the elbow and the suction flange. An incorrect installation could result in an uneven flow to both sides of the double suction impeller. This could cause a reduction in capacity and efficiency, an increase in thrust on the bearing, vibration, noise, and possible cavitation damage to the impeller. A dry-pit pump (Figure 1g) may operate with a suction lift and therefore will be located above the liquid source. All losses in piping and fittings will reduce the available suction pressure. Suction piping should be kept as simple and straightforward as possible. Any pipe flange joint or threaded connection on the suction line should be gasketed or sealed to prevent air in-leakage, which would upset the vacuum and keep the pump from operating properly (already at air rates of 1% and more).

If expansion joints are required at the suction of a pump, an anchor should be interposed between the pump nozzle and the expansion joint to prevent additional forces from being transmitted to the pump case and disturbing rotating clearances. The same requirement applies to a sleeve coupling used to facilitate installation alignment.

Reciprocating pumps must have additional consideration because of the pulsating nature of their flow. Suction piping should be as short as possible and have as few turns as possible. Elbows should be long-radius. Pipe should be large enough to keep the velocity between 1 and 2 ft/s (0.3 and 0.6 m/s). This will generally result in pipe one to two sizes larger than the pump nozzle. High points that may collect vapor are to be avoided or, if necessary, properly vented. A pulsation dampener or suction bottle should be installed next to the pump inlet. Available NPSH should be sufficient to cover not only reciprocating pump requirements and frictional losses but also acceleration head (see "Surge and Vibration").

Manifold Systems All comments relative to single pumps apply to manifold-pump systems, as well as some additional points.

In a suction manifold, the main-line flow should not be more than 8 ft/s (2.4 m/s). Branch outlets should be at 30 to 45° relative to main-line flow rather than 90° but never positioned opposite toward each other. The diameter of the branch outlet piping should be smaller than in the manifold piping (Figure 19). Uniformity of the velocity profile in the branch outlet is further increased through the use of cone- or bell-shaped fittings. Branch outlets can be spaced 2 header diameters apart from each other for larger ratios between branch outlet diameter and header diameter (≥ 0.3), while 3 branch outlet diameters is sufficient for diameter ratios less than 0.3. Also, if the angled manifold outlet is used, pumps can be set as close to the manifold as the elbow, valve, and tapered reducer will allow.

Manifold sections beyond each branch takeoff should be reduced to such a size that the velocity remains constant. One exception to this scheme is tunnel suction (Figure 20). The flow through the tunnel may operate independent of the pumps, which are suspended in boreholes drilled into the roof of the tunnel. Boreholes at least one-third of the tunnel diameter should be horizontally spaced at least 12 borehole diameters apart. Smaller ratios can be closer to a minimum of six diameters. Pump suction bells should be at least two borehole diameters above the tunnel roof. Velocities in the tunnel should be kept below 8 ft/s (2.4 m/s) for best pump performance.

NPSH The net positive suction head so essential to trouble-free pump operation is always reduced by losses in suction piping. An economic balance must be obtained between pump size and speed, required NPSH, pipe size, and suction velocity. If the suction source is a tank, such as a deaerator in a power plant, it is quite expensive to elevate the tank. Therefore, the available NPSH is low. The pipe size from the tank to the pump should be large for low velocity and the length should be short for minimum losses.

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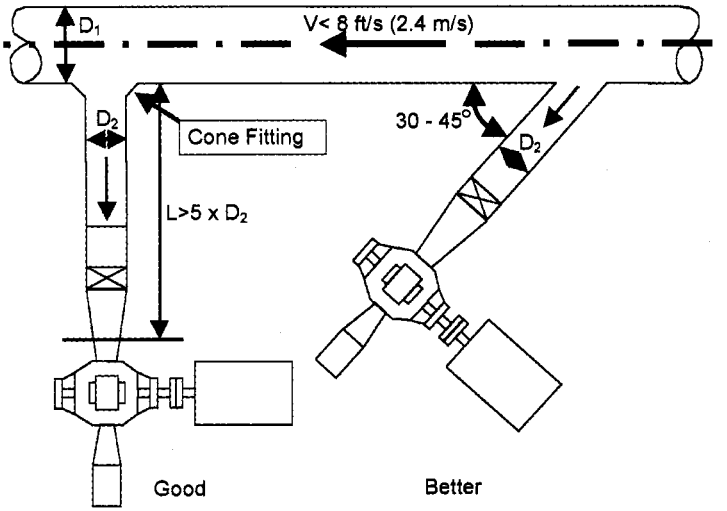


FIGURE 19 Suction pipe header recommendations for dry-pit centrifugal pump

A cooling tower on a hill supplying water to a pump station below could have a much higher pipe velocity and longer length without forcing an extremely low NPSH requirement on the pump.

A dry-pit pump should be as close to the suction source as possible. When the NPSH required indicates a suction lift is possible, the most advantageous solution is to reduce the suction losses by increasing the pipe size, flaring the inlet bell, and keeping the pump suction eye of the first-stage impeller close to the minimum water level.

High-Pressure Inlets Pumps in series build up pressure, so the second and following pumps will have a high-pressure suction piping connection. This emphasizes the need for tight joints and flanges and careful welding. Expansion joints should not be used because the hydraulic forces on the pump would be large, difficult to restrain, and perhaps impossible for the pump to handle without distortion. As NPSH will not be a problem for pumps downstream of the first pump, the pipe size between pumps can be kept small to minimize design problems and valve costs.

Effect on Pump Efficiency Other than mechanical operation, the greatest effect of flow disturbance at the pump suction is on pump efficiency. The higher the pumping head, the less this effect becomes. For very-high-head pumps, a high velocity may be ignored completely unless there is an extremely large power evaluation factor. Greater attention

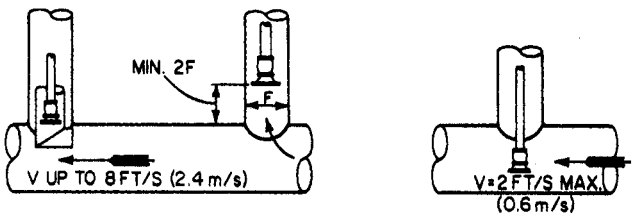


FIGURE 20 Higher tunnel velocities require isolation of pump from direct flow to prevent distortion of close clearance parts and shaft (Hydraulic Institute Standards, 13th Edition, 1975—out of print)

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should be given to the suction pipe design for pumps producing 100 ft (30 m) of head or less than those with high head, as efficiency may be worth many dollars in power costs over the life of the plant and equipment.

Pump efficiency relates to the efficiency of the whole system, and so it may be well worthwhile to invest more money in a large suction pipe.

Surge and Vibration One of the possibilities arising from a power failure at a pump station is the reversal of a centrifugal pump if a valve fails to close, and its subsequent operation as a turbine. Underrated head, a pump will run from 20 to 60% above rated speed in the reverse direction. In its transition to that phase, the forward motion of the water is interrupted and gradually reversed. At the time of the power failure, the flow velocity in the suction pipe may not decelerate slowly enough to prevent a surge in the direction of the pump. The suction piping should be designed to withstand the resulting pressure rise because absolute integrity of valving and power supply will be too costly as a design parameter (see Chap. 11 for additional information on this subject).

Rotating machinery must have a design vibration frequency that is sufficiently removed from a system frequency to avoid sympathetic activity. Any combination of vibration frequencies near enough to each other to react will do so when a prime source, such as a pump, excites them. When a piping system has been designed to allow only very low stresses to be transmitted to a pump nozzle, it is quite vulnerable to vibration. It is usually good practice to analyze a suction system made up of pipe, valves, hangers, restraints, pump nozzle loads, pump speed and impeller configuration, foundation, and anchors to be sure the system is not "in tune." At the design stage, it is relatively easy to change a valve or elbow location or to add a surge suppresser. It is usually much more costly to change a pump.

Reciprocating pumps are surge-producing machines. In particular, they require sufficient energy at suction to overcome pump required NPSH and pipe friction and a form of energy called *acceleration head*. The pump energy must overcome the acceleration-deceleration pulsation flow in the suction end, which could lead to liquid flashing with pump noise and vibration. Surges large enough to rupture the pump cylinders may also be produced.

The total NPSH required by a reciprocating pump must include the NPSH required by the pump plus frictional loss from the suction pipe plus acceleration head. Acceleration head in feet (meters) is

$$Ha = \frac{LVNC}{gK}$$

where L = actual (not equivalent) length of suction pipe, ft (m)

V = velocity of flow in the suction line, ft/s (m/s)

N = pump speed, rpm

C = pump constant, decreasing with number of cylinders from 0.2 to 0.04

g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

K = liquid constant: 2.5 for compressible fluids, 1.5 for water

The acceleration head can be reduced by installing a pulsation dampener in the suction line near the pump.

TRASHRACKS, SCREENS, AND STRAINERS

Except for special designs for handling slurries and other solids, most pumps are limited in their ability to handle dirt, trash, and solids of any size down to microparticles. Pump passageways (volutes, diffusers, impellers) handle generally spherical objects up to a "sticking size," but a severe problem may be caused by dust-size particles that lodge in sleeve bearings, bushings, and wear rings and cause either rapid wear or rotation seizure.

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Wet-pit sources with water intake structures are very likely to be polluted by different kinds of debris. Depending on the probable source of debris, such as a river subject to flooding with considerable flotsam in the runoff and with a very loose bottom, or a lake at constant level without disturbing inlet flows near the structure and with a solid bottom, the protection needed may include only a bar trash rack plus rotating, flushed, fine-mesh screens. If sediment deposit is likely, a settling basin may be required.

The designer must note that, if this equipment is useful, it will pick up debris and gradually increase the velocity through the openings as the net area decreases with blockage. When this occurs at the trash rack, the water level differential will build up, causing a waterfall with increased velocity and turbulence on the pump side of the rack. In addition, the increase in velocity may pull more debris through the bars than can be tolerated. It is best to rake these racks (Figure 21) frequently enough to keep the differential head across the rack below 6 in (0.15 m). The spacing of the bars should be such that objects that cannot be pumped would be excluded from passing through. This, in

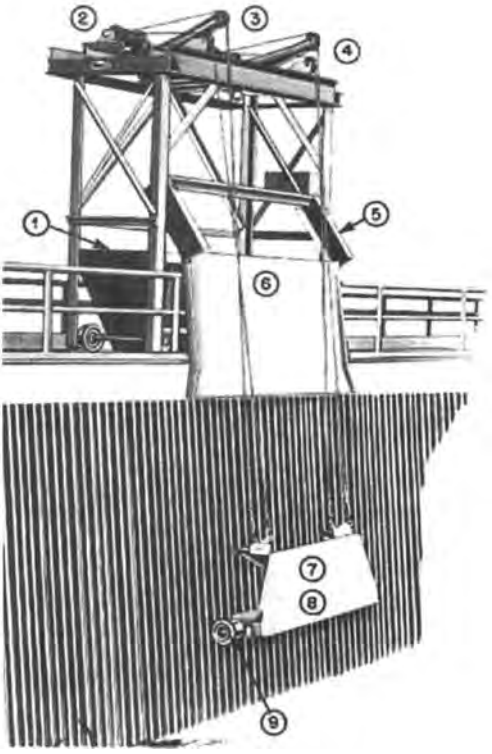


FIGURE 21 Trash rack with raking mechanism: (1) Large enclosed trash hopper contains debris discharged by rake. Hinged door in end of hopper opens wide for debris removal. (2) Heavy-duty, single-drum hoist, push-button controlled with two separate cables—one for carriage, one for rake teeth. (3) Walking beam actuated by hydraulic cylinder controls position of rake teeth. (4) Fixed sheave. Cable operating over stationary sheave raises and lowers rake carriage. (5) Discharge guide flanges assure positive positioning of rake over trash chute prior to dumping. (6) Dead plate, or apron, integral with superstructure guides rake to discharge point—prevents trash from falling off rake prematurely. Design permits operation over 3.5-ft. (1.1-m) high hand rail. (7) Self-centering rack-guided carriages allows rake to ride over obstructions in water during lowering cycle. Debris of all types picked up on lifting cycle rather than forced to bottom of channel. (8) Rake mechanism assures positive removal of debris with maximum carrying capacity. Hydraulic relief valve provides automatic overload relief. Teeth automatically open if overload occurs, permitting load to drop off rake. No cable failures due to overload. (9) Wide, flanged rollers ride on at least two rack bars (Envirex, Inc., a Rexnord Company)

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general, will call for the bar spacing to be in proportion to the size of the pump. A pump manufacturer can determine the maximum size sphere a pump will handle, and the bar spacing should be limited to 50% of that value. The size of the bar, the lateral distance between supports, and the pier spacing will influence the rate of debris accumulation and the allowable design differential head.

Rotating screens (Figure 22) will remove trash of a much smaller size because the accumulation is continuously removed and the open area is kept uniform. Finer screening than that required by the pump may be necessary in installations where the liquid pumped must pass through small openings in equipment serviced, such as condenser tubes or spray nozzles. Screens are usually installed in conjunction with trash racks so large, heavy pieces will not have to be handled by the screens. Because velocity through the screen is limited the pit cross section may be determined by screen requirements. The velocity through the screen can have maximum values up to 2 ft/s (0.6 m/s) if the amount of trash is limited. High amount of trash and/or environmental considerations require lower velocities. If flow is such that a maximum-width screen available would be too long (deep) for practical or economic reasons, two screens may be employed with a center pier. In this case, the distance to the pump should be increased 50% over single-screen distance. Piers should be rounded (radiused) on the upstream side and ogived (tapered to a small radius) on the downstream side. Any corners at the sidewalls should be faired at small angles to the opening and wall to prevent pockets where eddies can form.

The trash collected by the screens must be disposed of. Traveling screens carry trash up into a hood above the operating floor, at which point a series of spray nozzles wash the trash into a trough leading into a disposal area (Figure 23). The nozzles are supplied by pumps sized for 200 to 300 gpm (45 to 68 m^3/h) with pressures of 60 to 100 lb/in^2 (413 to 690 kPa). These pumps are normally deep-well turbine multistage units. They are located

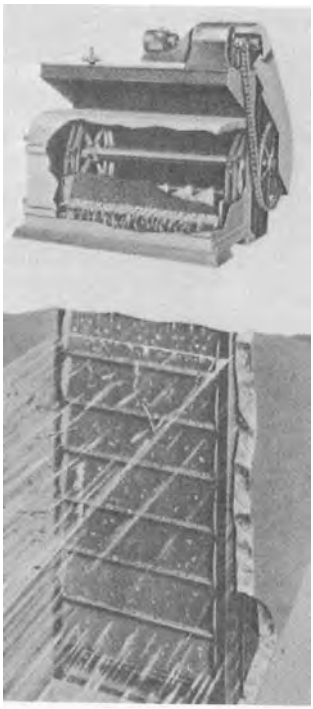


FIGURE 22 Traveling water screen (Envirex Inc., a Rexnord Company)

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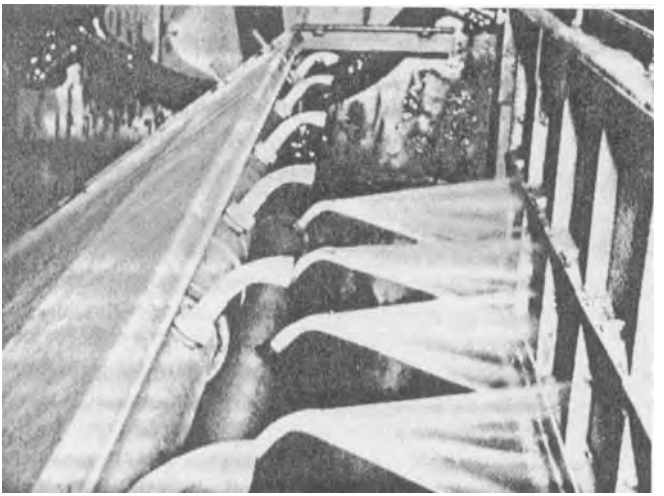


FIGURE 23 Spray nozzle cleaning of baskets on traveling water screens (Envirex Inc., a Rexnord Company)

in the clear well, if possible, close to a wall. If this is not possible, they can be suspended in the circulating water pit, to one side and ahead of the main pumps. Care must be taken that they do not disturb the flow to the larger pumps. Their submergence requirements are usually less than those of the main pumps, and this allows use of the pump setting that gives the least interference with either pump flow.

It is possible to have these pumps also dewater the pit. This will require additional piping and valves and a more careful location of the pump because it will, of necessity, be close to the bottom of the pit. A small chamber off the main pit, located far enough from the main pumps to avoid eddying, will be required.

The direction of the flow from the forebay through the screens and into the pump area should be continuous. Avoid right-angle screens, through which flow must change direction at least once and possibly twice. If screens must be at an angle to the flow into the pumps, increase the screen-to-pump distance by 100%. Environmental considerations may increase the possibility of problems in this area.

A wet-pit source may never be quite free from some degree of silt or sand. After it is accepted by the pump, this liquid goes into a piping system, usually one with other pumps involved. The original pit pump may also be a dry-pit type. In either case, if the pumps of a system are to operate for long periods of time with minimum maintenance, wearing parts must be protected by fine screens or strainers.

In any screening operation, it is essential to remember that if the screen is necessary and does a good job, it will plug up and thus requires cleaning. In a wet pit, therefore, neither the suction bell of a wet-pit pump nor the suction bell going to a dry-pit pump should be basket-screened. Collection of debris will soon increase inlet velocity to unmanageable proportions, and the pump operation will suffer.

It follows that if the basic screening discussed previously does not provide liquid clear enough to be pumped, additional strainers must be put into the system. The pump manufacturer usually specifies the need for such strainers, and the degree of particle reduction. In some instances, the system may require water as pure as it is possible to produce, as in boiler-feed service. For any situation, a variety of strainers are available.

Drum rotating strainers (Figure 24) can handle solids as small as approximately 0.01 in (250 μm). Backflushing is used to renew the strainer area. The addition of woven wire cloth to such strainers may remove materials down to 0.001 in (25 μm). Woven wore drum strainers used in conjunction with sand filters will reduce the amount of backflushing required for the sand filters.

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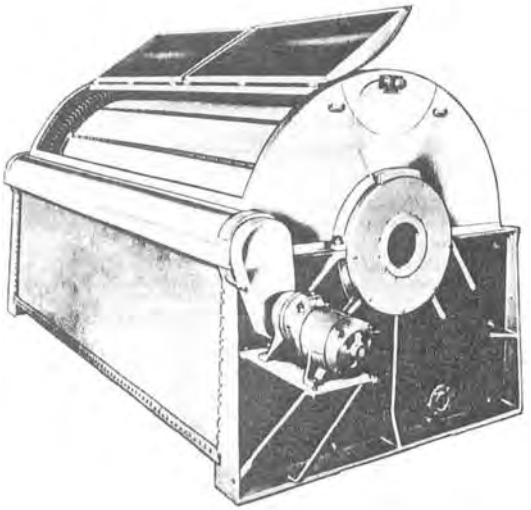


FIGURE 24 Drum filter (Green Bay Foundry and Machine Works)

Where a finer degree of filtration is required, a line filter is used. Single flow filters require system shutdown to remove and clean baskets, but either a motorized automatic type of single flow filter (Figure 25) or twin filters with alternate flow by valve arrangement (Figure 26) permit uninterrupted system operation. When a particularly difficult situation exists, a battery of filters in series may be used.

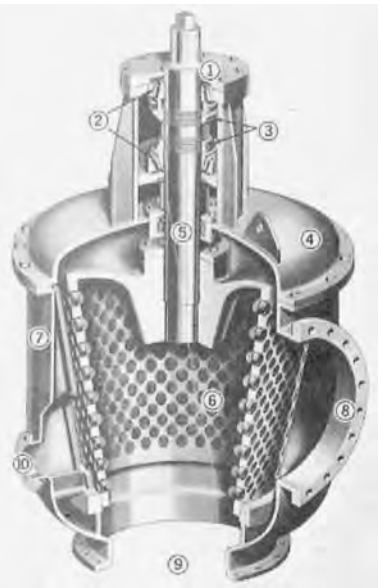


FIGURE 25 Single flow strainer with provision for backwashing without interrupting service: (1) strainer drive, (2) strainer drum support bearings, (3) lock nuts for drum adjustment, (4) cover, (5) shaft, (6) drum—tapered for adjustment, (7) body, (8) inlet, (9) outlet, (10) backwash opening (S. P. Kinney Engineers)

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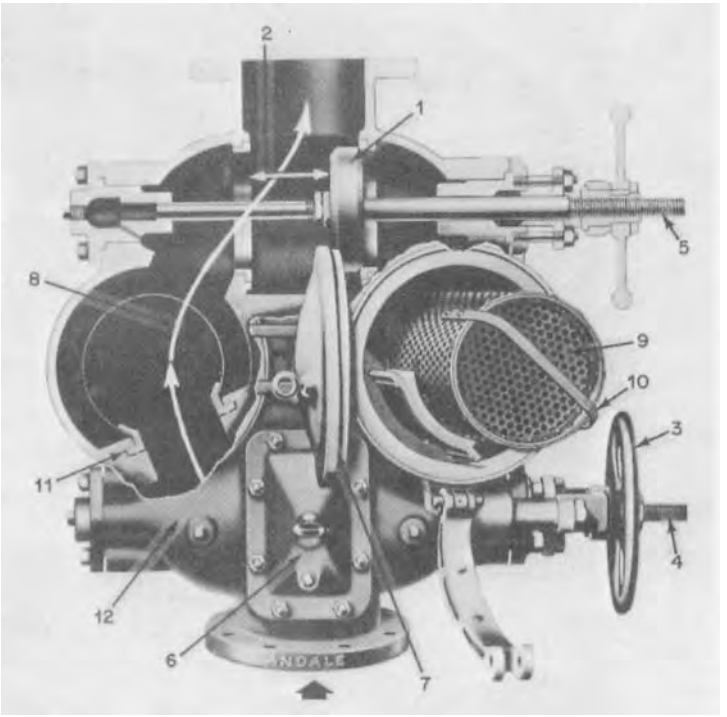
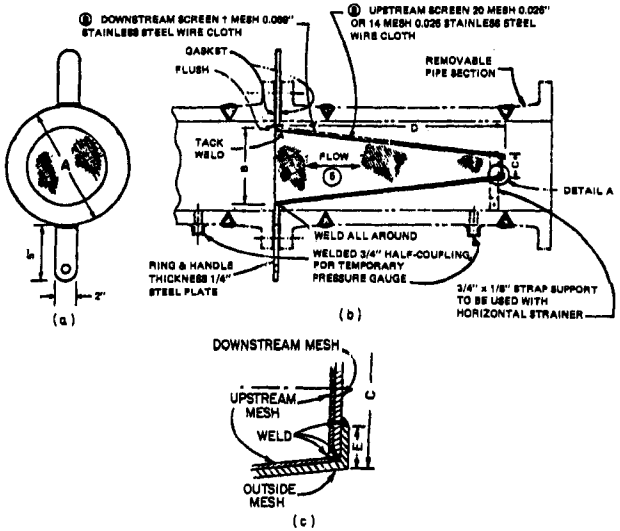


FIGURE 26 Duplex basket strainer. Flow through one side allows cleaning of opposite basket while line is still in service. (1) positive flow diversion, (2) rapid switchover, (3) free-turning handwheels, (4) visual indication of valve position, (5) external acme threads, (6) internal access, (7) cover and clamp(s), (8) streamlined flow, (9) basket, (10) basket handle, (11) basket, (12) one-piece body construction (Andale)

For areas where wear can be extremely critical, such as bearings and stuffing boxes (mechanical seals), additional filtering is necessary, especially if the entrained silt is abrasive. A vortex filter using centrifugal flow action will remove particles down to a few micrometers. These filters have continuous sediment removal and should not require any maintenance (Section 2.2.1, Figure 78).

Some systems, such as boiler feedwater, are "closed" after they are in an operating cycle. Chemical and mechanical cleaning components are designed into the system to obtain a high degree of water purity. Before the system is ready to operate, however, the piping, tanks, valves, and so on will have a residue of weld spatter, metal chips, and other debris from the fabrication process. Temporary strainers are used in the suction lines of boiler feed pumps and condensate pumps to gather up this debris. These strainers are in-line fabricated metal mesh of either a cone or box type. Feed pumps are most often equipped with cone strainers, pointing preferably upstream (Figure 27). The length and diameter are set to conform to the suction pipe size and the mesh so the velocity does not become excessive. These strainers, set between flanges in a pipe section, must be removed and cleaned whenever the differential pressure reaches 2 to 3 lb/in^2 (15 to 20 kPa). Condensate pump strainers are usually box strainers, a square section directly in front of the suction flange with a square box larger than the pipe diameter laced with egg crate metal mesh (Figure 28). This basket strainer can be lifted out of the box for cleaning as necessary. These strainers for collecting fabrication debris may be removed when the need disappears and the system will remain clean.

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Flange or pipe size	A	B	C	D: 20 mesh 0.025-in wire, 0.025-in opening	D: 14 mesh 0.025-in wire, 0.046-in opening	E
3	5	2-¾	1-½	16	11	¼
4	6-⅝	3-¼	1-½	16	11	¼
6	8-½	5	2	18	12	¼
8	10-¾	7-¼	2	20	13	¼
10	12-¾	9-¼	3	23	14	¼
12	15	11	3-½	24	15	¼

FIGURE 27 Cone strainer for start-up service in boiler-feed pump system. (a) cross-section, (b) lengthwise section, (c) typical detail A (nose pointing downstream). All dimensions are in inches (1 in = 0.0254 m = 2.54 cm). Strainer ring material is carbon steel. When a strainer has served its purpose, the screen portion is removed, the ID of ring is cut to suit ID of pipe, and the ring is used as a spacer. Cone strainers may be installed with nose pointing either upstream or downstream. The fine mesh screen should always be on the upstream surface of the strainer.

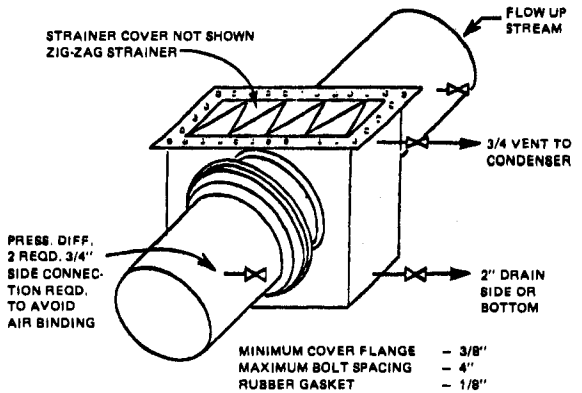


FIGURE 28 Box strainer for temporary start-up service. Pressure differential connections are on top or side only. Upstream mesh is #16 stainless steel with 0.018-in (0.46-mm) wire diameter. Downstream mesh is #2 stainless steel with 0.063-in (1.6-mm) wire diameter (1 in = 25.4 mm).

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SECTION 13.2
INTAKE MODELING

MAHADEVAN PADMANABHAN
JOHANNES LARSEN

PROBLEMS ENCOUNTERED IN PUMP INTAKES

The various hydraulic problems associated with a pump pit include formation of surface and subsurface vortices, prerotation and swirl, and flow separation at or near the suction bell of either wet-pit or dry-pit centrifugal pumps (Figures 1a and 2). Any of these problems can adversely affect pump performance by causing cavitation, vibrations, or loss of efficiency.[1] Usually there is more than a single reason for these problems, and the extent of the combined effects is difficult to predict reliably by mathematical modeling or computational fluid dynamics (CFD). Formation of vortices, for example, even though dependent on suction pipe velocity and submergence, is strongly influenced by added circulation from vorticity sources, such as a nonuniform approach flow resulting from intake and approach channel geometries; rotational wakes shed from obstructions, such as columns or piers; and the velocity gradients resulting from boundary layers at the walls and floor.[2] The circulation contributed by these vorticity sources is unpredictable and strongly dependent on intake design and operating conditions, especially for large pumping units with multiple bays fed by a common approach channel. In these cases, physical modeling is the best way of predicting the behavior of the prototype with a reasonable degree of reliability.

Free Surface Vortices This type of vortex is considered objectionable when it draws air bubbles or an air core into the pump inlet (Figure 1b). Under nominal approach flow conditions, strong air core vortices causing air ingestion with air concentration as high as 10% have been reported.[29] It has been established that an air concentration in the suction pipe of from 3 to 5% can lower pump efficiency.[3] Also, air in the form of large bubbles or slugs can cause the impeller to vibrate. Strong surface vortices that do not draw air are also objectionable if they draw debris into the suction pipes and because their high rotational velocity can reduce the local pressure sufficiently to induce cavitation.

Subsurface Vortices This type of vortex, also known as a submerged vortex (Figures 1c and 2), usually originates from a floor or wall and is induced by vorticity produced in separation zones close to the pump entrance or, in wet-pit pumps, below the bell. The reduced pressure in the vortex core causes fluctuating load on the pump impeller along with associated vibration and noise,[4,5] increased possibility of cavitation, higher inlet losses, and decreased pump efficiency, especially when the core pressure is sufficiently low to release dissolved air or other gases from the fluid.

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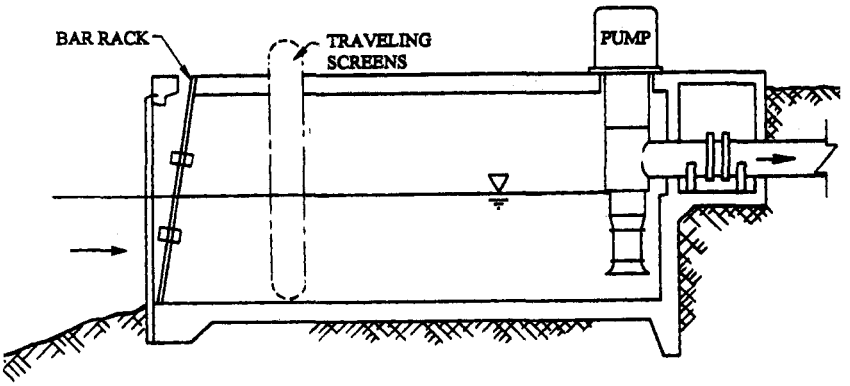


FIGURE 1A Typical vertical wet-pit pump intake

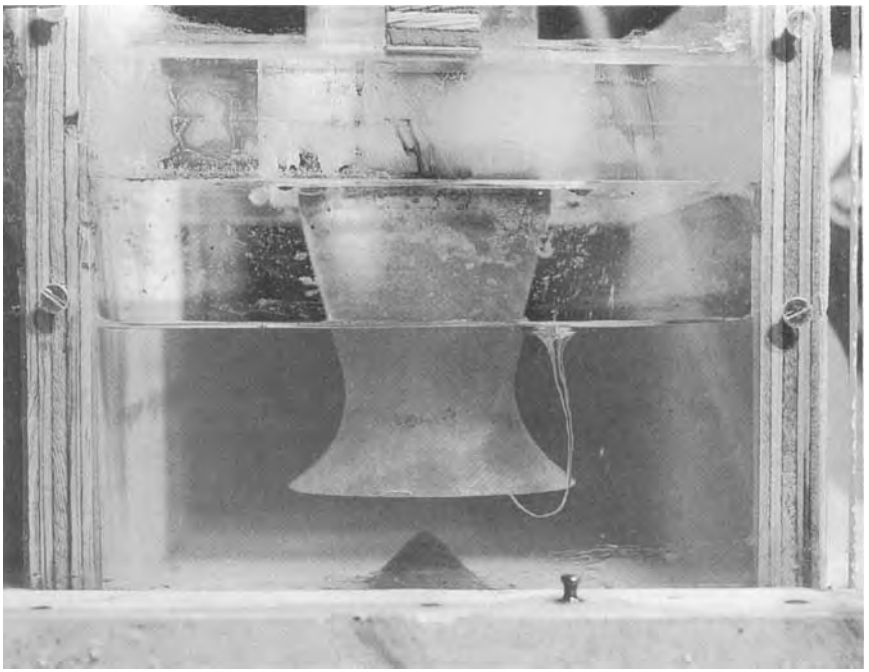


FIGURE 1B Free surface vortex can form at pump suction bell entrance due to combination of low submergence, high suction velocity, nonuniform approach flow, or other vorticity sources (Courtesy of Alden Research Laboratory, Inc.)

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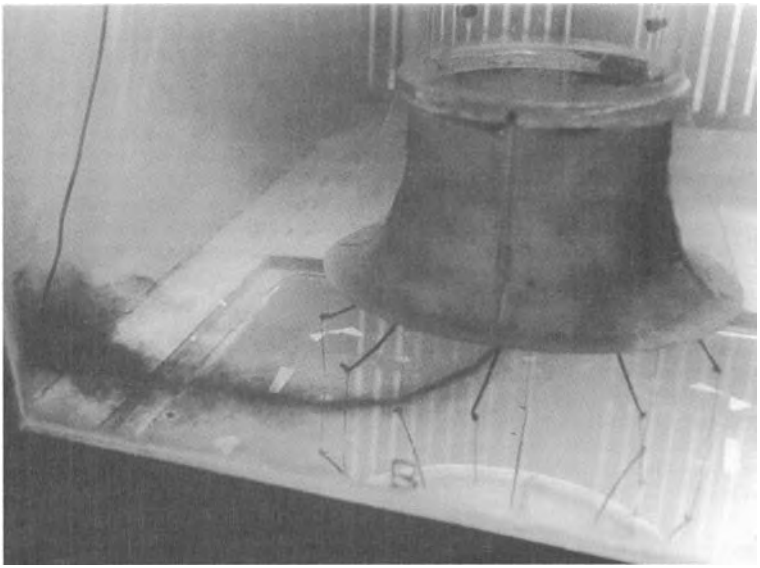


FIGURE 1C Subsurface vortex formed at the side wall due to poor flow guidance to the bell; vortex identified in the model with dye (Courtesy of Alden Resrearch Laboratory, Inc.)

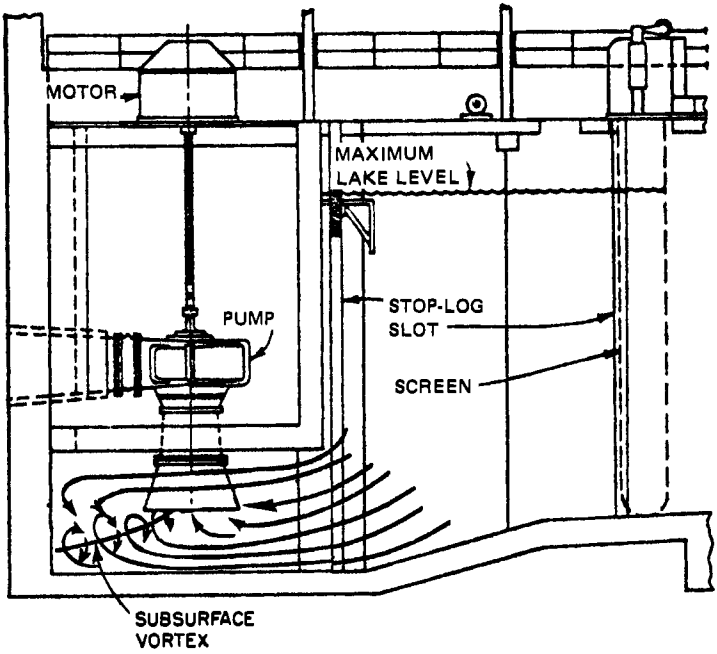


FIGURE 2 Typical vertical dry-pit pump intake. Improper spacing of suction bell relative to floor and back wall caused subsurface vortexing. Model test revealed problem and correction (Reference 4).

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Prerotation and Swirl *Swirl* is a general term for any flow condition (due to vortexing or a pipe bend) where there is a tangential velocity component in addition to a usually predominating axial flow component. *Prerotation* is a specific term to denote a cross-sectional average swirl in the suction line of a pump or, in case of a vertical wet-pit pump, upstream of the impeller.

The prerotation angle θ is a measure of the strength of the tangential velocity component u_t relative to that of the axial velocity component u in the flow approaching the pump impeller; that is, $\theta = \tan^{-1}(u_t/u)$. Adverse effects on the pump are decreased capacity and head when the rotation is in the direction of pump rotation and increased capacity and head when the rotation is opposite the pump rotation (antirotation). The increased capacity is associated with an increase in power requirement and may cause motor overheating.

Prerotation will influence pump performance because the flow approaching the impeller already has a rotational flow field that may oppose or add to the impeller rotation, depending on direction. The design of the pump blades (i.e., shape and angle) usually assumes no prerotation, and the existence of prerotation implies flow separation along one side of the impeller blades. The degree of prerotation that should be of concern depends on the type of pump and may not always be known. Prerotation could be quantified in a model by an average cross-sectional swirl angle, determined by detailed velocity measurements, or by readings on a swirl meter. Because swirl decays along a pipe as a result of wall friction, internal fluid shear, and turbulence, the swirl meter in a model suction pipe should be located near the impeller.

Losses Leading to Insufficient NPSH A poorly designed pump intake could result in large inlet losses. Losses caused by screens, poor entrance conditions, vortexing and swirl, and vortex suppression devices may add up to a value so great that the required NPSH of the pump is not satisfied. Increased inlet losses due to swirl have been reported in laboratory studies.[6] In a nuclear reactor residual-heat-removal sump model, inlet losses in a preliminary design wherein air core vortices and a high degree of swirl were present were 20% higher than in a revised design with no strong vortices and swirl, and with similar pipe entrance geometry and flows. Because the degree of vorticity and swirl cannot be predicted and it is therefore not possible to calculate inlet losses reliably; they are usually obtained from model studies. With the experimentally derived values of the inlet losses, the NPSH available should be checked by recalculation.

DESIGN GUIDES

As a preliminary design guide (Section 13.1), published information may be used to establish pump sump dimensions and a minimum desired submergence.[7-10] Figures 1 and 2 of Section 13.1 show the basic layout of a pump sump and present typical dimensions in relation to the flow required per pump, as published by the Hydraulic Institute.[7] Some of these references may express sump dimensions in terms of bell mouth diameter, and Figure 3 shows the typical relationship between design flow for a given wet-pit pump and the bell mouth diameter required to achieve reasonable and desirable velocities approaching the impeller. One should be aware that the sump dimensions and minimum derived submergences given by design guides are considered applicable for the ideal condition of a simple, straight approach flow with a constant, low approach velocity to the pump sump.

The need for a physical model study still exists when site or operating conditions make the ideal condition impossible. For example, for a particular pump intake configuration, a hydraulic model study indicated that a strong submerged floor vortex existed with floor clearance of 0.5 times the bell diameter (a usual design value) and that a reduced floor clearance of 0.3 times the bell diameter eliminated the vortex. Typically, for pump intakes of the following types, model studies should be considered essential:

1. Intakes with nonsymmetric approach flow; for example, an offset in the approach channel

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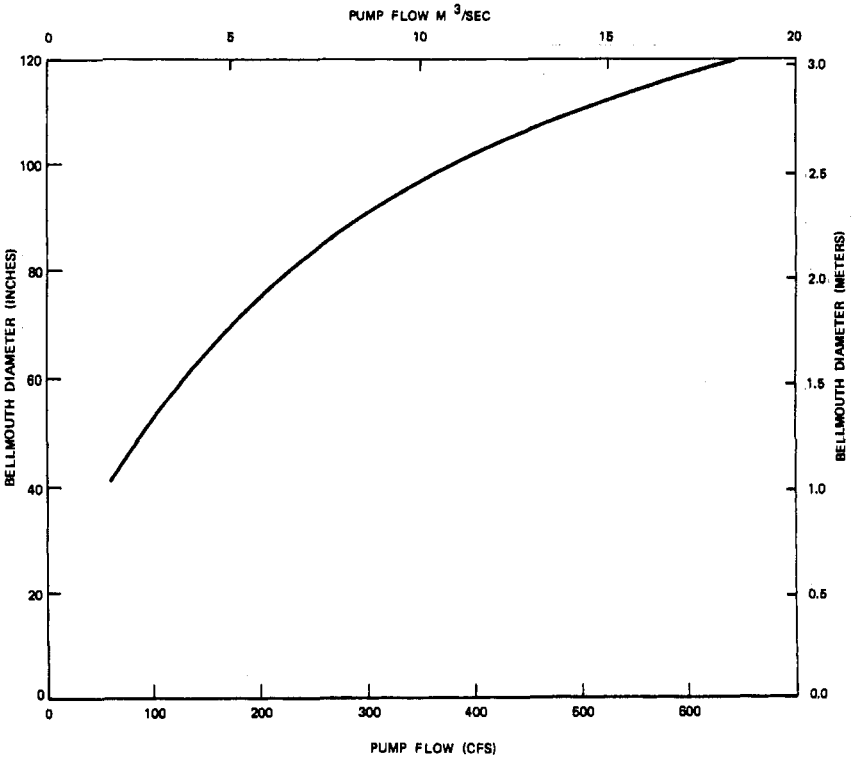


FIGURE 3 An approximate relationship between design flow and required suction bell diameters for vertical propeller pumps for preliminary evaluations

2. Intakes with multiple pump bays with a common approach channel and a variety of pump operating combinations
3. Intakes with pumps of capacities greater than 40,000 gpm (2.5 m^3/s) per pump
4. Intakes with expanding approach channel
5. Intakes with possibilities of screen blockages or obstructions close to suction pipe entrance; for example, reactor containment recirculation sumps, gate guides for dry-pit pumps
6. Dual flow screens intake

For items 1, 2, 4, 5, and 6, a model study is recommended because of the unknown effects a nonuniform approach flow can have on vortexing and swirl. For item 3, considering the cost of large pump installations and the cost of backfit, should problems occur, a model study is recommended to ensure proper pump operation.

MODEL SIMILITUDE

The principle of dynamically similar fluid motion forms the basis for the design and operation of hydraulic models and the interpretation of experimental data. The basic concept

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of dynamic similarity is that two systems with geometrically similar boundaries have similar flow patterns at corresponding instants of time.[11,12] To achieve this, all individual forces acting on corresponding fluid elements must have the same ratios in the two geometrically similar systems. The condition required for complete similitude may be developed from Newton's second law of motion:

$$F_i = F_p + F_g + F_v + F_t \tag{1}$$

where F_i = inertial force, defined as mass m times acceleration a
F_p = pressure force connected with or resulting from the motion
F_g = gravitational force
F_v = viscous force
F_t = surface tension force

Additional forces, such as fluid compression, magnetic, or Coriolis forces, may be relevant under special circumstances, but generally these forces have little influence and are, therefore, not considered in the following development.

Equation 1 can be made dimensionless by dividing all the terms by F_i. Two systems of different size that are geometrically similar are dynamically similar if both satisfy the same dimensionless form of Eq. 1. We may write each of the forces of Eq. 1 as

F_p = net pressure difference × area = $\alpha_1 \Delta p L^2$
F_g = specific weight × volume = $\alpha_2 \gamma L^3$
F_v = shear stress × area = $(\alpha_3 \mu \Delta u/\Delta y)$ (area) = $\alpha_3 \mu u L$
F_t = surface tension × length = $\alpha_4 \sigma L$
F_i = density × volume × acceleration = $\alpha_5 \rho L^3 u^2/L = \alpha_5 \rho u^2 L^2$

where

α_1, α_2, and so on = proportionality factors
Δp = net pressure difference
L = representative linear dimension
γ = specific weight = ρg
μ = dynamic viscosity
$\Delta u/Dy$ = depth-wise velocity gradient
u = representative velocity
σ = surface tension
ρ = density
g = acceleration due to gravity

Noting that the kinematic viscosity, ν, is given by μ/ρ, substituting the above terms in Eq. 1 and making it dimensionless by dividing by the inertial force, we obtain

$$\frac{\alpha_1}{\alpha_5} E^{-2} + \frac{\alpha_2}{\alpha_5} F^{-2} + \frac{\alpha_3}{\alpha_5} R^{-1} + \frac{\alpha_4}{\alpha_5} W^{-1} = 1 \tag{2}$$

where

$$E = \frac{u}{\sqrt{\Delta p/\rho}} = \text{Euler number} \propto \frac{\text{inertial force}}{\text{pressure force}}$$

$$F = \frac{u}{\sqrt{gL}} = \text{Froude number} \propto \frac{\text{inertial force}}{\text{gravitational force}}$$

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$$R = \frac{uL}{\sqrt{\mu/\rho}} = \text{Reynolds number} \propto \frac{\text{inertial force}}{\text{viscous force}}$$

$$W = \frac{u^2 L}{\sigma/\rho} = \text{Weber number} \propto \frac{\text{inertial force}}{\text{surface tension force}}$$

Because the proportionality factors α_1, α_2, and so on, are the same in model and prototype, complete dynamic similarity is achieved if the values of each dimensionless group, E, F, R, and W are equal in model and prototype. In practice, this is difficult to achieve. For example, to have $F_{\text{model}} = F_{\text{prototype}}$ and $R_{\text{model}} = R_{\text{prototype}}$ requires either a 1:1 'model' or a fluid of very low kinematic viscosity in the reduced-scale model. Hence, the accepted approach is to select the predominant force and then design the model according to the appropriate dimensionless group. The influences of the other forces become secondary and are called scale effects.[11,12]

Froude Scaling Pump intake models are generally designed and operated using Froude similarity because the flow is controlled by gravitational and inertial forces. The Froude number is, therefore, made equal in model and prototype:

$$F_r = F_m/F_p = 1 \tag{3}$$

where m, p, and r denote model, prototype, and ratio between model and prototype.

In modeling a pump intake sump to study the formation of vortices, it is important to select a reasonably large geometric scale to achieve large Reynolds numbers. At a large Reynolds number, energy loss coefficients usually behave asymptotically with Reynolds number. Hence, with $F_r = 1$ and a sufficiently high Reynolds number, the Euler number E will be equal in model and prototype. This implies that flow patterns and loss coefficients may be considered similar in model and prototype. From Eq. 3, the velocity, flow, and time scales are

$$u_r = L_r^{0.5} \tag{4}$$

$$Q_r = L_r^2 u_r = L_r^{2.5} \tag{5}$$

$$t_r = L_r^{0.5} \tag{6}$$

For example, if a model-to-prototype scale of 1:10 is used, a prototype velocity of 1.0 ft/s (0.3 m/s) becomes a model velocity of $(1 \div 10)^{1/2} = 0.32$ ft/s (0.09 m/s). In the model, all physical dimensions of the pump bays and approach channel are scaled to the ratio 1:10. Submergence, being a linear dimension, is also scaled to 1:10. A flow of 100 gpm (0.006 m³/s) in the model corresponds to a flow of $100 \times 10^{2.5} = 31{,}623$ gpm (2 m³/s) in the prototype. Similitude parameters and laws are treated in detail in References 11 and 12.

Similarity of Vortices The fluid motions involving vortex formation in pump sumps have been studied by several investigators. It can be shown by principles of dimensional analysis that the dynamic similarity of fluid motion that could cause vortices at an intake is governed by the following dimensionless parameters:

$$\frac{ud}{\Gamma}, \frac{u}{\sqrt{gd}}, \frac{d}{s}, \frac{ud}{\nu}, \text{ and } \frac{u^2 d}{\sigma/\rho}$$

where u = average axial velocity at the bell entrance
Γ = circulation contributing to vortexing
d = diameter of the bell entrance
s = submergence at the bell entrance
ν = kinematic viscosity of water
g = acceleration due to gravity

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σ = surface tension of water air interface

ρ = water density

The influence of viscous effects is defined by the parameter ud/ν = R, the Reynolds number. Surface tension effects are indicated by $u^2d\rho/\sigma$ = W, the Weber number. As strong air-core type vortices, if present in the model, would have to be eliminated by a modified sump design, the main concern for interpretation of model performance involves the similarity of weaker vortices. If the influence of viscous forces and surface tension on vortexing is negligible, dynamic similarity is obtained by equating the parameters ud/Γ, $u/\sqrt{gd}$, and d/s in model and prototype. A Froude model satisfies this condition, provided the approach flow pattern in the vicinity of the sump, which governs the circulation, Γ, is properly simulated.

Considerable research on scaling free surface and submerged vortices has been conducted in the past few years. From a study of horizontal outlets for a depressed sump, it was determined that for pipe Reynolds numbers above 7×10^4, no scale effect on vortex strength, frequency, or air withdrawal existed.[30] Another study indicated that an inlet Reynolds number of 3×10^4 is sufficient to obtain a good model to prototype correlation of vortices.[6] Surface tension effects on vortexing have been shown to be negligible for Weber numbers greater than 120 based on laboratory experiments.[14]

A review of all available data on model versus prototype vortex intensity indicated negligible scale effects for weak vortices and small scale effects for air drawing vortices, and that this effect could be overcome by a relatively small increase in model flow rate.[27] The model flow rate should only be increased by an amount such that sufficient Reynolds and Weber numbers result. Excessively increasing the model flow, particularly to prototype velocity, produces highly exaggerated vortices incompatible with prototype observations.

DESIGN AND OPERATION OF MODELS

Model Scale Scale effects are less as model size increases but construction and operation costs increase with model size, so a compromise must be made. In general, the formation of vortices, both free surface and submerged, is highly responsive to approach flow patterns, and it is important to select a geometric scale that achieves Reynolds numbers large enough to keep the flow turbulent and to meet the fluid mechanic criteria for minimizing scale effects.[13] Also, one should consider other factors such as access for instruments, accurate flow measurements, and ease of modification in selecting a proper scale.

Information on preferred minimum values of Reynolds number and Weber number, discussed earlier, may be used in designing a model and deciding geometric scale. However, adhering to these limits does not, in itself, guarantee negligible scale effects in a Froude model because these limits are based on tests run under ideal laboratory conditions. In real situations, there is usually more than one source of vorticity generation of unknown extent, and a generalization of scale effects for all cases would be inappropriate. To compensate for such unknown scale effects, a common practice is to test a model at higher-than-Froude scaled flows.

A special test procedure involving high temperatures may be used to determine any scale effects and to project the model results to prototype ranges of Reynolds numbers.[2] The water temperature in the model is varied over a range, say 50 to 120°F (10 to 49°C), and flow velocities in the pipes are varied over a range of values, if possible, up to the prototype velocities. Vortexing and other flow patterns over a range of Reynolds numbers are obtained from these tests and can be used to evaluate any possible scale effects. A prediction of the prototype performance can be made based on these tests.

Extent of Model It is very important to include a sufficient length of approach channel in the model because approach flow nonuniformities contribute greatly to vortex formation and swirl. The decision on what approach length should be included is usually based

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on the experience and engineering judgment of the model designer. The requirement is essentially the proper simulation of approach velocity profile to the intake and then on to the pump bays. In some cases, considering the cost and time involved in including a sufficient channel length in a pump intake model, a separate, smaller model can be built to determine the approach flow patterns to the intake and these flow patterns then simulated in the pump intake model.[18]

Figure 4a shows a 1:10 geometric scale model of a four-bay pump intake, with each pump designed to draw 140,000 gpm (312 cfs or 8.83 m^3/s) flow. Figure 4b shows a 1:18 geometric scale model pump intake for a flood control project involving several low head high flow pumps—each handling 360,000 to 450,000 gpm (800 to 1,000 cfs) at 12 ft head (22.68 to 28.35 m^3/s at 3.65 m head).

Modeling of Screens and Gratings In addition to providing protection from debris, screens suppress nonuniformities of approach flow. The aspects of flow through screens that are of concern in a model study are (1) energy loss in the fluid passing through the screen, (2) modification of the velocity profile, and (3) production of turbulence. As all these factors could affect vortex formation in a sump with approach flow directed through screens, a proper modeling of screen parameters is important.

The fluid passing through the screen loses energy at a rate proportional to the drop in pressure, and this loss dictates the effectiveness of the screen in altering velocity profiles. The pressure drop across the screen is analogous to the drag induced by a row of cylinders in a flow field and can be expressed in terms of a pressure drop coefficient K (or, alternately, a drag coefficient), defined as[19]

$$K = \Delta p/(0.5\rho u_a^2) = \Delta H/(u_a^2/2g) \tag{8}$$

where Δp = drop in pressure across screen

u_a = mean velocity of approach flow to screen

ΔH = head loss across screen

The loss coefficient is a function of three variables: (1) screen pattern, (2) screen Reynolds number $R_s = u_a d_w/\nu$, where d_w is the wire diameter of the screen, and (3) solidity ratio S', the ratio of closed area to total area of screen (Section 11.1).

If S' and the wire mesh pattern are the same in the model and prototype screens, the corresponding values of K are a function of R_s only. This is analogous to the drag coefficient in a circular cylinder. At values of R_s greater than about 1000, K becomes practically independent of R_s.[19] However, for models with low approach flow velocity and fine wire screens, it is necessary to ascertain the influence of R_s on K for both model and prototype screens before selecting screens for the model that are to scale changes in velocity distribution.

Velocity modification equations relating the upstream and downstream velocity profiles usually indicate a linear relationship between the two, the shape and solidity ratio of the screen, and the value of K.[20] If wire shapes and solidity ratios are the same in model and prototype, it is possible to select a suitable wire diameter to keep the values of K approximately the same for the model and prototype screens in the corresponding Reynolds number ranges. This produces a head loss across the model screen that is scaled to the geometric scale of the model and that produces identical velocity modifications in model and prototype. Some model designers consider it a conservative approach to leave out the screens in the model altogether, under the assumption that this omission will only worsen the nonuniformity of the approach flow. This approach is not recommended unless it is not practically possible to select an appropriate model screen.

Modeling of Pumps The exterior submerged surfaces of the pump bowl assembly and, for wet-pit pumps, the column including the bell mouth must be modeled to scale as well as the interior geometry from the bell mouth perimeter to the impeller eye. This is to ensure that flow patterns approaching the impeller are properly simulated.

Prerotation induced by the pump's rotating element is discussed in Sections 2.1.3 and 13.1 and in References 21 to 24. It has been shown that the rotating element does not affect upstream flow patterns when the pump is operated at design flow, and hence,

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FIGURE 4A A 1:10 geometric scale model of a four pump-bay pump intake (Courtesy of Alden Research Laboratory, Inc.)

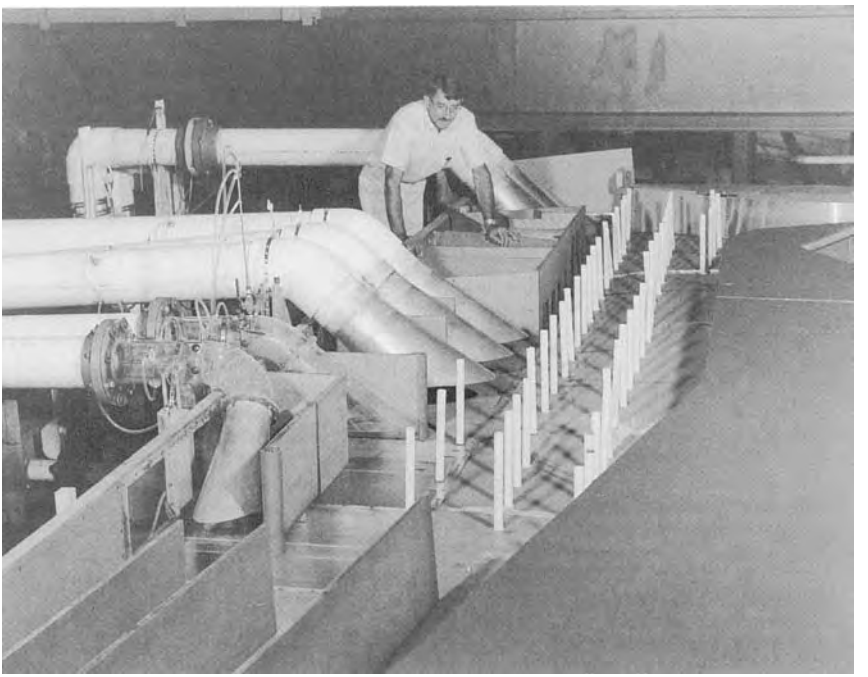


FIGURE 4B A 1:18 geometric scale model of a flood control pumping station with high capacity (800 to 1000 cfs/22.7 to 28.3 m^3/s), low head (about 12ft/3.65 m) pumps (Courtesy of Alden Research Laboratory, Inc.)

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including the rotating element in a pump intake study is not necessary. When the pump is operated at less than rated flow, a degree of swirl is induced upstream of the rotating element. This swirl increases rapidly at flow rates less than 45% of rated flow as reversed flow out of the impeller intensifies, and this may affect vortex activity in the pump well and flow distribution to the pump.

Model Operation Operating a model at the prototype suction pipe velocity is thought to be a conservative method to compensate for excessive viscous energy dissipation and the consequent less intense model vortices in a Froude model.[17] This method is often referred to as the Equal Velocity Rule. Operating a model at a higher than Froude scaled velocity should be considered a reasonable procedure for evaluating scale effects. However, operating a model based on the equal velocity rule may not be advisable unless the model is large enough, say at least a 1:4 scale model, because increasing the flow to many times the Froude scaled flow while keeping a scaled submergence could distort the approach flow patterns and turbulent intensities and, thus, cause unrealistic results. In general, a velocity increase to about 1.5 times the Froude scaled velocity can be considered reasonable. More appropriately, the information contained in References 25 and 26 may be used to decide the velocity ratio for exaggeration, which often is considered a function of model scale. If the final recommended design does not show any coherent core vortices in a model, no large scale effects should be expected, and operating the model at higher than Froude scaled flows may be unnecessary in such cases.[27]

Model Cost The cost of model studies varies considerably and is dependent on such factors as number of pump bays, number of operating conditions, and complexity of approach flow. Typically, $40,000 to $90,000 (in 1999 U.S. dollars) can be expected to cover the range from simple one- or two-bay sumps to multibay installations with complex approach flow and several operating modes.

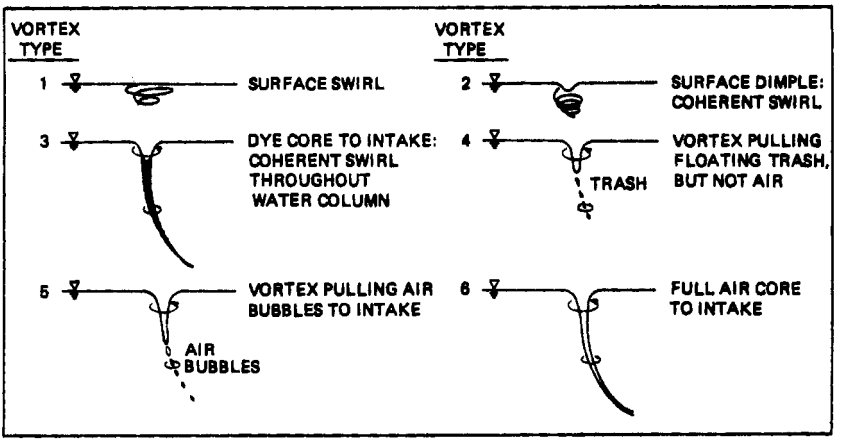
MODEL OBSERVATION TECHNIQUES

Free Surface Vortices Vortices in a model usually contain little energy, and therefore, vortex circulation cannot readily be measured because any measuring device will affect the strength of the vortex. Therefore, vortex strength is best characterized by visual observation and classified by comparison with a vortex strength scale, such as the one shown in Figure 5a. Observations are carried out continuously for a representative period of time, usually one hour prototype, and the resulting data are analyzed in terms of the maximum observed vortex strength and in terms of persistence; that is, the percentage of time a given vortex strength occurred.[28] Vortices of Type 5 and 6 are clearly objectionable in an intake design because of the effect air entrainment and localized pressure reductions have on pump performance. Usually in model studies, a vortex type of 3 is considered the maximum allowable, as prototype strength may be slightly underestimated as a result of possible scale effects.

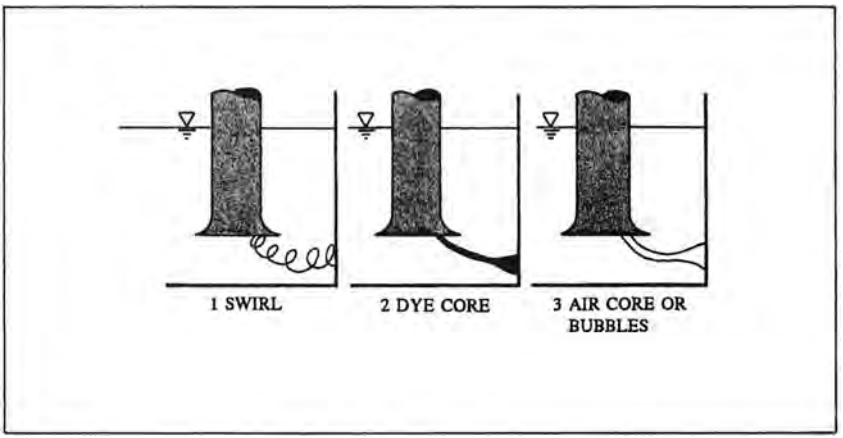
Subsurface Vortices Vortices attached to fixed boundaries below the surface can be difficult to detect and may require a careful search with a dye tracer or an air-injection technique. The vortex strength is best characterized by visual observations and classified by comparison to a vortex strength scale such as one shown in Figure 5b. A strong subsurface vortex may reduce the pressure locally within its core sufficiently low to allow dissolved air to come out of solution resulting in an air core (Type 3 in Figure 5b).

Submerged vortices are most commonly found below the center of wet-pit pumps and around splitter plates introduced to even out flow into the bell mouth. Pressure measurements made with electronic pressure transducers located in the floor under the pump may provide information on rapidly forming and disappearing vortices that cannot be detected with flow visualization. Subsurface vortices should, as far as practical, be eliminated from any design because their presence imposes fluctuating loads on the impeller and the resulting low-pressure areas may cause local cavitation damage to the pump.

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a. FREE-SURFACE VORTICES



b. SUBSURFACE VORTICES

FIGURE 5A and B Vortex strength scale (Courtesy of Alden Research Laboratory, Inc.)

Prerotation The tangential velocity component may be measured by a swirl meter, or vortimeter, which is an impeller with zero pitch in the axial direction mounted axially in the pump column two to four column diameters downstream from the suction bell. A typical vortimeter is shown in Figure 6. Vortimeter rotation is measured by either electronic or manual counting of rotation over a given time. Prerotation can also be determined from velocity traverses obtained with a two-dimensional pitot tube.

Intake Loss Coefficient Water manometers connected to piezometric taps along the pipe are used to measure the hydraulic gradient along the suction pipe. An intake loss coefficient is then determined by extrapolating the measured hydraulic gradient to the suction pipe inlet and computing the head loss $h_L = \Delta h - u^2/2g$, where Δh is the difference

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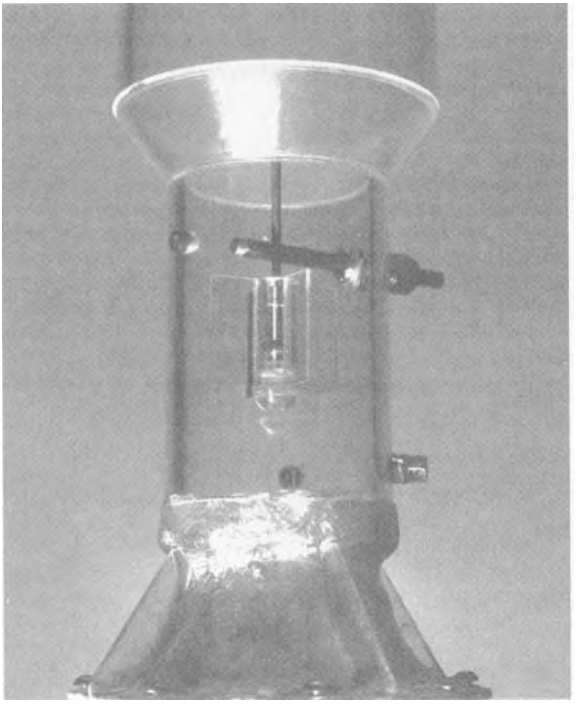


FIGURE 6 Swirl meter in a model pump column (Courtesy of Alden Research Laboratory, Inc.)

between the pump pit water surface and the extrapolated pressure gradeline at the pipe inlet. With this procedure, the computed head losses will not include any pipe frictional losses and consequently any model pipe Reynolds number influence is avoided. At the same time, pressure measurements are done far downstream from the region of flow separation near the entrance, and hence, more accurate measurements of total pressure loss, including entrance losses, are obtained. The loss coefficient C_L is usually defined as $C_L = h_l/(u^2/2g)$, where $(u^2/2g)$ represents the velocity head in the pipe. The computed model loss coefficient is valid for the prototype, and hence, the prototype inlet losses are C_L times the prototype pipe velocity.

Flow Distribution Radial flow under the perimeter of the pump bell is required to avoid vibrations and loss in efficiency caused by uneven impeller loading. Flow direction is usually indicated by six to eight yarn streamers equally spaced and mounted on wire supports 0.2 bell mouth diameter below the bell mouth perimeter. Figure 7 shows a typical installation. For an acceptable design, these streamers should not deviate more than 10 to 15° from radial. Velocity distribution approaching the pump is usually measured with miniature propeller meters.

CORRECTIVE MEASURES

Problems in pump sumps such as severe vortexing, intense swirl, or uneven flow distribution at the bell can be reduced or eliminated by deriving proper corrective methods using models. Because no single remedial method to correct these problems exists, it is important to try several suitable methods in the model to derive an effective and practical one.

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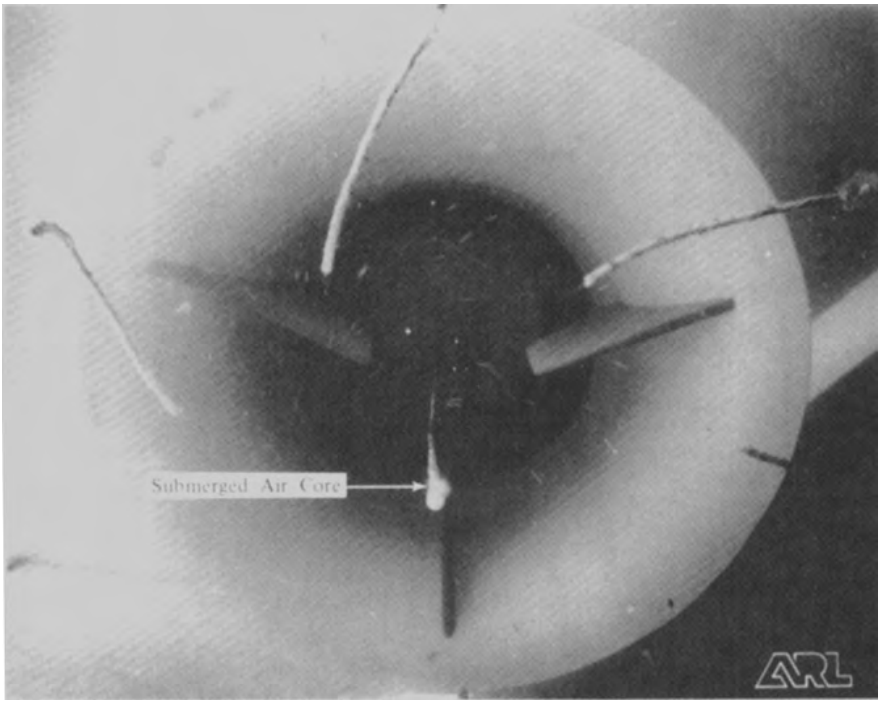


FIGURE 7 Yarn streamers around the suction bell of a model intake to observe flow direction into the bell (Courtesy of Alden Research Laboratory, Inc.)

Even though the pump bell velocity and submergence are very important parameters contributing to problems, it is seldom possible to change these values. Hence, the corrective measures are usually incorporated in the geometry of the pump pit and include the introduction of suitable appurtenant devices.

Severe free surface vortices may be prevented by providing a uniform approach flow to the pumps while maintaining a sufficient submergence. Required submergences can be predicted by empirical equations available in the literature.[31] Relocation of pumps, introduction of horizontal grids below the water surface, changing of wall and floor clearances, improvements in approach channel configurations, and changes in lengths and spacing of piers are some of the common techniques for reducing vortex activity. Surface vortices can also be controlled by installing a curtain wall immediately upstream of the pump. Figure 8 indicates various methods of eliminating poor flow conditions as given in Reference 7.

Submerged vortices are usually eliminated by installing splitter vanes or floor cones under the bell. Variations in floor and wall clearances can also be effective and should be tried in the model. Figure 9 shows a 1:4 scale model of a two-bay pump intake—each pump designed to draw 25,000 gpm (1.575 m^3/s). Flow distributor walls are installed to assure a nearly uniform flow to the pumps, and splitters and fillets are provided to guide the flow into the pump bells without any subsurface vortices.

The choice of corrective measures should always be made with the pump use in mind. For example, in testing pump intakes that will be used for sewerage, corrections that will allow trash build up, such as vanes or grids, should be avoided. It is also important that a continuous dialogue exist between the intake designer and the test laboratory so the most practical and inexpensive solution for a specific installation can be obtained with a minimum of effort.

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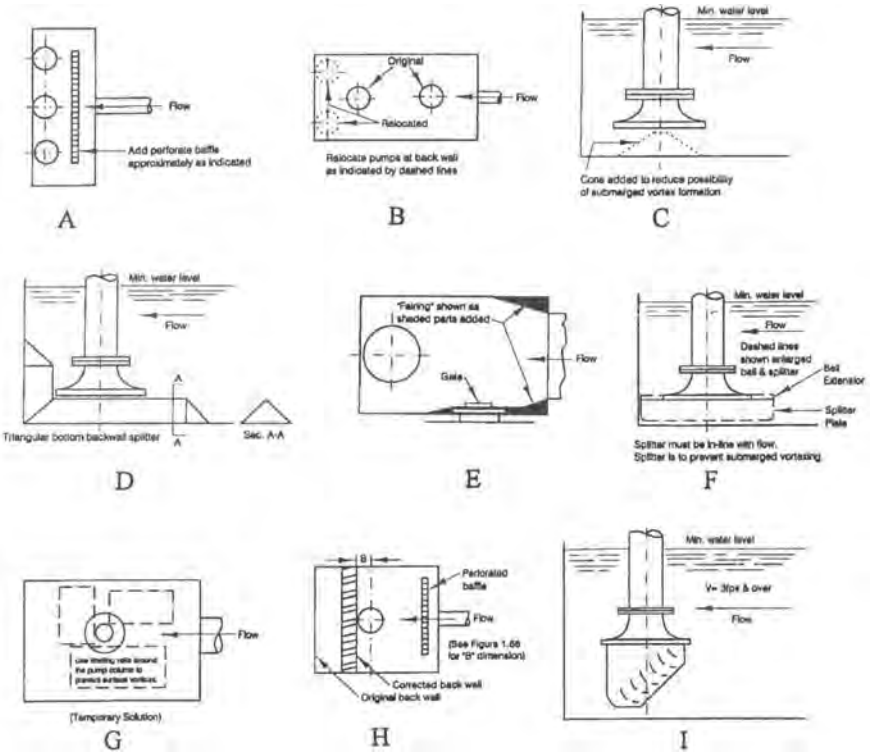


FIGURE 8A through I Common method of eliminating vortexing problems in pump pits (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 7)

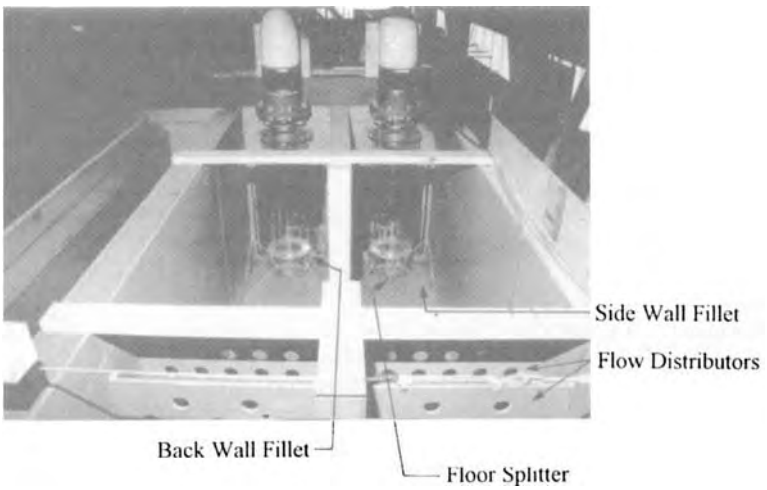


FIGURE 9 A 1:4 geometric model of a two-bay intake showing flow distributors, splitters, and fillets to eliminate objectionable vortices (Courtesy of Alden Research Laboratory, Inc)

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C • H • A • P • T • E • R • 14

SELECTING AND PURCHASING PUMPS

Trygve Dahl

Vinod Patel

INTRODUCTION

Enterprises engaged in the application, design, manufacture, and installation of pumping machinery are operating in an environment of increasing worldwide competition. The competitiveness of the firm's pumping machinery is substantially determined as a consequence of the system design, the selection and purchasing process for pumping equipment, as well as the ongoing operating and maintenance practices employed during the life cycle of the system. The design of the pump system, including the piping network and the components of the pumping system are performed first. Then, during the pump selection and purchasing process, purchasers expect manufacturers to design and offer high-quality equipment which meet their defined pumping requirements. Alternative pump designs that meet the technical requirements are compared based on total cost. This is either a simple evaluation based on total equipment cost or a detailed analysis taking into account the cost of energy, maintenance, and other total life cycle costs (LCCs). Finally, preferences are typically given to manufacturers that have a good reputation for quality and service and that deliver their product within an acceptable lead time.

The majority of the market for pumping equipment utilizes modular product families which are offered as premanufactured or assemble-to-order products, or customized into engineered-to-order (ETO) design variants. In such a design process, the "best" design is not necessarily globally optimum but rather, one that technically satisfies all of the functional requirements, is delivered in an acceptable period of time at the lowest evaluated cost, and meets other commercial constraints imposed by the purchaser. This "front-end" selection and purchasing process is crucial since the overall pump performance and long-term operating and maintenance costs of the equipment are determined by the purchasing decision.

This chapter is organized into two sections:

- *Steps in the Process.* This section describes the steps involved in specifying, selecting, and procuring a pumping system with particular emphasis on the pump and its associated appurtenances.
- *Emerging Trends in Selecting and Purchasing Pumps.* The information age is transforming the way the selection and purchasing process is performed. The emerging technologies and processes that will be more prevalent in the near future are outlined in this section.

STEPS IN THE PROCESS

The process in which a pump is selected and purchased varies considerably depending on factors such as the pumping application, the product architecture, the manufacturing quantity, and the degree of innovation required in the design process. These factors are illustrated in the following examples:

1. Consider a small pump used in a washing machine. These pumps are designed once to meet a single application and manufactured in quantities of hundreds of thousands of units a year. The pump is a fairly simple device, comprised of only a few parts, and requires little or no engineering design activity during the life span of the product. The major emphasis is on manufacturing and minimizing production costs.
2. Consider a line of bilge pumps used in recreational boats. A range of pumps are designed to meet a variety of pumping capacities and developed head (or pressure). Each model is designed based on preferred sizes established over years of sales. These pumps are designed for a well-understood application (i.e., bilge pumps for either fresh- or saltwater) and generally for a forecast demand. These pumps are not uniquely designed for each particular application but instead, chosen based on a catalog selection process.

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3. Consider a line of pumps conforming to API Standard 610/ISO 13709 "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." The applications addressed by these specifications are broad covering a wide range of centrifugal pump classes. Pump manufacturers satisfy this wide range of pumping requirements by configuring or adapting pre-engineered part modules to build up a unique pumping system. This adaptation of preexisting product range designs to meet unique application requirements is known as an ETO design process.
4. Finally, consider a boiler feed pump used in a central station combined cycle power generation application. There are only a limited number of these plant constructions underway at any given time in the world. The low manufacturing quantity of these types of pumps and specialized conditions of service precludes pre-engineering a line of pumps to satisfy this market. As a result, each pumping system is built from substantially new parts designed specifically to meet the needs of the application. Although each pump is new, the design concepts are fairly well known and rely to a large extent on modifying design concepts that were successfully applied in previous applications. This type of application also requires an ETO design process but one that contains more innovation than the previous example.

The last two examples typify the selection and purchase of ETO pumping equipment where each pumping system is usually a unique design that transforms a specific set of pumping requirements into a system that fulfills those requirements. This is the type of selection and purchasing process described in this section.

Once the initial decision that pumping equipment is required, the selection and purchase of the equipment can be divided into the following general steps:

- Engineering the pumping system
  - Defining the pumping requirements
  - Designing the pumping system
  - Pump specification and data sheet preparation
- Pump selection and configuration
  - Pump selection
  - Pump configuration
  - Pump quotation
- Evaluating and purchasing the pumping equipment
  - Evaluation of bids and negotiation
  - Purchase of the selected pump and driver

In the process of specifying pumping equipment, the engineer must determine system requirements and system head curves, select pump type, write the pump specification, complete the pump data sheet, determine testing, inspection and vendor drawing and data submittal requirements and develop all the data necessary to define the required equipment from the supplier.

Having completed this phase of work, the engineer is then ready to take the steps necessary to purchase the equipment. These steps include issuing the pump inquiry to the bidders, technical and commercial evaluation of pump bids, selection of the supplier, and release of data necessary to issue the purchase order. The ultimate result of this process is the selection of a pump/driver combination that satisfies both the process and mechanical requirements.

ENGINEERING THE PUMPING SYSTEM

Defining the Pump System Requirements The governing functional requirements of the overall pumping system must be defined. Generally speaking, a pumping system requires transporting a fluid between two given locations with sufficient pressure to overcome static elevation changes, system friction losses, and to provide the desired dynamic

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pressure at the end location. These functional requirements are generally described by the service, fluid type, operating conditions, site conditions, and industry specification.

SERVICE Service is a general description of the particular pumping system. Examples might include booster pump, boiler feed pump, gasoline transfer pump, and so on.

OPERATING CONDITIONS Operating conditions define the general fluid capacity, pressure, and elevation change required by the pumping system. At this point, the number of pumps required for the service may not be known, so the specific operating conditions or each pump in the system is not fully defined.

SITE CONDITIONS These describe the location of the pumps, the altitude, and utility conditions for the driver (steam, electrical, and so on).

FLUID TYPE AND PROPERTIES A thorough description of the fluid to be handled must be developed. This includes properties such as viscosity, density, vapor pressure, corrosiveness, erosiveness, volatility, flammability, and toxicity. Depending on the process and the system, some or all of these properties may have an important effect on the pump and system design. For example:

- The temperature, pressure, and corrosiveness of the fluid will strongly influence the materials of construction.
- The fluid viscosity will influence the choice of pump technology, where low viscosity fluids are generally suitable for centrifugal pumps while higher viscosity fluids are better handled by positive displacement pumps.
- If the fluid contains solids in suspension, suitable types of pump seal designs and abrasion-resistant pump construction must be considered.
- Erosion due to high particle content may cause premature performance decline. Large particles may favor open impeller design.
- Fluid toxicity may necessitate the use of dual (tandem or double) mechanical seals due to government regulations or safety considerations.
- Entrained gases may affect the pump's ability to produce the required differential pressure.

The specified fluid physical and chemical properties must cover the entire expected operating range of the pumping system. Influences such as varying temperatures and pressures must also be defined.

INDUSTRY SPECIFICATION In many cases, the pumping application is governed by, or conveniently referenced to some type of industry specification. The design, construction, rating, and testing of most pumps used in refining and chemical industries are governed by standards such as API (American Petroleum Institute), ASME (American Society of Mechanical Engineers), the Hydraulic Institute, NFPA (National Fire Protection Association), PIP (Process Industry Practices), ISO (International Organization for Standardization), and various other international standards. The severity of the service in which the pump will be applied, as well as the location of the plant, will determine which industry standard (or standards) will be used, if any.

In the case of a fire pump service, NFPA compliance might be mandatory to meet the user's insurance company requirements. If a pump will be installed in an oil refinery or chemical plant, API and/or ASME standards will be applied depending on the severity of the service and client preferences. International standards such as DIN (German), BS (British), JIS (Japanese), or ISO can also be used. These standards are intended to provide a pump with a level of quality to match the needs and expectations of the end user of the equipment. It is obvious that the quality requirement for an emergency feedwater pump in a nuclear power plant needs to be much more stringent than a potable water booster pump in an office building. The quality issues covered by these codes/standards range from detailed design issues to inspection and performance testing requirements.

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Many end users, engineering consultants, and EPCs (engineering, procurement, and construction) firms have also developed their own specifications for different types of services that document best practices for those applications. These should always be referenced before commencing with the overall system design.

EVALUATION CRITERIA Before starting the pump system design, it is important to formalize the evaluation criteria. The engineer and the pump manufacturer shall be given guidance with respect to technical requirements, commercial requirements, lead time requirements, and the relative weighting of these factors. Further, as a "systems perspective" is gaining more recognition, the total LCC is becoming more recognized in the commercial evaluation, considering initial capital costs as well as ongoing operating and maintenance costs.

Designing The Pumping System

SYSTEM HEAD CURVES The engineer must have a clear understanding of the process and system in which the pumping equipment will operate. A preliminary design of the system should be made and should include an equipment layout and a piping and instrumentation diagram (P&ID). These preliminary drawings will show the various fluid flow paths for system operation, preliminary pipe diameters and lengths, relative elevations of system components and all valves and other piping components that will be used to establish the system head losses. These drawings will be used by the engineer to calculate the final piping sizes and pumping system head requirements.

With this information, the engineer can develop system head curves that show the relationship between flow rate and hydraulic losses in the piping system. In determining the hydraulic losses, the engineer must include adequate allowances for future corrosion and scale deposits in the piping system over the plant life.

Because hydraulic losses are a function of flow rate, pipe size, and layout, each individual flow path alignment in a given system will have its own characteristic operating curve. Care must be taken when specifying the required pump characteristics to take into account all possible system operating flow paths. It is convenient to add the effects of static pressure and elevation differences in the system to form a combined system head curve. This combined curve shows the total head required of the pumping equipment to overcome system resistance as well as differential static pressure and elevation. The pump head must be at or above the combined system curve at all required operating points and fluid conditions for the various system flow paths. Refer to Sections 11.1, 11.2, 12.1, and 12.2 for guidance in constructing system head curves.

MODES OF SYSTEM OPERATION System operating modes are important considerations when specifying pumping equipment. Will the pump be used in continuous or intermittent operation? Will the pump operate in parallel or series with other pumps? Will there be significant differences in head or flow rate requirements in different system alignments? Will a single pump be used as a common spare for two different pumping applications? These and other questions arising from analyzing the different modes of operation will help influence decisions as to the number of pumps needed, heads and capacities and whether booster pumps are desirable in some systems. It should be noted that unnecessarily conservative hydraulic requirements may increase pump complexity (such as the selection of a more elaborate multistage or double suction pump in place of simpler single stage, overhung pump) and increase the associated acquisition and operating cost.

The engineer should also consider plant maintenance service intervals in the design of the pumping system. This factor will influence the decision of quantity, pump type, requirement for installed spare(s), and the manufacturing quality required of the specified pumps. Frequently, due to the critical nature of a pumping service where high reliability is necessary, installed spares are provided. In some cases, 2 to 100% pumps are provided. When system flow rate requirements fluctuate, 3 to 50% pumps may be called for. When reduced flow rates will not adversely affect operations, 2 to 50% pumps can be specified. Plant operating philosophies will dictate if automatic start of a spare pump is required.

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PUMP FLOW/HEAD MARGINS Pumps are normally specified with a capacity margin above what has been determined necessary for the process. In addition, the calculated system head losses are also determined conservatively. The reasons for this include the following:

- During system design, many assumptions are made while determining pump requirements, some of which might eventually be determined to be incorrect.
- During the plant life cycle, process conditions are likely to change due to aging catalyst, changes in feed stock, or seasonal feed temperature variations.
- Final piping design may be significantly different from preliminary design.
- System hydraulic losses may change due to corrosion.
- Plans for future operating condition changes may be hampered if the piping system does not provide for adequate net positive suction head at the future flow conditions.

During preliminary system design, these potential future changes in head/capacity must be studied to determine the required design margin. On the other hand, care must be taken when applying margins to ensure that the purchased pump is not oversized. If the total head produced is too large, impellers may be trimmed within the allowable range of the pump model provided. Similarly, a pump should be selected to operate close to its best efficiency point, it is important to minimize the selected margin. Margins of 5 to 10% on flow are typical but even 20% is common, for example, in reflux tower service. In cases where the process is well proven and understood, and system operating requirements are well defined, a zero margin is sometimes appropriate. If the flow is significantly oversized, a costly energy penalty can result over the life of the plant. This is caused by lower pump operating efficiency at off-design flow rates and discharge throttling losses that may be necessary to control the flow rate to the desired value.

TYPE OF PUMP CONTROL The type of control for the required pump is also an important consideration for pump specification and selection. Since the actual piping system usually incorporates a design margin, a control valve is normally employed in centrifugal pump applications. This control valve (not typically supplied by the pump manufacturer) is used to adjust the system curve over the life of the unit. Flow sensing control provides the most stable operation for most systems. Pressure control can have very large swings in flow when operating in a flat or drooping area of a centrifugal pump curve. For this reason, most centrifugal pump specifications incorporate the requirement for the centrifugal pump operating curve to rise continually to pump discharge valve shut in (also known as the pump shut off). Variable speed pump operation is used more commonly as advances in variable frequency drives and control methods are more commercialized (see Figure 1). Also, the benefits in operating control, the possible elimination of control valves, and the lower energy costs obtained by an improved choice of operating speed is also contributing to the popularity of variable speed pumping (Hydraulic Institute and Europump, 2004).

Both temperature and level sensing control can lead to a pump running at shutoff or at the end-of-curve condition during upset or failure modes. Centrifugal pumps with a 10 to 20% head rise between the specified operating point and the shutoff point may be preferred for some services such as for parallel pump operation.

Pumps that may operate in the shutoff condition may need a continuously open bypass to prevent pump damage. Pump manufacturers will advise of the minimum required flow for an offered pump, but many end users require an additional margin above the manufacturers recommended minimum flow. If a continuous flow bypass is incorporated in the system design, this additional flow must be added to the specified pump requirement.

FUTURE SYSTEM CHANGES A final factor to be considered is the possibility of providing for future system changes. When future system changes can be predicted with a degree of certainty, the system can be designed with this in mind. Rather than selecting a pump that is operating at the high end of its preferred operating region, a larger impeller diameter or the next larger size pump operating at the beginning of its preferred operating range might be considered. In addition, the capability for installing a larger diameter

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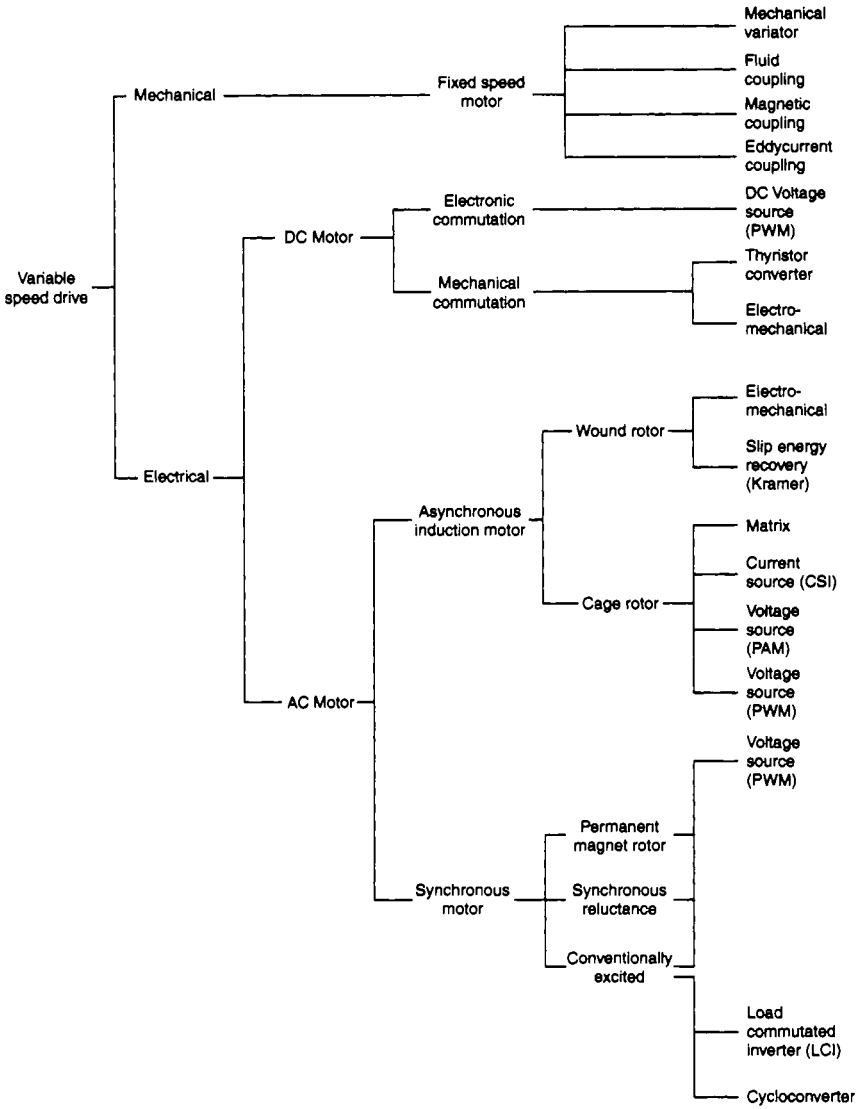


FIGURE 1 Types of variable speed drives (Courtesy Hydraulic Institute)

impeller to handle future higher head requirements must be considered. Because minimizing capital costs for a project is usually the prime consideration, oversizing a pump for future operations is not normal practice. Pumps are expected to operate efficiently and reliably in the present system, and this fact should be noted during the selection process.

Pump Specification and Data Sheet Preparation The pumping system requirements and design are typically documented in the pump specification and pump data sheet. This

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documentation is created by the purchaser and conveyed to the pump supplier, forming the basis for the request for quotation (RFQ).

REQUEST FOR QUOTATION In the RFQ, it is necessary to fully define the scope of supply for the desired piece of equipment. To do so requires the right amount of documentation provided by the purchaser to match the type of equipment being purchased. In some cases, the bid requisition can be as simple as a one-page listing of the pump requirements or it may be a document that incorporates data sheets, technical specifications, shipping specifications, purchasing terms and conditions, vendor drawing requirements, and any other document that will help define the full requirements of the intended purchase. A fully detailed requisition for a complicated pumping service might include over 100 pages of requirements.

As a minimum, the requisition must include a clear scope of supply, applicable specifications and data sheets, and commercial terms and conditions. In addition, the requisition can be used to specify additional requirements that have not been adequately addressed in other documents. These are described in more detail below.

PUMP TECHNICAL SPECIFICATION As discussed previously, there are many different designs of pumps that can be purchased. Engineering and operating companies may have standard technical specifications for all categories of equipment that are commonly purchased. When a pump service is being prepared for bids, a technical specification can be used in standard form, or updated to incorporate special requirements based on client or other project specific needs.

A technical specification is usually written to cover a wide range of equipment within a given category. Because many variations exist within any equipment category, the technical specification is usually written to be applicable to this range of equipment. Data sheets may be used to provide the specific requirements not covered in the technical specification.

Most technical specifications are written as performance specifications rather than design specifications. Care must be exercised to ensure that the pump manufacturer (not the purchaser) is responsible for the actual equipment design. If design requirements rather than performance requirements are specified, the vendor may claim that he is not responsible when performance requirements are not met. Care must be taken to assure that the equipment selected is from qualified vendors with proven design experience in similar applications. The purchaser must resist the temptation to force the vendor to modify a proven design and build a pump with which the vendor does not have proven experience.

As a minimum, a technical specification must list the following:

1. Industry codes and standards to which the pump must be designed, constructed, and tested. It is also possible to create a technical specification that stands alone and does not reference (or minimally references) codes and standards. This stand-alone technical specification will require a more thorough listing of needed requirements than one that references common design, fabrication, and testing standards.
2. A list of desired deviations and preferences from the main technical standards used to help define the pump requirements.
3. A definition of any technical terms used in the specifications that are not defined in referenced standards. This will help prevent misunderstandings between the purchaser and pump vendors.
4. A list of documents to be submitted by the successful bidder, both with the quotation and after the order is placed.

Pump Specification and Data Sheet

PUMP DATA SHEET The industry uses various layouts of equipment data sheets to collect, organize and exchange the technical information about the functional and physical requirements for pumping equipment. These data sheets range from company specific formats to industry standard layouts. The data sheet is used to list specific requirements for each individual pump service. These requirements are not general enough to be listed in

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the technical specification. As a minimum, the following items must be included on the data sheets for each service:

1. Pump name and item number
2. All required system design conditions and pump performance values needed to fully define the pump operating requirements, including any known turndown, start-up, or upset conditions
3. Materials of construction (unless the vendor is free to recommend standard materials)
4. All accessory requirements (unless the vendor is free to offer the standard selections) such as driver, coupling, seal, and packaging needs
5. Utility conditions (e.g., electrical voltage, frequency, and so on)
6. Intended site location and annual environmental conditions in this location
7. Electrical hazardous area classification
8. Instrumentation and control requirements (usually out of the vendors scope)
9. Noise requirements
10. Any specific preferences that deviate from requirements of the technical specifications
11. Inspection and testing requirements

In addition, the vendor should be advised to fill in the applicable blanks in the data sheet and submit the proposal. The manufacturer's data will help define the offering. See Figure 2 for excerpts from an API-610, 10th edition / ISO 13709 data sheet.

OTHER REQUISITION DOCUMENTS AND REQUIREMENTS A purchaser may include many additional options in the bid requisition. These requirements may be included in the requisition narrative or data sheets. Some possible options are addressed in the following paragraphs.

1. *Alternates.* It is extremely difficult for a specification to cover all possible pumps offered by various manufacturers. In addition, the pump industry constantly updates their products due to competition and revisions to industry standards. Because of this, it is good practice to allow manufacturers to offer alternatives to the specified pumping equipment. This allows manufacturers to present their best offerings and gives the purchaser the advantage of obtaining commercially and technically attractive alternate offerings. However, the choice of whether to accept an alternative is retained by the purchaser.
2. *Vendor data requirement (VDR) form.* For complicated equipment where many different documents are requested for submittal by the purchaser, a list of these documents is usually included in the requisition. This list should include a listing of each generic document, the number of copies required and the time after placement of purchase order that the document is required by the purchaser. Requirements for drawings are significantly different for an end user maintenance group and an engineering contractor. Because documentation can add a significant amount to the purchase cost, care should be taken when specifying required drawings and drawing quantities.
3. *Inspection and testing checklist.* For critical and special pump applications, the purchaser may require significant shop inspection and testing of the equipment above the vendor's standard procedures. This might include materials certification, NDE, welding inspections, rotor balancing, inspection reports, and mechanical/performance tests.
4. *Noise data sheet.* If the equipment is large enough or of a type known to be a noise concern, the purchaser may specify the maximum allowable noise level attributable to the equipment being purchased. One way to accomplish this is to include a noise data sheet that lists the allowable levels for the equipment provided. This data sheet should be designed to require the vendor to fill in the expected noise level for the equipment being offered. If the expected noise is greater than the specified maximum value, the vendor should be directed to provide special design options (i.e., noise enclosure, WP II motor special noise ducting, and so on) to lower the expected noise level.

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PAGE 1 OF

CENTRIFUGAL PUMP
API 610/ ISO 13709 TENTH EDITION
PROCESS DATA SHEET
US CUSTOMARY UNITS

JOB NO. _____ ITEM NO. (S) _____
REQ/SPEC NO. _____ /
PURCH ORDER NO. _____ DATE
INQUIRY NO. _____ BY

APPLICABLE TO: ○ PROPOSALS ○ PURCHASE ■ AS BUILT
FOR _____ UNIT _____
SITE _____ SERVICE _____

NOTES: INFORMATION BELOW TO BE COMPLETED: ○ BY PURCHASER □ BY MANUFACTURER ■ BY MANUFACTURER OR PURCHASER

○ DATA SHEETS

	ITEM NO.	ATTACHED	ITEM NO.	ATTACHED	ITEM NO.	ATTACHED
PUMP		○		○		○
MOTOR		○		○		○
GEAR		○		○		○
TURBINE		○		○		○

APPLICABLE OVERLAY STANDARD(S):

REVISIONS

NO.	DATE	BY
1		
2		
3		
4		
5		

○ OPERATING CONDITIONS (5.1.3)

FLOW, NORMAL _____ (gal/min) RATED _____ (gal/min)
OTHER _____
SUCTION PRESSURE MAX/RATED _____ / _____ (psig)
DISCHARGE PRESSURE _____ (psig)
DIFFERENTIAL PRESSURE _____ (psi)
DIFF. HEAD _____ (ft) NPSHA _____ (ft)
PROCESS VARIATIONS (5.1.4) _____
STARTING CONDITIONS (5.1.4) _____
SERVICE: ○ CONT. ○ INTERMITTENT (STARTS/DAY) _____
○ PARALLEL OPERATION REQD (5.1.13)

○ SITE DATA (5.1.3)

LOCATION: (5.1.30)
○ INDOOR ○ HEATED ○ OUTDOOR ○ UNHEATED
○ ELECTRICAL AREA CLASSIFICATION (5.1.24/6.1.4)
CL _____ GR _____ DIV _____
○ WINTERIZATION REQD. ○ TROPICALIZATION REQD.
SITE DATA (5.1.30)
○ ALTITUDE _____ (ft) BAROMETER _____ (Psia)
○ RANGE OF AMBIENT TEMPS: MIN/MAX. ____ / ____ (°F)
○ RELATIVE HUMIDITY: MIN/MAX _____ / _____ (%)
UNUSUAL CONDITIONS: (5.1.30) ○ DUST ○ FUMES
○ OTHER _____

○ DRIVER TYPE

○ INDUCTION MOTOR ○ STEAM TURBINE ○ GEAR
○ OTHER _____

○ MOTOR DRIVER (6.1.1/6.1.4)

■ MANUFACTURER _____
□ _____ (HP) □ _____ (r/min)
□ FRAME ■ ENCLOSURE _____
■ HORIZONTAL ■ VERTICAL ■ SERVICE FACTOR _____
■ VOLTS PHASE/HERTZ _____ / _____ / _____
○ TYPE _____
○ MINIMUM STARTING VOLTAGE (6.1.5)
■ INSULATION _____ ○ TEMP. RISE _____
■ FULL LOAD AMPS _____
■ LOCKED ROTOR AMPS _____
■ STARTING METHOD _____
■ LUBE _____

BEARINGS (TYPE/NUMBER):
□ RADIAL _____ /
□ THRUST _____ /
□ VERTICAL THRUST CAPACITY
UP _____ (lb) DOWN _____ (lb)

○ LIQUID (5.1.3)

LIQUID TYPE OR NAME _____
○ HAZARDOUS ○ FLAMMABLE ○ _____ (5.1.5)

	MIN.	NORMAL	MAX.
PUMPING TEMP (°F)			
VAPOR PRESS (psia)			
RELATIVE DENSITY (SG):			
VISCOSITY (cP)			

SPECIFIC HEAT. Cp _____ (Btu/lb °F)
○ CHLORIDE CONCENTRATION (6.5.2.4) _____ (ppm)
○ H_2S CONCENTRATION _____ (ppm) WET (5.12.1.12c)
CORROSIVE/ EROSIVE AGENT _____ (5.12.1.9)

MATERIALS (5.12.1.1)

○ ANNEX H CLASS (5.12.1.1) _____
○ MIN DESIGN METAL TEMP (5.12.4.1) _____ (°F)
○ REDUCED HARDNESS MATERIALS REQD (5.12.1.12)
□ BARREL/CASE _____ IMPELLER _____
□ CASE/IMPELLER WEAR RINGS _____
□ SHAFT _____
□ DIFFUSERS _____

■ PERFORMANCE:

PROPOSAL CURVE NO. _____ □ R/MIN _____
□ IMPELLER DIA. RATED _____ MAX _____ MIN _____ (in.)
□ IMPELLER TYPE _____
□ RATED POWER _____ (5hp) EFFICIENCY _____ (%)
□ MINIMUM CONTINUOUS FLOW:
THERMAL _____ (gal/min) STABLE _____ (gal/min)
□ PREFERRED OPER. REGION _____ TO _____ (gal/min)
□ ALLOWABLE OPER. REGION _____ TO _____ (gal/min)
□ MAX HEAD@RATED IMPELLER _____ (ft)
□ MAX POWER@RATED IMPELLER _____ (BHP)
□ NPSHR AT RATED FLOW _____ (ft) (5.1.10)
○ MAX. SUCTION SPECIFIC SPEED: _____ (5.1.11)
○ MAX. SOUND PRESS. LEVEL REQD. _____ (dba) (5.1.16)
■ EST MAX SOUND PRESS. LEVEL _____ (dba) (5.1.16)
■ EST MAX SOUND POWER. LEVEL _____ (dba) (5.1.16)

○ UTILITY CONDITIONS (5.1.3)

ELECTRICITY	VOLTAGE	PHASE	HERTZ
DRIVERS			
HEATING			

SYSTEM VOLTAGE DIP ○ 80% ○ OTHER _____ (6.1. 5)

STEAM	MAX. PRESS.	MAX. TEMP.	MIN. PRESS.	MIN. TEMP.
DRIVERS				
HEATING				

COOLING WATER: (5.1.19) SOURCE _____
SUPPLY TEMP. _____ (°F) MAX. RETURN TEMP. _____ (°F)
NORM. PRESS. _____ (psig) DESIGN PRESS _____ (psig)
MIN. RET. PRESS. _____ (psig) MAX. ALLOW. D.P. _____ (psi)
CHLORIDE CONCENTRATION: _____ (ppm)

FIGURE 2 API 610 / ISO 13709 Data sheet—page 1 (Courtesy American Petroleum Institute)

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COST EVALUATION The "total cost" of the pumping equipment, in the most simplistic sense, is the purchase price of the equipment as quoted by the pump supplier. Many purchasers base their economic purchasing decision on this purchase price alone. More commonly purchasers are including an energy evaluation or even an LCC evaluation in their cost evaluation.

ENERGY EVALUATION The purchaser may choose to include operating costs in the equipment evaluation. If so, the vendor needs to be informed of this requirement so the most attractive proposal can be offered. In this case, the overall equipment cost will be a combination of first cost and the differential cost of energy used over a specified period of time. If so, the vendor must be informed of the present worth payback time for this analysis (such as 3 years), cost of a unit of energy, and the number of hours per year that the equipment will operate. Based on the differential power consumed over the payback period and the energy cost of this differential, the penalty or credit can be easily calculated.

LCC EVALUATION A more sophisticated LCC evaluation method is gaining acceptance in the industry. Traditionally, cost evaluations have been limited to the cost of the initial capital equipment plus some evaluation for the cost of energy. An LCC cost equation, developed by the Hydraulic Institute, is represented by the following equation:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_s + C_{env} + C_d$$

where

- C_{ic} = initial costs, including the purchase price of the pump set (pump(s), motor, motor starter, and so on)
- C_{in} = installation and commissioning cost (includes installation costs such as foundations, electrical installation, piping, isolation valves, and so on)
- C_e = energy cost (usually electricity because most pumps are driven by electric motors)
- C_o = operating costs (includes labor cost of system supervision and operating costs such as cooling water, steam heating, centralized oil lubrication systems, and so on)
- C_m = maintenance (includes routine servicing and unplanned repairs)
- C_s = down time costs (potential loss of income due to unplanned downtime, sometimes neglected if a spare pump is included in purchase price)
- C_{env} = environmental cost (contamination from pumped liquid and auxiliary equipment)
- C_d = disposal/decommissioning costs (to dispose of the equipment, in an environmentally responsibly way)

The LCC is usually discounted to a present value, based on an assumed discount rate, inflation rate, and expected equipment life. See the section titled "Life Cycle Cost Calculations" and Figure 18 later in this chapter for more information on this topic.

COMMERCIAL TERMS AND CONDITIONS The requisition should include the following commercial terms and conditions:

- Name of buyer, place to which proposals must be delivered, information on ownership of documents, time allotted for submission of bids, governing laws and regulations.
- Location of plant site.
- Site storage conditions and anticipated length of storage (preparation requirements).
- Schedule for submittal of drawings/documentation, and pump delivery.
- Guaranty/warranty requirements.
- Instruction on minimum information to include in the proposal, number of copies that vendor must provide, status of alternative offerings, and a statement on the owners right to accept or reject bids that are not in accordance with the bid package.

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- Acceptable terms of payment.
- Method of transportation to site that establishes the vendor's responsibility. If the vendor will not be responsible for providing shipping, he must be requested to provide enough information in the quote for others to estimate shipping costs.
- Customer shop inspection requirements.
- Any penalty or bonus related to late or early shipping.

The list of specifications and data sheets must include document name or description, document number, revision number, and revision date. This will provide a record of what documents have been sent to a vendor.

Formalizing the RFQ Submittal

BIDDERS LIST PREPARATION The preparation of a bidders list for the required equipment on a project is of utmost importance. This list is comprised of suppliers that the client and purchaser agree are qualified to supply the needed equipment. Qualification requirements may include a variety of categories such as the following:

- Vendor proven successful experiences supplying similar equipment
- Vendor ability to meet required delivery schedule
- Vendor ability to support field installation
- ISO 9001 certification standing
- Location of manufacture or material supply
- After sales service, spare parts supply

If a proposed vendor is not known to the purchaser, a shop survey may be required to satisfy the purchaser of a potential supplier's ability. This type of survey will usually include an inspection and assessment of the manufacturer's shops, engineering and design facility, and quality assurance organization. The complexity of the required equipment and the project needs will determine the level of qualification required of the vendors. Remember that a vendor is required to provide not only the specified equipment, but may also need to provide civil, piping, and electrical interface information required for project design. A vendor must possess the skills necessary to support all of these requirements.

The number of vendors that are to receive bid requests is usually between three and five. This number will provide for effective competition and will limit the quantity of quotes that require evaluation.

Pump alliance agreements, or the need to duplicate existing equipment, may reduce the list to a single bidder. In addition, unique design or vendor experience may dictate a sole source supplier.

EVALUATION CRITERIA Upon preparing the formal RFQ submittal to the selected pump suppliers, the purchaser should clearly define the criteria in which the suppliers' quotations will be evaluated. This helps the suppliers optimize their pump technical and commercial offer to meet the purchasers' requirements.

In summary, the evaluation criteria for the selection and purchase of pumping equipment should include:

- *Satisfaction of technical requirements.* This is found in the pump specification, data sheets, and other requisition documents and requirements.
- *Minimum total cost.* This is described by the purchaser as either a purchase price evaluation, energy evaluation, or the more sophisticated LCC evaluation.
- *Commercial terms and conditions*

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PUMP SELECTION AND CONFIGURATION

Once the RFQ (or *inquiry*, as it is often called) has been prepared by the pump purchaser, a pump quotation (or *bid)* must be obtained from one or more qualified pump manufacturers. The quotation will contain both technical and commercial information, sufficiently detailed to allow the pump purchaser to make a purchasing decision.

The primary tool that pump manufacturers have traditionally used to prepare a quotation is commonly known as the "price book." One might envision a price book as a simple price list which sales engineers use to quote pumps. In the most routine case, this is true. More often, however, the price book is a technical manual which contains extensive technical information such as pump performance curves, materials of construction tables, engineering data including dimensions and cross-sectional drawings, a plethora of guidelines on proper application of pumping equipment, and yes, pricing information. Essentially, the price book is an engineering design, specification, and pricing manual traditionally used by pump manufacturers to support the preparation a pump quotation.

Generating a pump quotation is described by the following three-step process:

1. *Pump selection*. Generating the "best" pump selection for the given set of customer requirements and specifications
2. *Pump configuration*. Generating the pump scope of supply, including supporting documents such as data sheets, price calculations, and drawings
3. *Pump quotation*. Preparing a complete quote package, including sufficient technical and commercial data to support a purchasing decision

Pump Selection Pump selection is a process that follows the following major steps:

1. Establish the conditions of service (COS)
2. Identify the most suitable product lines that might satisfy the COS
3. Search all candidate pump sizes within the selected product lines toward satisfying the COS
4. Size and calculate the complete performance characteristics for all candidate pumps

Skilled pump applications engineers typically perform the pump selection process. This process has been historically accomplished with aid of a technical manual and hand calculations. More recently, computer-aided selection programs are used to automate the procedure and assure a more accurate and optimized result. Examples are shown in the sections below.

Pump Conditions of Service The pump conditions of service are initially acquired directly from the purchasers pump data sheet and specifications. Unless the specification dictates the type of pump, the COS are described without prejudice to the type of pump for the service. This allows the manufacturer to consider a broader range of pump designs, each having their own inherent strengths and weaknesses. A typical COS list is shown in Figure 3 and outlined in the following paragraphs.

BASIC OPERATING CONDITIONS Hydraulic performance for pumps are established by the rated pump capacity (flow rate), the rated differential head (or pressure) required, and driver speed selection strategy to bound the allowable range of operating speeds of the pump. These parameters are sufficient to perform a basic hydraulic selection defining the pump size and type, impeller design and impeller trim, casing or diffuser design, the number of pump stages and the operation speed.

ADVANCED OPERATING CONDITIONS Advanced operating conditions extend beyond the basic hydraulic selection by introducing parameters that affect the pump design. These

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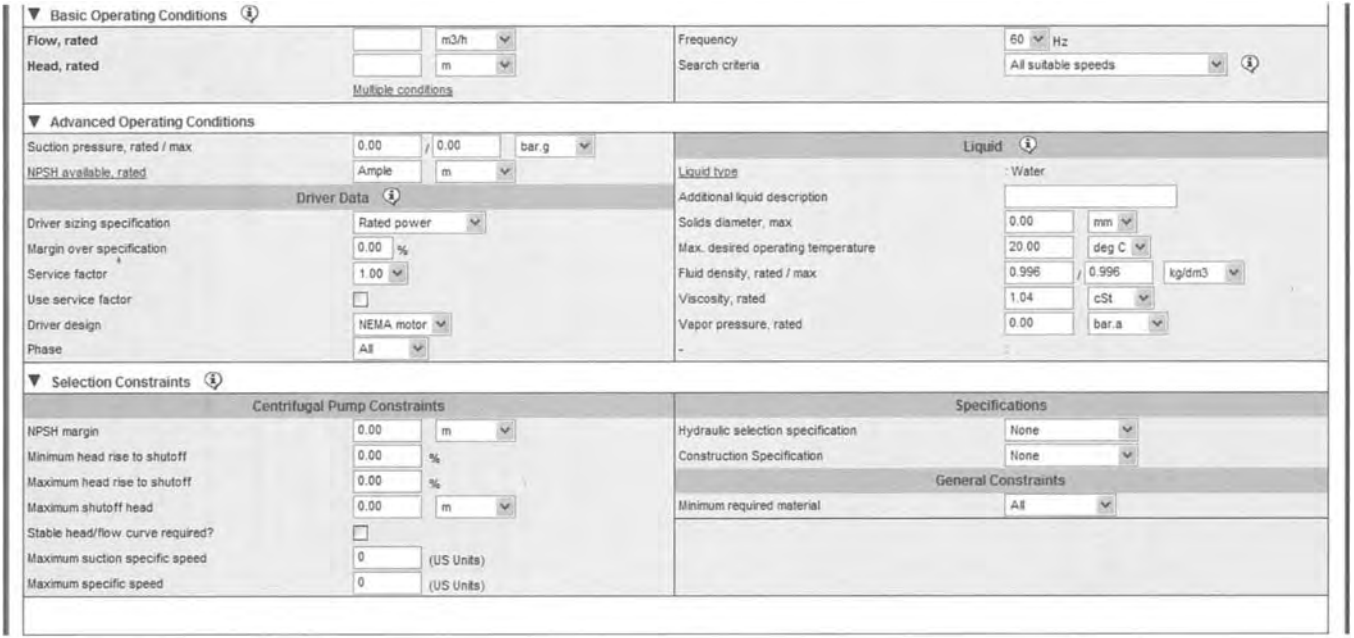


FIGURE 3 Pump conditions of service (Courtesy Intelliquip, LLC)

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parameters include suction pressure, fluid characteristics, net positive suction head available (NPSHA) and driver design. These parameters strongly influence the suitability of various pump types.

FLUID CHARACTERISTICS Fluid characteristics such as viscosity, density, vapor pressure, volatility, chemical stability, solid content, and entrained gases are important factors to be considered for proper pump selection. Pumps are available to handle a full range of fluid types, but each pump type has its limitations. A positive displacement progressing cavity pump can be used to pump toothpaste, peanut, butter, and shampoo, but it will not usually be a good choice for pumping water or gasoline. A rotary gear pump or variable displacement piston pump is a good choice for a hydraulic control system, but not for a potable water application. A rotary sliding vane pump can be successfully applied for pumping hot asphalt and for limited application in a lube oil system.

Making the best pump selection for a certain fluid application is often difficult. Previous successful experience is usually the best guideline for proper pump selection. This information can be obtained from end users, from process licensers, and from pump manufacturers. Recommendations from all of these sources should be carefully considered.

PUMP MATERIALS Material selection is affected both by the pumped fluid and the environment. Resistance to corrosion and erosion are of prime importance. The engineer must determine which material is most suitable and economical for a particular service. This requires that an evaluation comparing the more expensive longer life material to a less expensive material, which may provide a shorter pump life. Requirements such as continuous or intermittent operation, critical or noncritical service and plant life cycle should be considered when selecting materials.

Pumps are commonly available in cast iron, ductile iron, bronze, carbon steel, alloy steels, and in some cases composite materials or special alloys such as Monel, Hastelloy, or titanium. In addition to the importance of pump design life, safety must also be considered when selecting materials. Cast iron construction is not used for pressure casing parts of pumps that are to handle flammable or hazardous liquids because cast iron is more subject to fracture when thermally shocked. For these services, pressure-casing parts must be high strength ductile materials such as carbon or alloy steel.

DRIVER SELECTION The choice of driver type for a pumping service is as important as the pump selection. Factors that affect the driver choice are capital cost, driver type availability, operating reliability and the availability and cost of utilities.

Constant speed electric motors are most economical when only the first cost is considered. Often there is excess steam available within a facility that, when compared to the cost of electricity, will justify the extra cost of a steam turbine. Reliability requirements may necessitate the use of both a steam driven main pump and an electric motor-driven backup pump. In the case of firewater pumps, a battery-start, diesel-fueled internal combustion engine is needed to be completely independent of plant utilities. More expensive variable speed electric motors can sometimes be justified if the pump is operated well below its design conditions and there is the potential for significant savings in power.

There are other factors that should be considered when selecting the pump driver. The capital cost as well as the installation cost is more expensive for a steam turbine due to required piping. Steam turbines also require more maintenance during plant life, which may be undesirable to the owner. Selection based on past proven performance and selection to match existing plant equipment to minimize spare part inventory is a common consideration.

Air-operated diaphragm-type pumps are available in relatively small capacities, and these can be particularly effective in hazardous area classifications where use of electric motors may be undesirable.

SPECIFICATION REQUIREMENTS Specification requirements that affect the pump selection need to be recognized early in the selection process. For example, conformance to an API specification for a process pump application might invoke API 610 requirements for centrifugal process pumps or API 676 for rotary pumps. These specifications have implied

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requirements that will limit the range of manufacturers and product lines suitable for the service. Other examples include suction specific speed limits and minimum head rise to shutoff limits, as shown in Figure 3.

Product Line Selection Now that the conditions of service are defined, one must select those product lines that satisfy the conditions of service. The suitability of a pump product lines is based on many factors, including basic product design, reliability, previous installations under similar service, and of course, cost. A brief outline of different pump types and their relative design features and characteristics are described below.

PUMP TYPES There are several different types of basic pump designs. Each design can be used for a range of flow and head combinations. Refer to Chap. 1 for a general overview of the type of pump that can be used for various heads and flow rates.

In addition to capacity and head considerations, other operating characteristics of a pump will help the engineer to select a pump type for a given application. These are typically based on maximum temperature, viscosity, and liquid handling capability, available materials of construction, solids handing, and general pump design. These various types of pumps are shown in Figure 4, each with their respective design advantages.

The two main branches in pump types are kinetic pumps and positive displacement pumps. Most applications engineers consider pumps from the either the centrifugal or positive displacement pump branches. Centrifugal pumps, like all dynamic pumps, add energy continuously to the fluid to increase the fluid velocities within the pump such that a velocity decrease near the pump discharge results in an increase in fluid pressure. Alternatively, positive displacement pumps periodically add energy to the liquid by direct application of force to moveable enclosed fluid volumes, resulting in an increase in pressure up to the point where the fluid passes through a port or valve into the discharge line. An important feature of a centrifugal pump is that for a given operating speed, the discharge flow will vary from the rated flow condition to zero flow, depending on the discharge pressure of the pump. Conversely, a positive displacement pump delivers nearly the same discharge flow rate at a given operating speed, irrespective of the discharge pressure of the pump.

In general, centrifugal pumps are favored by specifying engineers when both types of pumps are suitable for the conditions of service. This is due to the belief that centrifugal pumps are more reliable than a corresponding positive displacement pump since they generally have fewer moving parts, no check valves, and lower pressure pulsations than the positive displacement pump. There are certain applications that are certainly more suitable for positive displacement pumps, such as applications with:

- Low flow and high pressure
- High viscosity
- Low shear
- Self-priming
- Accurate flow rate (flow metering)
- Multiphase flow
- Solids handling
- High efficiency

A more detailed description, including design and application details of kinetic and positive displacement pumps are described in detail in Chap. 3. Some useful considerations in pump-type selection are detailed in the following sections.

VARIABLE HEAD/FLOW REQUIREMENT Centrifugal and axial flow pumps are able to operate in variable head/flow conditions. By reviewing the performance curve for a given pump, the head/flow range capability for these types of pumps can easily be determined. Refer to the centrifugal pump performance curve in Figure 5. For a specific impeller size, a centrifugal pump will produce any flow rate within its head-flow rate characteristic curve

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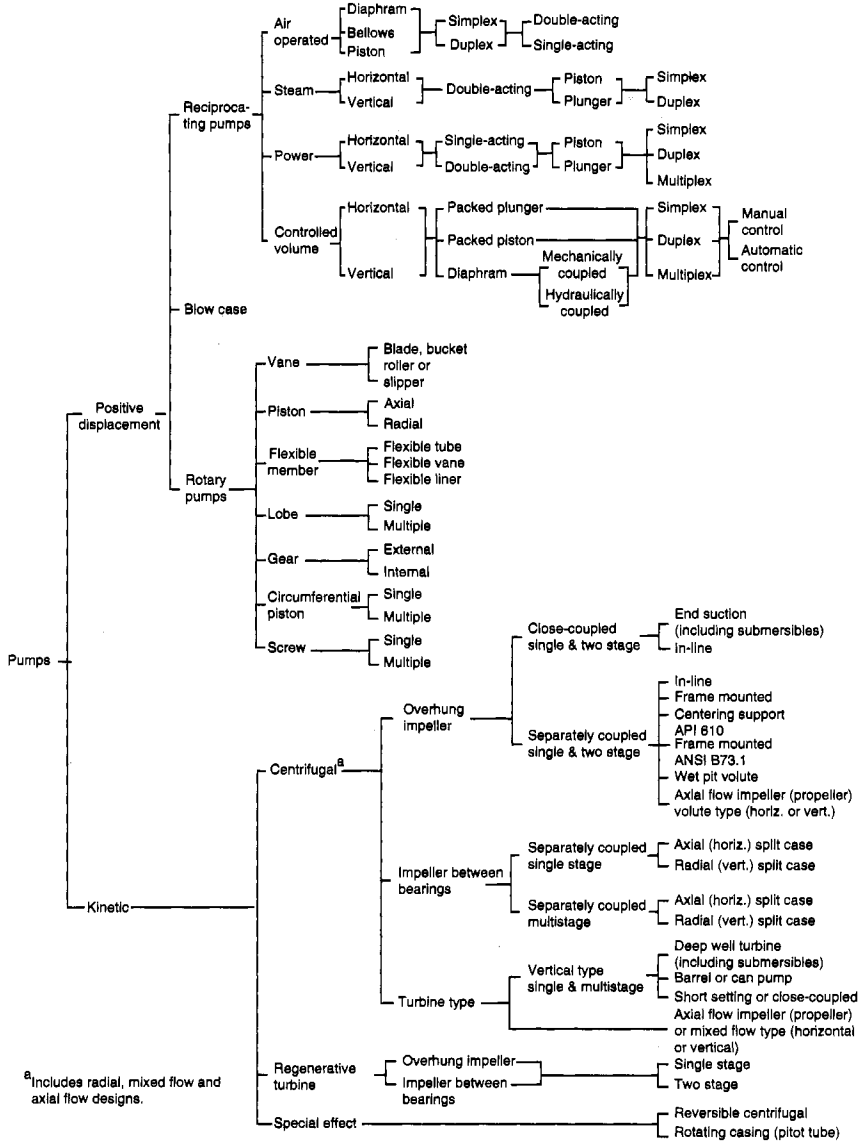


FIGURE 4 Types of pumps (Courtesy Hydraulic Institute)

which corresponds to intersection with the system head curve (see Figure 5), if sufficient NPSH is available. The system head characteristics can be changed to vary the flow by discharge throttling or by varying pump speed.

CONSTANT FLOW REQUIREMENT Positive displacement pumps deliver a nearly constant flow rate as a function of the operating speed. A slight decrease in flow rate accompanies

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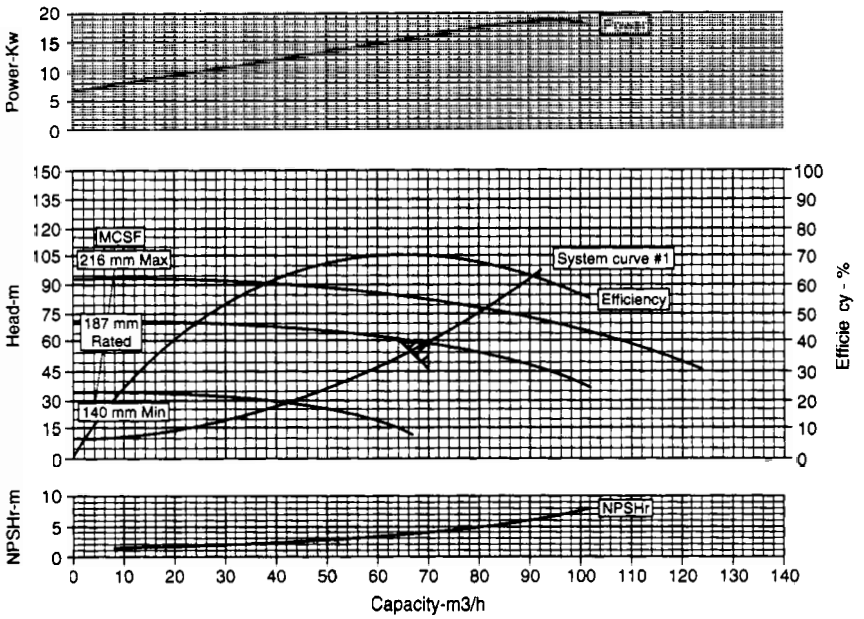


FIGURE 5 Centrifugal pump performance characteristic (Courtesy Intelliquip, LLC)

a larger differential pressure, due to internal leakage or "slip" within the pumping cavities (see Figure 6). Since a positive displacement pump will attempt to deliver a constant flow, irrespective of the discharge pressure level, a relief valve is usually installed to prevent over pressurizing the discharge piping system.

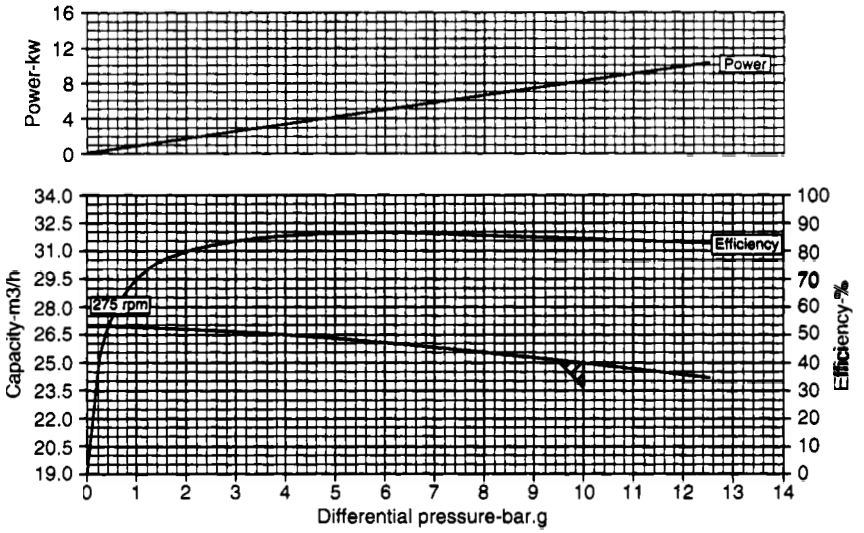


FIGURE 6 Positive displacement pump performance characteristic (Courtesy Intelliquip, LLC)

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Notice that the conventional format for centrifugal pump performance in Figure 5 and the positive displacement pump performance in Figure 6 inverts the abscissa and ordinate curve axis. If a positive displacement pump flow versus pressure characteristic is plotted on a centrifugal pump curve axis (as in Figure 5), the line curve would appear nearly vertical.

HIGH HEAD REQUIRED (ABOVE SINGLE STAGE CENTRIFUGAL PUMP ABILITY) Depending on the required flow rate, either a centrifugal, regenerative turbine pump, or a piston pump may fulfill the need for high differential head. If a relatively small flow is required, either an integrally geared high-speed centrifugal pump, regenerative turbine pump, or a piston pump may be applied. When selecting between these choices, other questions to ask might be:

- Will the pulsating flow of a piston pump be detrimental to system operation? Will a pulsation dampener take care of this problem?
- Is the liquid clean enough to avoid premature wear on pistons and cylinders?

For medium to high flow and high head combinations, a multistage centrifugal pump can be used. These are commonly available in horizontal, between bearing design in either a split case or radially split configuration. Alternatively, vertical multistage pumps, often known as turbine-type pumps are also used. Various designs of multistage centrifugal pumps are available with a wide range of prices reflecting special designs for a whole range of applications (high temperature, cryogenic, water, hydrocarbon, and so on).

LOW FLOW WITH PRECISE FLOW ADJUSTMENT ABILITY For low-flow applications where accurate flow metering is necessary, a proportioning or metering pump is appropriate. This type of pump can also be provided with variable flow capability. Certain types of gear, plunger, and diaphragm pumps can also be used in combination with a variable speed drive for flow rate regulation.

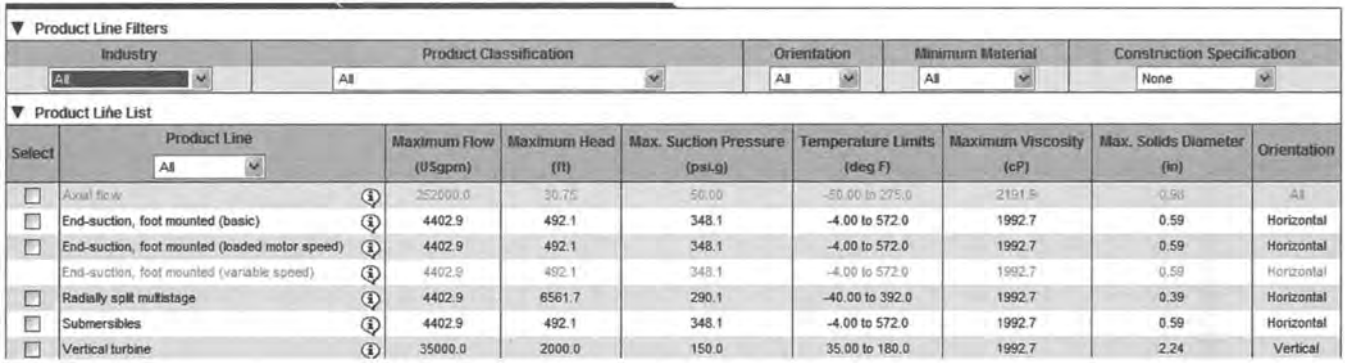
SELF PRIMING REQUIREMENT If a pump is taking suction from a source below the pump suction nozzle, a self-priming capability may be necessary. Positive displacement pumps such as a piston pump, rotary screw, or gear pump are able to self-prime within limits in the smaller capacity range. There are also special centrifugal pump designs that will self-prime in this situation.

LOW AVAILABLE *NET POSITIVE SUCTION HEAD* If the NPSHA is low, specially designed centrifugal pumps can be considered. Depending upon how low the NPSHA is, either a horizontal end suction with a suction inducer or a horizontal double suction arrangement may be applied. A vertical turbine pump may also be used, either immersed in the process fluid, possibly in a tank or vessel, or in a specially designed vessel (known as a suction can) that can be installed below grade to increase the NPSHA.

Evaluating Alternative Pump Selections Once the COS are established, a list of candidate pump product lines are identified (see Figure 7). The candidate product lines are evaluated or "filtered" based on technical criteria such as flow, head, suction pressure, temperature, viscosity, and solids handling capability. Further, other criteria such as material compatibility to the process fluid, pump-type classification, industry specification, pump orientation (vertical vs. horizontal), and industry application experiences also filter the choices to the "best" candidates.

Next, each pump model within the candidate product line families are evaluated against the conditions of service. These pump models are sized to suite the COS. For centrifugal pumps, this includes defining the operating speed, hydraulic design, impeller trim, number of stages, driver size, and materials of construction (see Figure 8). For positive displacement (PD) pumps, the hydraulic design may include size and number of plungers, gear size, or screw diameter and pitch. Equally important, the operating speed of a PD pump generally requires a speed reducer between the pump and the driver. Examples include gearbox drive, V-belt drives, or more popularly today, variable frequency drives (VFD).

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▼ Product Line Filters

Industry	Product Classification	Orientation	Minimum Material	Construction Specification
All	All	All	All	None

▼ Product Line List

Select	Product Line	Maximum Flow (USgpm)	Maximum Head (ft)	Max. Suction Pressure (psi.g)	Temperature Limits (deg F)	Maximum Viscosity (cP)	Max. Solids Diameter (in)	Orientation
☐	Axial flow	252000.0	30.75	50.00	-50.00 to 275.0	2191.5	0.98	All
☐	End-suction, foot mounted (basic)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	End-suction, foot mounted (loaded motor speed)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
	End-suction, foot mounted (variable speed)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	Radially split multistage	4402.9	6561.7	290.1	-40.00 to 392.0	1992.7	0.39	Horizontal
☐	Submersibles	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	Vertical turbine	35000.0	2000.0	150.0	35.00 to 180.0	1992.7	2.24	Vertical

FIGURE 7 Example of candidate pump product lines (Courtesy Intelliquip, LLC)

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Results List					Thumbnail Curves (All)					
Result	Pump Size	Display Thumb-nails	Stages	Speed Criteria	Pump speed, rated (rpm)	Pump Efficiency (%)	NPSHr (m)	Max. Power (kW)	% BEP	% Max Diameter or Pitch
✓	3X8 MULTI	☐	4	Sync.	1,775	73.77	1.19	21.37	80.51	98.53
✓	3X8 MULTI	☐	11	Sync.	1,185	72.27	1.03	17.85	120.42	98.53
✓	1.5X6 MULTI	☐	2	Sync.	3,550	71.17	2.79	20.57	105.27	100.00
✓	3X10 MULTI	☐	6	Sync.	1,185	70.70	0.66	17.76	107.09	98.26
?	2X6 MULTI	☐	9	Sync.	1,775	68.43	2.26	18.61	121.55	100.00
✓	2X8 MULTI	☐	5	Sync.	1,775	68.37	1.51	18.17	129.79	97.79
✓	2X6 MULTI	☐	10	Sync.	1,775	66.94	2.29	18.02	126.89	97.00
✓	80x50x200 IEQ	☐	1	Sync.	3,500	66.26	2.69	27.76	65.82	97.79

FIGURE 8 *Example of pump search results list (Courtesy Intelliquip, LLC)*

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Choices among different models are generally based first on technical requirements; namely, does the pump model satisfy all of the conditions of service? (Note: In Figure 8 this is designated by the "check mark" in the left most column.) Then, among technically compliant candidate pump models, the "best" pump is chosen based on criteria established by the purchaser. Typically, this is balance between cost (as depicted by the "Price Index" in Figure 8), pump efficiency, and pump reliability. Pump reliability and operating performance have been correlated to many factors, including the "fit" within the operating range with particular emphasis on how closely the rated flow corresponds to the best efficiency flow rate for the rated impeller diameter. These features are easily visualized using a family curve presentation as shown in Figure 9. Here, the rated condition is depicted graphically between the operating boundaries of the minimum and maximum impeller trim, and the minimum flow and runout flow.

LCC considerations of pumps and their included pumping systems are becoming more important. These will be discussed in a later section.

Upon selecting the "best" pump model, final pump performance calculations are generated, as shown in Figure 10. These calculations are necessary for the pump system engineer to use in making detailed comparisons between alternative pump offerings and in sizing the final piping and foundation design.

Pump Configuration In the previous section, the pump selection process was described, resulting in a pump model choice with supporting technical and pump performance data. The subsequent pump configuration process involves choosing among an array of option choices to complete the pump scope-of-supply. While the design, performance, and operating characteristics of the pump are substantially fixed by the pump selection, the pump configuration has a substantial influence on installation costs, maintenance, reliability, lead time, and quality of the pump package.

CONFIGURATION WORKUP The manufacturer's application engineer begins by preparing the configuration workup. This configuration workup is a methodical process of:

1. Interpreting the customer's pump specification,
2. Comparing the specification to the product standards available for the manufacturers product line
3. Making scope-of-supply choices that simultaneously adhere to the customer's specification and the product line standards, while providing an economical and quality technical solution

Seasoned applications engineers for pump manufacturers provide this expertise and perform this function. The configuration workup is accomplished by making option choices from different option categories, as described below.

MATERIALS OF CONSTRUCTION Most product lines have pre-engineered material choices that are offered. These range from standard materials specified by the American Petroleum Institute for various pump types (see API Standard 610, 674, 675, 676, 685) to nonmetallic choices offered for specialty services. Configuration choices often allow upgraded material choices for impeller, shafts, wearing rings, and so on to improve the service life and corrosion resistance of those components. Pump specifications often dictate the materials based on prior applications experience. Otherwise, the manufacturer is responsible to offer material recommendations or upgrades considering fluid compatibility, corrosion, applications experience, or operating experience. See Chap. 8 for more information on "Materials of Construction."

SHAFT SEAL AND SEAL FLUSH This choice of shaft seals is sometimes straightforward for well-known pump applications using off-the-shelf mechanical shaft seals, pre-engineered for use with each pump. Alternatively, some demanding process applications involving flammable and hazardous fluids at extreme temperature conditions (either cryogenic or high temperature) require very careful seal selections that can be more costly than the

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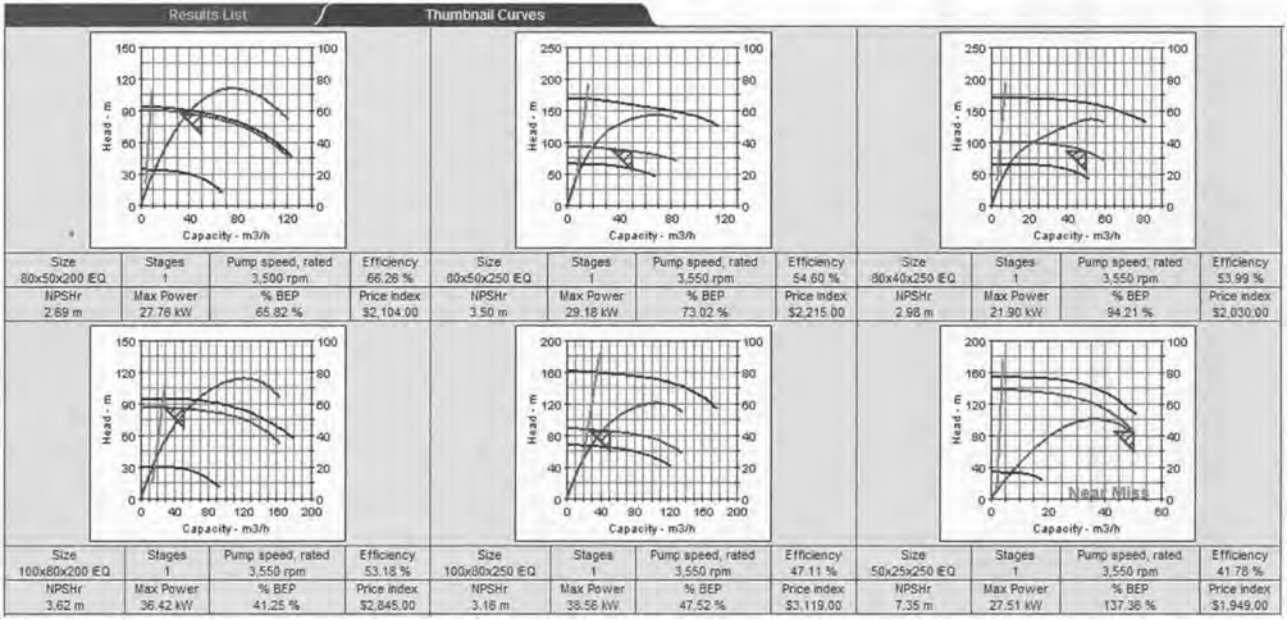


FIGURE 9 Collection of candidate pump family curves (Courtesy Intelliquip, LLC)

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Pump Performance Datasheet

Customer	-	Quote number	SP9_2
Customer reference	-	Pump size	80x50x200 IEQ
Item number	003	Stages	1
Service	-	Based on curve number	200IEQ-13 Rev 0
Quantity of pumps	1	Date last saved	07 Apr 2007

Operating Conditions		Liquid	
Flow, rated	50.0 m3/h	Liquid type	Water
Head, rated (requested)	85.0 m	Additional liquid description	-
Head, rated (actual)	84.5 m	Solids diameter, max	0.00 mm
Suction pressure, rated / max	1.0 / 1.5 bar.g	Temperature, max	20.00 deg C
NPSH available, rated	8.0 m	Fluid density, rated / max	0.996 / 0.996 kg/dm3
Frequency	60 Hz	Viscosity, rated	1.04 cSt
Performance		-	-
Pump speed, rated	3,500 rpm	**Material**	
Impeller diameter, rated	211 mm	Material requested	Grey cast iron
Impeller diameter, maximum	216 mm	Material selected	Grey cast iron
Impeller diameter, minimum	140 mm	**Pressure Data**	
Efficiency	66.3 %	Maximum working pressure	10.3 bar.g
NPSH required / margin required	2.7 / 0.0 m	Maximum allowable working pressure	19.7 bar.g
Specific speed / Suction specific speed	994 / 9,565 US units	Maximum allowable suction pressure	19.7 bar.g
MCSF	7.7 m3/h	Hydrostatic test pressure	31.0 bar.g
Head, maximum, rated diameter	90.2 m	**Driver & Power Data**	
Head rise to shutoff	5.9 %	Driver sizing specification	Rated power
Flow, best eff. point (BEP)	76.0 m3/h	Margin over specification	0.0 %
Flow ratio (rated / BEP)	65.8 %	Service factor	1.00
Diameter ratio (rated / max)	97.8 %	Power, hydraulic	11.5 kW
Head ratio (rated dia / max dia)	96.8 %	Power, rated	17.4 kW
Viscous coefficients (CQ / CH / CE)	1.00 / 1.00 / 1.00	Power, maximum, rated diameter	27.8 kW
Selection status	Acceptable	Minimum recommended motor rating	18.6 kW / 25.0 hp

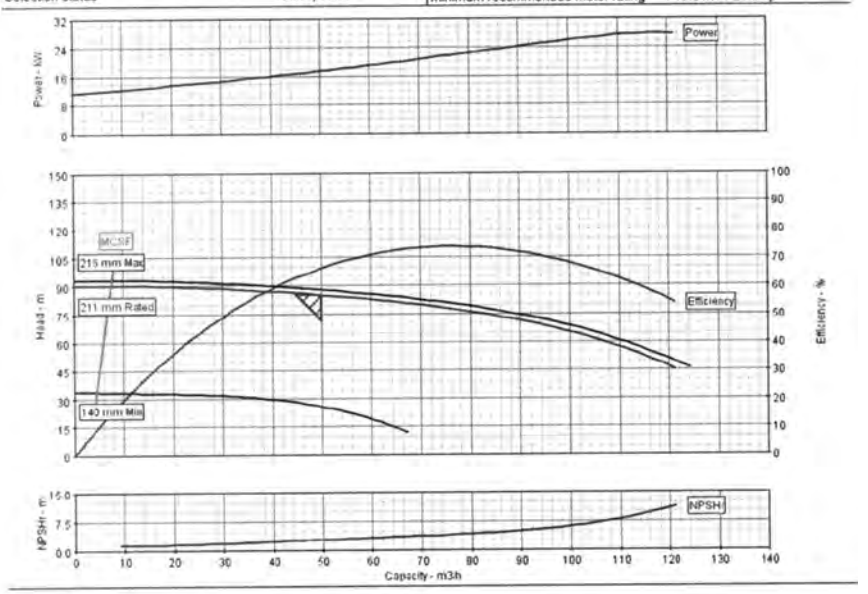


FIGURE 10 Pump performance data sheet (Courtesy Intelliquip, LLC)

pump itself. Mechanical seal vendors are generally consulted by the manufacturer to assure proper selection and configuration of the seal choice and corresponding seal flushing plans. See Chap. 5, or the API 682 standard for shaft sealing systems for centrifugal and rotary pumps is a useful industry reference.

DRIVER AND DRIVER MOUNTING The driver choice is typically outlined in the purchasers pump specification. The vast majority of drivers are synchronous speed electric motors. Variable speed drivers offer greater operational flexibility to vary output flow and

Previous Page

performance. These include electric motors with variable frequency drives, engine drives, or steam turbines.

For both technical and commercial reasons, the purchaser may decide to purchase the pump/driver combination in various ways.

The pump and driver may be purchased separately. This may be advantageous if either the pump or driver (but not both) can be purchased locally. This will save shipping costs and possibly allow the purchaser to meet client requirements for locally manufactured content for a project. Separate purchase of drives on large capital projects can also lead to quantity price discounts and limited spare part inventories.

The purchaser must consider the risk associated with the separate purchase of the pump and driver. Equipment installation and alignment problems (with resulting start-up delay) are more probable than when the pump vendor takes single source responsibility for the purchase and skid mounting of all components.

If the decision is made to purchase the driver separately from the pump, the pump manufacturer can provide the equipment for block mounting or provide a skid on which the driver can be installed in the field. Either way, additional shop inspection is recommended to verify dimensions for field interface. It should be noted that pump manufacturers may purchase a high volume of electric motors and obtain greater discounts than most operating companies and engineering contractors.

See Chap. 9 for more information on "Pump Drivers."

QUALITY AND TESTING Depending on the pump service and technical risk associate with the equipment purchase, the pump specification may require special pump testing or quality requirements. Typical testing options are performance tests (hydrotest, pump performance, or *NSPHR*) that may provision for a customer to be present on-site to witness the test. See Chap. 16 for more information on "Pump Testing." Similarly, quality requirements often require material certifications or nondestructive examination of critical areas. The API Mechanical Equipment Standards such as API 610 and the Process Industry Practices (PIP) offer recommendations for pump quality requirements.

COMMERCIAL PRICE MAKEUP The configuration workup is used to prepare a commercial price makeup, as depicted in Figure 11. Various option categories and configuration option choices are depicted in this figure, as described in the previous section. In addition, the commercial price implications are also described. All price makeups involve engineering trade-offs between technically sufficient offers versus more robust but costly offerings. This is dictated by the pump specification and the skill of the applications engineer.

GENERAL ARRANGEMENT DRAWINGS General arrangement drawings are typically required to be provided by the manufacturer within the quotation submittal package. A sample drawing is found in Figure 12. The general arrangement drawing provides the overall outline dimensions of the pump package, as well as any other connections or interfaces that the purchaser/engineer must consider upon pump installation. These interfaces include the driver and electrical requirements, seal flush and cooling water piping connections. Weights of the pump, baseplate, and motor are also used in finalizing the foundation design.

For complex installations with specially procured drivers, seal flush plans, or baseplates, certified general arrangement drawings are often provided after the order is confirmed since design details are not yet finalized.

CONSTRUCTION DATA SHEET A construction data sheet is also provided as part of the quotation submittal package. Most manufacturers provide a proprietary data sheet automatically created from their quote generation system. An example is shown in Figure 13. In lieu of accepting the manufacturer's data sheet, many purchasers or specifiers require completing their own proprietary data sheet. Alternatively, an industry recognized data sheet like API or PIP is used. These alternative data sheet formats are laborious to fill out and to maintain through the life cycle of the quotation and procurement process. Improved methods using *electronic data exchange* are under development as described in a later section in this chapter.

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Description	Qty	Extended Sell Price
80x50x200 IEQ	1	$ 2,442.00
Manufacturing Details		
Booking location:: Houston		
Bareshaft pump supply: Houston		
Unit packaging: Houston		
Construction		
Material		
Grey cast iron		Included
4140 carbon steel, sleeved shaft		
Cast iron impeller material		
Carbon steel shaft sleeve		
Standard material casing & gland gaskets		
Cast iron support head		
Cast iron bearing housing		
Mechanical		
PN10 flat face flanges		
Double row thrust bearings - oil lube		
Sealing Options		
Crane Standard Materials Seal 1A	1	$ 152.77
Casing cover type 2 - No flush tap		
No casing cover bushing		
Flush gland for flat gasket	1	$ 357.00
Seal flush piping options		
No seal flush piping		

FIGURE 11 Configuration options and pricing (Courtesy Intelliquip, LLC)

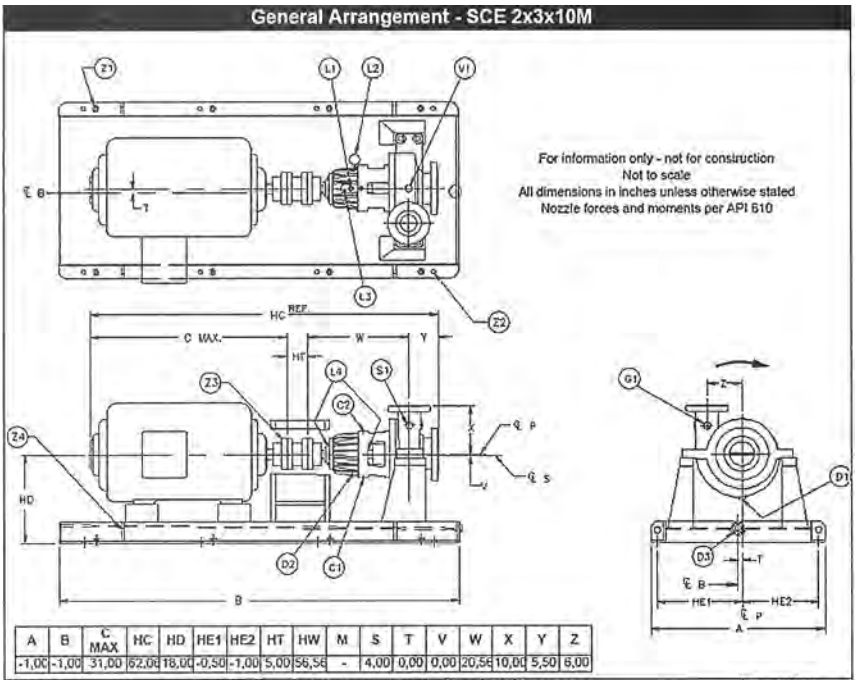
PUMP QUOTATION A final quotation submittal package is provided to the purchaser (the initiator of the RFQ). This contains documents such as a price makeup, performance curves, data sheets, and general arrangement drawings. Depending on specification requirements, additional documents such as a vendor drawing and data requirements, installation lists, or quality control plans may be required.

EVALUATING AND PURCHASING THE PUMPING EQUIPMENT

After the bid requisition has been prepared and sent to the approved pump vendors for quotation, purchasers begin preparing bid review documents. Both commercial and technical evaluations are necessary and can be accomplished on the same or separate documents.

Technical and Commercial Bid Evaluation Review strategies differ between projects and clients. Often, technical reviewers are not made aware of the quoted costs and are concerned only with technical specification compliance and scope of supply. In this case, the commercial evaluation is done by others. The final comparison of costs is made after the various vendors quotes are conditioned to meet all technical requirements. If the cost is withheld from the technical reviewer, an important piece of evaluating information is missing, which increases the chance of missing key requirements. Based on a quick

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A	B	C MAX	HC	HD	HE1	HE2	HT	HW	M	S	T	V	W	X	Y	Z
-1,00	-1,00	31,00	62,00	16,00	-0,50	-1,00	5,00	56,56	-	4,00	0,00	0,00	20,56	10,00	5,50	6,00

Nozzles	Suction	Discharge
Size	3 inch	2 inch
Rating	ANSI 300	ANSI 300
Face	RF	RF
Position	End	Top

Driver	
Manufacturer	RuhrPumpen choice
Type (enclosure)	TEFC
Rating / Frame	40,00 hp / 324TS
Speed / Poles	3600 / 2
Volts / Phase / Hz	230-460 / 3 / 60

Piping	
Seal plan - primary/auxiliary	None / -
Cooling water plan	None
-	-

Mechanical Seal	
Manufacturer	John Crane
Type	1648
-	-

Coupling	
Manufacturer	Metastream
Type & size	TSCS 0013
Guard	Non-spark

Weights (Approx.)	
Pump	600,0 lb
Baseplate	Special baseplate - consult factory
Driver	362,0 lb

Item	Size	Qty	Description	Connection
C1	.50 NPT	1	Cooling water inlet, bearing housing	Plugged
C2	.50 NPT	1	Cooling water outlet, bearing housing	Plugged
D1	.75 NPT	1	Drain, pump case	CS, SW, Flgd.
D2	.50 NPT	1	Drain, bearing lube oil	Plugged
D3	2.00 NPT	1	Drain, baseplate	Open
G1	N/A	N/A	Gage connection (N/A)	N/A
L1	.75 NPT	1	Fill / vent, bearing lube oil	Plugged
L2	.25 NPT	1	Constant level oiler	By RuhRPumpen
L3	.25 NPT	1	Oil mist inlet	Plugged
L4	.25 NPT	1	Oil mist outlet	Plugged
S1	N/A	N/A	Seal recirculation connection (N/A)	N/A
V1	N/A	N/A	Vent, pump case	N/A
Z1	.75 inch	6	UNC heavy hex nut for vertical jacking screw	
Z2	1.0 inch	6	Holes for 0.75 inch anchor bolt	
Z3	-	-	Coupling	
Z4	-	4	Lifting lug	
CLB	-	-	Centerline of baseplate	
CLP	-	-	Centerline of pump	
CLS	-	-	Centerline of suction	

Customer	
Customer reference	
Project name	-
Service	-
Item number	001
Pump size	SCE 2x3x10M
Quotation number	30939

FIGURE 12 General arrangement drawing (Courtesy Intelliquip, LLC and Ruhrpumpen)

comparison of cost, it is easy for the document reviewer to establish the quoted general scope for each vendor. If large cost differences are noted between vendors, the reviewer must carefully determine what major differences in quoted scope likely exist.

The number of quotes that will be fully evaluated will depend partly on the number received. The goal is to ensure that the least expensive, technically acceptable bidder can be determined while also minimizing evaluation effort.

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Centrifugal Pump Data Sheet

Customer:	Abastible	Sulzer Tender ID:	001	
Project Name:	-	Inquiry Date:	February 16, 2006	
Inquiry Number/ID:		Bid Submitted Date:	-	
Customer Item Number:	004	Revision Date:	May 9, 2006	
Application:	-	Quantity:	1	
Operating Conditions		**Pump Design Data**		
Liquid Type:	Water	Pump Type:	No Data	
Temperature, Max:	20.00 deg C	Product Line:	AhlstarUp	
Fluid Density, Rated/Max:	0.998 kg/dm3	Pump Size / No. of Stages:	A 11-50_O / 1	
Vapor Pressure, Rated:	0.02 bar.a	Rotation (viewed from drive end):	No Data	
Viscosity, Rated:	1.00 cSt	Suction Impeller Type:	Open	
Consistency:	-	Casing Mounting:	-	
Air Content:	-	Casing Split:	-	
Discharge Flow, Rated:	68.46 m3/h	Casing Type:	-	
Discharge Head, Rated/Actual:	50.84 m / 50.96 m	*Nozzle* / *Size* / *Rating* / *Face* / *Pos'n*		
Suction Pressure, rated/max:	0.00 bar.g / 0.00 bar.g	Suction: - / - / - / -		
NPSH Available:	Ample	Discharge: DN50 / PN16 / - / -		
Performance		Lineshaft Diameter:	-	
Performance Curve No.(s):	K18709	Bearing Type, Radial:	-	
Pump Speed:	2,980 rpm	Lineshaft Bearing / Bowl Bearing (Vert. pumps only):	-	
Frequency:	50 Hz	Bearing Type, Thrust:	-	
Fixed/Variable Speed:	-	Bearing Lubrication:	-	
Impeller Diameter, Rated:	190 mm	Baseplate Type:	PMS 3	
Impeller Diameter, Maximum:	210 mm	**Materials**		
Impeller Diameter, Minimum:	150 mm	API Material Class:	-	
Efficiency:	70.44 %	Barrel/Can:	-	
NPSH Required (3%) / Margin Required:	2.26 m / 0.00 m	Case / Bowls:	-	
Ns / Nss:	1,177 US units / 11,175 US units	Column:	-	
Head, Maximum, Rated Diameter:	54.12 m	Discharge Head:	-	
Head Rise to Shutoff:	6.44 %	Impeller:	-	
Flow, Best Eff. Point (BEP):	86.58 m3/h	Case/Impeller Wear Rings:	-	
Diameter Ratio (Rated / Max):	90.48 %	Shaft:	-	
Head Ratio (Rated Dia. / Max Dia.):	77.89 %	Diffusers:	-	
Viscous Coefficients (CQ / CH / CE):	1.00 / 1.00 / 1.00	**Shaft Sealing, Flush & Cooling Piping Plans**		
Pressure Data		Seal Size / Type:	DS11 / -	
Maximum Working Pressure:	5.30 bar.g	Seal Code:	-	
Working Pressure Limit:	16.00 bar.g	Seal Manufacturer:	Sulzer	
Suction Pressure Limit:	0.00 bar.g	Seal Flush Piping, Primary:	-	
Hydrostatic Test Pressure (Suction/Discharge):	0.00 bar.g	Seal Flush Piping, Secondary:	-	
Suction Pressure, rated/max:	0.00 bar.g / 0.00 bar.g	Cooling Water Piping:	-	
Discharge Pressure, Rated:	4.98 bar.g	**Driver & Power Data**		
Differential Pressure, Rated:	4.98 bar.g	Driver Size:	18.00 kW	
Equipment Weights (Approximate)		Volts/Phase/Frequency:	- / 2 / 50 Hz	
Pump:	66.00 kg	Service Factor:	n.a.	
Driver:	0.00 kg	Power, Rated:	13.42 kW	
Baseplate:	140.0 kg	Power, Maximum, Rated Diameter:	17.80 kW	
Total Package:	206.0 kg	Enclosure:	-	
Accessories				
Driver:	-			
Coupling:	-			

FIGURE 13 Construction data sheet (Courtesy Intelliquip, LLC and Sulzer)

SHORT LISTING OF QUOTES When bids are first received, a quick commercial and technical tabulation is often prepared that lists the main scope of supply, extent of specification compliance, major deviations, and the associated total cost. This is done to get a general idea of the completeness of the quotes as well as to provide information to determine which of the quotes will be fully evaluated.

TECHNICAL BID EVALUATION Quotations must be evaluated against the specified requirements. A rigorous feature-by-feature comparison between alternative bids is often difficult,

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since a centrifugal pump may be compared against a positive displacement pump. In addition, each manufacturer's submittal requires a significant amount of effort and ingenuity in providing a cost-effective yet technically suitable pumping solution. Professional ethics must be respected to maintain the intellectual effort provided by each manufacturer.

After the decision is made on which bids to fully evaluate, the formal review and conditioning process can begin. A technical bid tabulation form is necessary to ensure that the equipment is provided in compliance with the requisition. This tabulation will list the important points of each vendor quote in table form. Each quote is then compared in a technical bid tab that includes:

- Equipment model number and size
- Compliance with process duty requirements (flow, head, NPSHR, and so on)
- Mechanical design limitations (pressure and temperature)
- Mechanical and hydraulic operating parameters (such as absorbed power, head rise to shutoff, capacity rise to run out, percentage of best efficiency point)
- Mechanical design features (such as materials of construction, seal type, seal flush type, bearing and lubrication type)

When the technical bid evaluation form is being completed, the reviewer must highlight all items that do not meet requisition requirements or that are not clearly defined in the vendor's quote. In addition, features of the vendor's quotation should be highlighted to clearly indicate advantages, disadvantages, ambiguities, and noncompliances. A typical API-610 pump technical bid evaluation is shown in Figure 14.

TECHNICAL QUOTE CLARIFICATION When the initial review is complete, a list of clarification questions needs to be prepared and sent to each vendor being evaluated. Questions should be direct and concise to ensure that responses answer the questions asked. The vendors must be encouraged to provide cost adders, if necessary, to meet requisition requirements. It is usually appropriate to allow the vendors 1 week to reply.

When answers to the clarification questions are received, the bid tabulation must be updated to include any revisions in the vendors quotation. Care must be exercised in tracking revisions to the original quotation. If additional clarification is needed, there are various ways to go about it, depending on the situation. Some suggestions are detailed in the following sections.

TECHNICAL PURCHASE RECOMMENDATION When all issues of bid requisition noncompliance have been satisfactorily addressed, the engineer must recommend which vendor or vendors may be considered for purchase of the required pump. Usually, part of this recommendation is the completed and signed technical bid tabulation. The final bid tabulation may show that all (or none) of the vendors are in complete compliance with all requisition requirements. The bid tab should clearly show where all noncompliance occurs. On the tabulation, a statement of technical compliance or noncompliance for each quote that has been reviewed must be provided. Notes that clarify any ambiguous line item on the tabulation are highly recommended so reviewers will have as few questions as possible on the bid tabulation content.

The bid tab should be attached to a recommendation memo or letter that states the recommendations and provides adequate justification. If a vendor is not in full compliance with the bid requisition, this should be noted, along with other reasons that support the recommendation. All revisions made since the original submission of the quotations that are necessary for purchase must be addressed. This might take the form of a scope of supply listing for each recommended vendor that clearly defines what must be purchased to meet the requisition requirements.

If a vendor is clearly favored over others for technical reasons, this must be clearly stated. Because the purchase order is usually given to the lowest cost technically acceptable bidder, this type of preference must be compelling.

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Bid Tab Comparison
Centrifugal Pump : P1252

Operating Conditions	Specified	Pump Mfg # 1	Pump Mfg # 2	Pump Mfg # 3
Flow, Rated (gal/m)	6,417	6,417	6,417	6,417
Discharge Pressure (psig)	155	155.1	155.1	155.1
Differential Pressure (psi)	101	101	101	101
Differential Head (ft)	323	323	323	323
NPSHA/NPSHR (ft)	85	18.7	18.7	18.7
Manufacturer		Pump Mfg # 1	Pump Mfg # 2	Pump Mfg # 3
Model Number		4 × 6 – 11	4 × 6 – 11	4 × 6 – 11
Material				
Barrel/Case		Carbon steel	Carbon steel	Carbon steel
Impeller		12% Chrome	12% Chrome	12% Chrome
Case/Impeller Wear Rings		12% Chrome / 12% Chrome	12% Chrome / 12% Chrome	12% Chrome / 12% Chrome
Shaft		AISI 4140	410SS	410SS
Performance				
RPM		3560	3560	3560
Impeller Diameter, Rated (in)		9.3125	9.3125	9.3125
Impeller Diameter, Max (in)		11.125	11.125	11.125
Impeller Diameter, Min (in)		8.0	8.0	8.0
Rated Power (bhp)		66.1	66.1	66.1
Efficiency (%)		71.5	71.5	71.5
Min Continuous Flow, Thermal (gal/m)				
Min Continuous Flow, Stable (gal/m)		2,671	2,671	2,671
Max Head @ Rated Impeller (ft)		351	351	351
Max Power @ Rated Impeller (bhp)	85	85	85	85
NPSHR @ Rated Flow (ft)		18.7	18.7	18.7
Max Suction Specific Speed	10660	10660	10660	10660
Coupling				
Coupling Manufacturer	*Thomas*	*Thomas*	*Thomas*	*Thomas*
Coupling Model Number	71	71	71	71

FIGURE 14 Pump technical bid evaluation

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For vendors that have been listed as technically acceptable, recommended for purchase, a list that addresses vendor's exceptions to requisition requirements should be attached. These exceptions should be justified in the technical purchase recommendation.

COMMERCIAL PURCHASE RECOMMENDATION Prior to completion of the technical evaluation, certain commercial items can be reviewed and negotiated with the vendors being evaluated. Terms and conditions of payment, guarantee/warranty, and proposed delivery schedule can usually be discussed in parallel with the technical conditioning of the bids.

After the technical recommendation for purchase is complete, the commercial purchase recommendation may be completed. In addition to including the technical bid analysis, this recommendation should cover the following items (if applicable):

1. Listing of original vendor bid scope of supply and the associated cost(s)
2. Listing of revisions to the vendor's original quotes needed for requisition compliance with the associated cost(s)
3. Required spare parts cost
4. Shipping costs
5. Miscellaneous costs associated with documentation, inspection, and so on
6. Utility consumption cost evaluation
7. Agreed delivery schedule
8. Exceptions taken to commercial terms and conditions
9. Economic adjustments for different terms of payment

At this point, the purchase recommendation can be made, usually based on the lowest cost from a technically acceptable bidder.

Final Purchase Decision and Purchase Order Issue Now that the commercial purchase recommendation is complete, it is forwarded to the parties responsible for making the purchasing decision. After the selection is made, the engineer must decide if additional discussions with the vendor are needed prior to issue of the purchase order. For complex, costly equipment, and large orders, this extra discussion often takes place in a pre-award meeting.

During this meeting, the final agreed-upon equipment, technical, and commercial requirements are confirmed. A final review of the vendor's technical exceptions to the requisition can also be completed at this point. As with previous meetings, recording all discussions and agreements is very important. Both parties should read and sign the meeting notes.

After final discussions are complete, the last step in purchasing the pump is the issue of the purchase order. Because the paper trail for the purchasing cycle can be very complex, it is strongly suggested that the purchase order be a stand-alone document. It may or may not be sufficient to only reference previously transmitted narrative specifications. Many options are likely to have been discussed and agreed to during the process. All these agreements, even if properly documented during the process, should be incorporated into the purchase order. This includes revising data sheets as necessary and also incorporating agreed vendor exceptions to the inquiry. Establishing "clean order requirements" at the purchase order will avoid costly interpretation, manufacturing errors, or shipment delays later in the process.

Often it takes several days (or weeks) to incorporate all agreements to the various documents that are contained in the requisition. In order to permit the vendor to proceed in advance of the formal purchase order issue, a letter can be written by the purchaser that confirms the order and requests the vendor to begin work immediately.

SUMMARY

Selecting and purchasing pumps is a complex but routine process that involves rich engineering experience from both the purchaser and supplier of the pumping equipment. The

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multistep process outlined in the chapter, involving: (1) engineering the pumping system, (2) pump selection and configuration, and (3) evaluating and purchasing the pumping equipment routinely followed for each new pumping system.

Many firms, including owners, operators, engineering/procure/construct (EPC) contractors, pump manufacturers, and pump distributors/packagers have all developed competencies and internal procedures to streamline the selection and purchasing process. This engineering effort can represent a significant portion of the overall equipment cost and has seen increased attention in recent years toward improving the overall efficiency.

The methods outlined in the previous sections represent the recommended approach using modern and effective practices. In the next section, emerging trends that seek to further improve the effectiveness of the selection and purchase of pumping equipment will be explored.

EMERGING TRENDS IN SELECTING AND PURCHASING PUMPS

Selecting and purchasing pumps involves the same basic process steps today as used in the recent past. While the process steps are the same, an engineer who performed those tasks 50 years ago would find many differences in the pump technology, the way purchasers and suppliers now interact, and the process automation tools used today. Globalization has had a profound effect in the way business is conducted. Rapid economic growth is now occurring outside of industrial nations in North America, Western Europe, Japan, and Australia. Information technology is enabling some of that growth as borders and distance are diminished through better communication technology and logistics to move people and materiel. On the other hand, this growth is placing a growing burden on the world's energy and natural resources. All of these factors are having an affect on the pump industry. The following sections will explore emerging trends affecting "how" the selection and purchase of pumps are taking place, and how they will continue to change in the coming years.

Pumping Systems Perspective

ENERGY EFFICIENCY The emphasis on energy conservation is becoming increasingly relevant. The world's demand for energy continues to expand, particularly in rapidly growing economies. In some cases, energy demand is exceeding availability. Energy costs continue to rise as sources of nonrenewable energy are depleted. Further, greenhouse gases created by industry, power production, and transportation are also affecting the world's environment. A Department of Energy report in 1998 reported that electric motor systems in the United States consumed over 679 billion kWh of electricity per year. Interestingly, it was found that pumping systems represented about 25% of that total (DOE 1998) in applications such as HVAC (heating, ventilation, and air conditioning), municipal water and wastewater treatment, and chemical process and refinery applications, to name a few.

The purchase and operation of pumping systems is a significant expense for most facilities. As energy costs rise, these expenses will reflect an increased proportion of overall facility costs. Ironically, most pumping system procurement decisions are based primarily on the initial purchase price of the pumping equipment. This is particularly prevalent in the purchase of new equipment where major expenditures are made on new capital facilities with business driven on fixed price contracts. However, it is well known that the initial cost of a pump is typically a small portion of the total LCC to operate the pumping system. Energy, maintenance, and other operating costs generally outweigh these initial costs. Therefore, it makes economic and environmental sense to consider these total lifetime costs when designing and procuring these pumping systems.

The selection and purchasing practices in the pump industry can thus have a relevant effect on worldwide energy consumption. Pumping system, energy efficiency improvement opportunities fall into two types, the purchase of new systems or the modification of existing systems. New systems need to be examined from an LCC perspective. Existing systems should not be ignored and actually represent a greater opportunity since the number of existing systems far exceeds the number of new systems installed each year. Efficiency

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improvements in existing systems normally require changes to the pump, the control system, or the incident piping systems. In many cases, changes to these systems are not economically practical. However, the DOE estimates that the average energy savings available in a typical installed system is 20%, through better pump selection, control systems, or piping system design. Certain installations may even enjoy much greater energy savings opportunities of 25 to 50%.

A SYSTEMS APPROACH The importance of proper pumping system design is becoming more evident as energy efficiency and LCCs are more broadly evaluated in the purchasing decision. While this transition has been evolutionary rather than revolutionary, the effort to reorient the evaluation process in the pump industry is underway. At the macroeconomic level, energy costs and diminishing resources will motivate change. At the industry level, new initiatives are under development to educate and provide tools to help engineers to evaluate pumps from a broader systems perspective. The Hydraulic Institute (HI) has historically provided pump standards but is now extending its reach to include standards, guidelines, and educational programs on pumping systems. The HI and Europump, a similar organization of European pump manufacturers, collaborated to create two user guides on the subject; *Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems;* and *Variable Speed Pumping: A Guide to Successful Applications*. More recently, the HI has organized national pump systems educational initiative called *Pump Systems Matter* TM (HI 2004b). This initiative is directed toward shifting the focus to LCC metrics to realize improved energy efficiency and longer-term performance and cost improvements. This initiative is providing educational materials suitable for facilities and maintenance personnel, specifiers, purchasers, and manufacturers alike. The PSM initiative is an industry-driven and industry-organized national market transformation initiative that complements broader energy savings efforts sponsored by government organizations.

PUMP SYSTEMS OPTIMIZATION TOOLS Pump systems optimization has been a labor intensive process requiring expertise in the application and design of pumping systems. The educational component of the "systems approach" is important, but tools (i.e., computer software) to aid in the detailed calculations of LCCs and pump system analysis and optimization are also needed. These are discussed in the section on "Process Automation Technologies."

Selecting and Purchasing Pumps in the Information Age Many firms are applying emerging information technologies during the process of selection and purchasing pumps to improve their competitiveness. Computer-based applications are often used to aid in generating pump proposals or to check part inventory status. Design departments use CAD/CAM systems to shorten the design cycle and run simulations using structural finite element methods. However, the use of computer technology alone does not guarantee a measurable economic benefit.

Organizational and process changes are usually necessary to achieve the benefits of computer automation. These process improvements often extend beyond a single firm with the formation of formal and informal alliances between pump users, owners, engineering contractors, architect-engineers, pump suppliers, and other equipment suppliers. These trends place a greater emphasis on the purchaser–supplier interface during the pump selection process. From the purchaser's perspective, the optimal choice of pumping equipment has significant cost implications over the service life of the equipment. From the manufacturer's perspective, configuring the preferred offering of pumping equipment is crucial in securing a competitive advantage during the purchase evaluation.

The information age is redefining virtually all aspects of conducting business including the way pumping systems are designed, evaluated, procured, manufactured, and maintained through their entire life cycle. In this section, the process of selecting and purchasing pumps will be revisited in the context of the way critical information is communicated between pump purchaser and supplier. Then, four of the emerging information technologies that are contributing to improved quality and cycle time in the overall process will be described.

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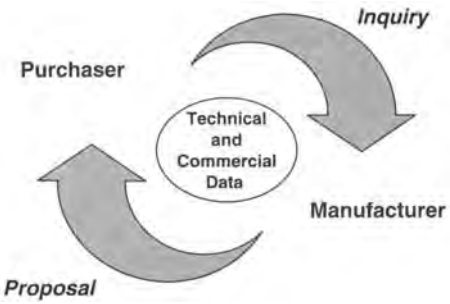


FIGURE 15 Purchaser–supplier information exchange

INFORMATION FLOW BETWEEN PURCHASER AND SUPPLIER Following the decision that pumping equipment is required, a selection and purchasing process is undertaken involving: (1) engineering the pumping system, (2) pump selection and configuration, and (3) evaluating and purchasing the pumping equipment. The entire process is information intensive, consisting of both technical and commercial information, as shown in Figure 15. The first two steps of the process (steps 1 and 2) are technical in nature, involving the exchange of system design, pump specifications, and performance and construction details of the pump. The last step of the process (steps 3) transitions toward the commercial elements of the purchasing decision such as equipment costs, LCC evaluations, terms and conditions, and delivery lead times. This flow of technical and commercial information is described by the inquiry-quotation information exchange.

This inquiry-quotation information exchange is not limited to only one purchaser-supplier interaction. With each new procurement opportunity, this information exchange effects every trading partner participating in the entire supply chain. Consider the simplified example (in Figure 16) of an operating company that gives three engineering/design contractors the opportunity to bid on an expansion of a chemical process plant. If each of these contractors issue three inquiries to pump manufacturers, nine inquiries are issued. Now, if each manufacturer issues three inquiries to their subsuppliers, a total of 27 inquiries are pending for this single plant expansion. In the end, only one contractor, one equipment-supplier, and one subsupplier actually receive orders to supply equipment for the project. Thus, only 3 inquiries out of 27 representing 11% of the total application and

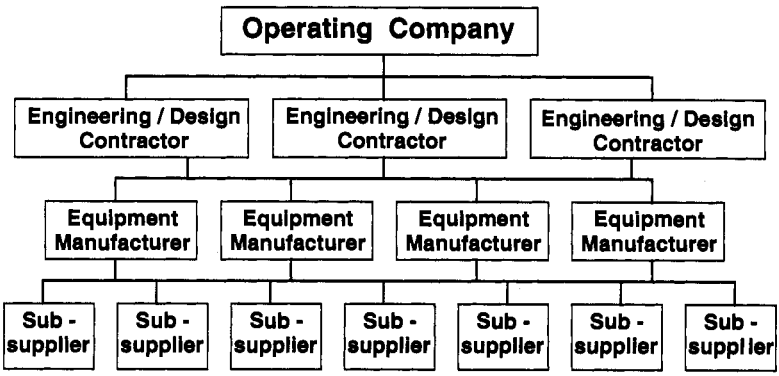


FIGURE 16 Multifirm information flow

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quotation effort represent "useful" work. These engineering costs are recovered only when equipment is actually purchased. The cost of the other 89% of effort by those participants in the inquiry/proposal process who did not receive a customer order are "wasted" and absorbed as a sales and engineering overhead. This example actually understates the total number of inquiries issued indicating that the "wasted" effort is probably much greater than calculated.

To reduce costs, purchasers are seeking ways to reduce the engineering resources required to process this information without compromising the quality of the competitive evaluation. Similarly, suppliers are developing methods for responding to inquiries (or requests-for-quotation [RFQ]) with less resources and effort while striving to still provide high quality quotations to their customers. Simplifying the inquiry-quotation process is an attractive proposition for both purchaser and manufacturer.

STRATEGIES FOR IMPROVING THE SELECTION AND PURCHASING PROCESS Both purchaser and manufacturer in the inquiry/quotation process are striving to enhance their competitiveness by reducing costs through process and information technology-oriented productivity improvements. Intracompany productivity gains have been the focus of many internal reengineering programs within a given firm. By their nature, these gains are limited to changes made within a single company and cannot effect improvements between multiple companies. However, the "ripple effect" on the total cost across multiple firms, described earlier, is a significant cost and represents a relatively unexplored opportunity for intercompany productivity gains. Some of these cooperative strategies engaged by both purchaser and supplier include the following:

- Form alliances to streamline the inquiry/quotation process and flow of information across multiple firms
- Reduce the number of participants in the process by defining a preferred set of customers and suppliers
- Move the pump selection task earlier in the pumping system design process by providing purchasers' access to computerized selection programs allowing the purchaser to optimize their process design
- Structure the flow of information between companies to reduce ambiguity and enhance common work processes

Existing and emerging information technologies offer a technology platform necessary to implement many of these strategies. The next section describes four system and information technologies that are particularly important in improving the selection and purchasing process for pumping equipment.

System and Information Technologies Information technologies are used throughout the design, manufacture, and maintenance of pumping equipment and systems. With respect to the specific activity of selecting and purchasing pumps, there are four system technologies that are gaining broader popularity and are playing a more meaningful role, including: (1) pump system analysis software, (2) pump system and selection programs, (3) pump configuration and quotation systems, and (4) electronic data exchange.

PUMP SYSTEM ANALYSIS AND OPTIMIZATION Designing the piping network and sizing the components for a pumping system are performed very early in the overall pump selection and purchasing process. The piping system design involves numerous components that introduce friction losses in the system. These must be calculated in order to estimate the behavior of the system resistance curve needed to properly size the pumps needed in the system. Changes in process conditions (pressure, temperature, fluid properties, tank elevation, and so on) or multiple/variable branch systems introduce additional operating conditions in the system design that must be predicted. This design process, when done manually, is tedious and time consuming. However, handling design iterations and performing system optimization studies have become more practical with the use of computerized design and analysis programs.

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Use of design and analysis programs became more prevalent in the mid-1980s with the advent of personal computers as powerful engineering workstations on the desktops of pumping equipment designers. These computers, with their easier to use software interfaces, encouraged the development of more robust and capable software applications that were more economical to deploy. Consequently, a substantial amount of effort went into the development of engineering programs dedicated to the sizing of piping systems, pumps, and other components.

Numerous piping system design/analysis computer applications are now available to aid in the design or analysis of piping systems and their components. Most of these programs are based on simultaneous path solutions. The more complete programs easily model piping networks and are capable of calculating friction losses through pipes and fittings. They also contain large libraries of standard pumped liquids and their fluid properties such as density, viscosity, and vapor pressure as a function of temperature. These programs are more time efficient and give more predictable results than former manual methods. An example is shown in Figure 17.

PUMP SELECTION SYSTEMS Most pump selection processes are now performed using computer-aided selection programs. Pump manufacturers performance curves are digitally programmed and accessed by the program to establish preliminary pump performance and design specifications. These programs can be reasonably sophisticated, using specialized mathematical algorithms to predict pump performance under varying operating conditions of speed, temperature, *NFSHA*, pressure, viscosity, and so on. Some are even capable of adjusting performance based on alternative mechanical seal design,

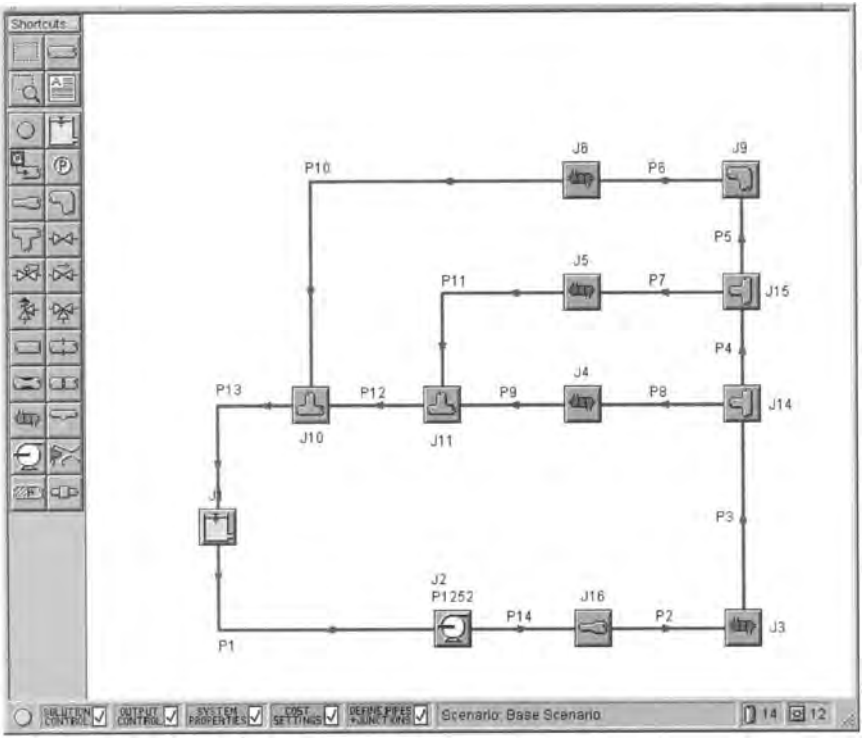


FIGURE 17 Pump system analysis and design schematic (Courtesy Applied Flow Technology)

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wearing ring design and clearance, bearing design, materials of construction, or other mechanical design features. Most pump manufacturers now offer some form of pump selection programs. In some cases, these pump selection programs are specially developed by the pump manufacturer using proprietary selection and searching methods. These programs are available under a licensing agreement directly from the pump manufacturer and are usually the same programs used by their internal applications engineers. Third-party programs are now available incorporating product lines from multiple pump manufacturers or customized to meet the specific needs of each manufacturer. The newest selection programs are designed to work directly from a browser on the Internet, providing immediate access to end users, specifiers, and applications engineers. These are convenient since special software is not required, data are easily shared between purchaser and supplier, and concerns over software updates become irrelevant.

Pump system and selection programs have improved the ability to evaluate large numbers of alternative design alternatives in a short period of time. Accurate performance calculations, even with variable pump speeds, fluid properties (viscosity, temperature, pressure), and mechanical configurations are readily predicted. Design and calculation errors have diminished and the overall quality of the process has greatly improved (Dahl and Ochs, 1997)

LCC CALCULATIONS One of the valuable features of a modern pump selection program is the ability to efficiently calculate LCCs. LCCs contain many different cost elements, as described in the earlier section, *Life Cycle Cost Evaluation.* In the example shown in Figure 18, the LCC equation is organized into the following sections:

Load Profile and Energy Costs The energy cost (C_e) is readily calculated with knowledge of the pump performance characteristics and the different load profiles in which the pump is expected to operate during its lifetime. With the optimization features of the LCC module, a comparison between fixed speed operation and variable speed operation is conveniently calculated, including the recalculation of pump performance coincident with the defined system curve(s).

Additional One Time Costs at Year 0 These are the initial costs (C_{ic}) of the pump and the installation and commissioning costs (C_{in}).

Additional Annual Costs These include the annual operating costs (C_o), maintenance costs (C_m), down time costs (C_s), and environmental costs (C_{env}).

Additional One Time Costs at Year, N The decommissioning costs (C_d) at the end of the service life of the equipment (year "N") is the final component of the cost model.

The additional costs are more difficult to model and are generally obtained from the pump manufacturer's quotation, estimates from the construction engineer, or operating experience from the owner/operators maintenance logs (see Hodgson and Walters, 2002).

Finally, while not always considered, the time value of money may be considered for both interest and inflation rates to calculate a net present value for the LCC.

PUMP CONFIGURATION AND QUOTATION SYSTEMS Pump quotations are complex documents containing a myriad of technical and commercial information. These quotations typically include data sheets, performance curves, general arrangement drawings, and a price makeups including terms and conditions. In many cases, alternative selections, comments to the customer's specification, or other supplementary information are also provided. The diverse array of information, expertise, and resources needed to generate a customized proposal has prompted the need to systematize the configuration and quote development process using computer-aided tools.

Some manufacturers have responded by implementing computerized product configurators. Product configurators aid the applications engineer in developing a pump quotation according to a prescribed product configuration model. These configurators are based on design guidelines and configuration rules that limit configuration choices. Examples of design rules are maximum casing pressure, maximum pump torque for a given shaft

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▼ Load Profiles and Energy Costs

Expected pump life: 10.0 years	Load Profile #1	Load Profile #2	Load Profile #3	Load Profile #4	Load Profile #5	Total
Flow: ○ USgpm ◉ % of rated flow	100.0	80.00	50.00			-
Operation: ○ hours per year ◉ % of year	50	25	25	0	0	100
Energy cost, present value ($ per kWh)	0.10	0.10	0.10	0.10	0.10	-
Enable	☑	☑	☑	☑	☑	-
Pump speed, rated (rpm) ○ Synchronous ◉ Variable	3,500	2,800	1,750	0	0	-
Head (ft)	200.0	128.0	50.00	0.00	0.00	-
Efficiency (%)	70.1	70.1	69.6	0.0	0.0	-
Power, rated (hp)	21.55	11.03	2.71	0.00	0.00	-
Motor efficiency (%)	100.00	100.00	100.00	100.00	100.00	-
Drive/gear efficiency (%)	100.00	100.00	100.00	100.00	100.00	-
System curve	System Curve #1	System Curve #1	System Curve #1	System Curve #1	System Curve #1	-
Energy, total (kWh)	703,907.4	180,200.3	44,305.8	0.00	0.00	928,413.5
Energy cost, per year	$7,039.07	$1,802.00	$443.06	$0.00	$0.00	$9,284.14
Energy cost, total present value	$60,313.38	$15,440.22	$3,796.29	$0.00	$0.00	$79,549.89

▼ Life Cycle Cost

Additional Annual Costs ($)		Additional One-time Costs, Year 0 ($)		Interest and Inflation Rates	
Routine maintenance cost	0.00	Initial investement cost	0.00	Interest rate, %	6.00
Repair cost	0.00	Installation and commissioning cost	0.00	Inflation rate, %	3.00
Operating cost	0.00	Other one-time costs (Year 0)	0.00	**Total Net Present Value Costs**	
Downtime cost	0.00	**Additional One-time Costs, Year 10.0 ($)**		Total energy cost	: $79,549.89
Environmental cost	0.00	Decommissioning cost	0.00	Total additional annual cost	: $0.00
Other annual costs	0.00	Other one-time costs (Year 10.0)	0.00	Total additional one-time cost	: $0.00
Total, present value	: $0.00	Total, present value	: $0.00	Total life cycle cost	: $79,549.89

FIGURE 18 Life cycle cost evaluation sheet (Courtesy Intelliquip, LLC)

design, or the allowable temperature range for a given material. Using these design rules, a configurator could automatically upgrade a flange rating on a casing based on casing pressure or restrict the use of a gasket material as a function of temperature. Similarly, configuration rules ensure that a complete and permissible pump scope-of-supply is generated. For example, the mechanical seal type dictates the list of allowable seal flush piping options or the choice of pump and motor automatically defines coupling size required. These rules are coded into a knowledge base and serve to enforce product standardization and reduce configuration errors.

The pump selection program and the product configurator are the hub of the computer-aided tools used by the pump supplier to produce customized quotations. Supplementary programs are used to develop additional information such as pump performance curves, data sheets, general arrangement/outline drawings, and price makeups. These programs have reduced the quotation cycle time while improving the quality and accuracy of the technical and commercial information required by the purchaser.

A graphical description showing the interface of a pump selector and configurator are found in Figure 19. Important features of this model are:

- Workflow modeled after the well-known steps in the selection and configuration of pumping systems starting with defining the conditions of service, product selection and pump selection, and ending with configuration and quote generation.
- A rules engine and associated knowledgebase. Together, these assure reliable and consistent application of design rules, applications guidelines, and expert calculation methodologies traditionally known to only the most seasoned applications engineers.
- One knowledgebase referenced to produce all relevant quotation documents, including technical data sheets, general arrangement drawings, and price makeups.

ELECTRONIC DATA EXCHANGE The communication of inquiry and quotation information is still predominately managed through paper-based methods in spite of the increasing use of computer-aided design, selection, inquiry, and quotation systems. The exchange of pump technical data has been traditionally handled using the pump data sheet. Both purchaser and manufacturer have developed their own data sheet formats. Consequently, each data sheet exchange requires a laborious translation and interpretation of information from one data sheet format to another. This interpretation and translation requires highly skilled engineers and can be a source of errors and delays in completing the selection and purchase decisions.

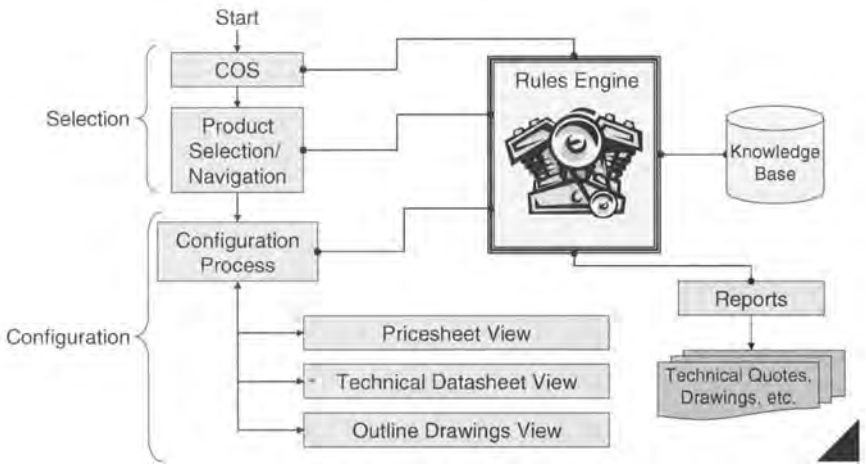


FIGURE 19 Selection and configuration workflow process (Courtesy Intelliquip, LLC)

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With the implementation of computerized selection programs and bid-tab programs, this manual translation is an untapped opportunity to streamline the data communication and purchasing process. In response to this situation, some firms have developed proprietary data exchange formats to leverage their own proprietary systems. Unfortunately, that approach has only shifted the burden to their trading partners who must develop special data translators to accommodate this wide variety of data sheet formats. It has been shown that the total number of translators needed for M data exchange formats follows the relationship, $M \sim (M - 1)$. Therefore, 3 unique data formats require 6 translators, whereas 10 unique formats require 90 translators.

To address this problem, standards organizations have developed neutral data exchange specifications to perform electronic data interchange (EDI). Well-known standards include UN/EDIFACT and ANSI X12 to support basic buyer and seller transactions. However, exchange of technical information such as that found in purchaser–supplier transactions in the pump industry is not supported by these EDI standards. Electronic data exchange of pump technical data is only possible through the efforts of pump industry related standards organizations. One such specification was developed under the auspices of the American Petroleum Institute's Standard 610, Centrifugal Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services (eighth edition, August, 1995). Since that time, the emergence of the World Wide Web on the Internet, and XML have provided a more fertile foundation for pump-related electronic data exchange.

To address these challenges and the need to automate the reliable exchange of equipment information, the FIATECH organization started the AEX (Automating Equipment Information Exchange) Project. This nonprofit industry consortium is working with other industry organizations and standards organizations to provide electronic data exchange methods to support the design, selection, and procurement of equipment involved in capital facilities projects, including not only pumps but valves, heat exchangers, compressors, motors, seals, and so on. AEX is now delivering XML schemas for a number of these equipment types. Simply stated, XML is a user-definable text file format that provides the capability to store highly structured digital information in a standard, *software neutral* text file. XML will work on virtually any computer hardware and operating system platform and with any software program. As a result XML can be used both for software neutral data exchange today, and for long-term data archival and reuse. This work, in coordination with other standards organizations such as the Hydraulic Institute (HI), the American Petroleum Institute (API), American Society for Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), Design Institute for Physical Properties (DIPPR), and Process Industry Practices (PIP) are expected to be used in actual purchaser–supplier transactions in the next few years.

THE INTERNET The use of the Internet has grown dramatically since the advent of the World Wide Web in the mid-1990s. The rate of change on the Internet is witnessed in terms of days rather than years. Information or Web sites that exist on the Internet today may be eliminated or replaced by tomorrow. As a consequence, any references about the use of the Internet for the selection and purchasing of pumping equipment will most likely be obsolete at the time the reader sees it. Nonetheless, it is necessary to anticipate the effect the Internet will have on this process in the pump industry.

The Internet is rapidly becoming the medium by which business-to-business (B-to-B) or electronic commerce will be conducted between trading partners in the future. The pump system design, selection, configuration, and purchasing systems are rapidly becoming Web-enabled, allowing users to perform these activities directly on the Internet. Similarly, the process of electronic data exchange will be performed across the Internet using structured data exchange methods that eliminate the manual reentry of technical information between different systems. Standards created by either international standards bodies or a de facto standard created by a consortium or commercial entity will provide the necessary framework for this data exchange.

Finally, the actual business transactions involving inquiry, quotation, purchase order, and invoice will expand over the Internet. This process will evolve starting with commodity-type pumps and eventually will involve more sophisticated types of pumping equipment. Interactions during the order fulfillment cycle will be available such as order

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status information. Drawings, certifications, test data, and other technical information will be available as secure business transactions. The use of voice and video are already supplanting many face-to-face meetings or equipment inspections that often involve detailed coordination and expensive travel.

SUMMARY

The basic activities involved in the selection and purchasing of pumps as described at the beginning of this chapter are substantially the same now as in the past, and will be in the future. However, the processes and technologies employed in performing these activities are changing rapidly because of new system and information technologies, the influence of globalization and the cost of energy. The motivations surrounding these changes are driven by the desire for shorter cycle times, higher quality, and lower costs in the entire pump system life cycle. These objectives are driven by the availability of new and emerging information technologies that offer a seamless and structured flow of information between the purchaser's and the supplier's sales, applications, engineering, and manufacturing functions. The availability of computer systems guarantees only that the infrastructure is in place to achieve the anticipated benefits. However, common work practices in both an intercompany and intracompany environment must be adopted and adapted to these new technologies. These process changes, not the availability of new information technologies, will govern the speed at which the pump industry changes in the future. The information age will support, but not replace, the basic requirement to design, select, configure, and manufacture high-quality and reliable pumps that support a substantial part of our modern life and industry today.

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C • H • A • P • T • E • R • 15

INSTALLATION, OPERATION, AND MAINTENANCE

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C. C. Heald

The information contained in this chapter is general and should be supplemented by the specific instructions prepared by the manufacturer of the pump in question.

INSTALLATION

Instruction Books Instruction books, or Technical Manuals, are intended to help keep the pumps in an efficient and reliable condition at all times. It is necessary, therefore, that instruction books be available to all personnel involved in this function.

Preparation for Shipment After a pump is assembled in the manufacturer's shop, it should be suitably prepared for the type of shipment the purchaser has specified. This can include blocking of the rotor, when necessary. If the rotor is blocked, this should be identified by weather-resistant tags. As shipped, the equipment should be suitable for at least 6 months of outdoor, uncovered storage from the time of shipment, with no disassembly required at the time of installation and operation except for possible inspection of bearings and seals. If storage for a longer time is required and specified, the manufacturer and purchaser should agree on the preparation and storage procedures and requirements prior to shipment.

The pump and driver should be prepared for shipment only after all testing and inspection processes have been completed and the equipment has been released for shipment by the purchaser. If packing was used for testing, it should be removed before shipment. Usually pumps are not disassembled after performance testing. The pump must be completely drained and dried, and all internal parts should be coated with a suitable rust preventative within 4 hours of testing. Alternatively, within 4 hours of testing, the pump and seal chamber should be drained to the extent practical, filled with a water-displacing inhibitor, and redrained before shipment.

Flanged openings should be provided with covers, usually metal, at least $\frac{3}{16}$ in (5 mm) thick and sealed with an elastomeric gasket. Threaded openings should be closed with steel caps or plugs. Any openings that have been prepared for welding in the field should be provided with closures designed to protect against damage to the welding surface as well as the entrance of foreign materials into the equipment.

Exposed shafts and shaft couplings should be wrapped with waterproof cloth or paper and sealed with oil-proof adhesive tape. Bearing assemblies should be protected from moisture and dirt (dust). If vapor phase inhibitor crystals in bags are installed to absorb moisture, the location of the bags should be tagged so they will be removed before installation of the equipment in the field.

Usually a copy of the manufacturer's standard installation instructions is packed inside the equipment prior to shipment.

Care of Equipment in the Field The manufacturer should provide the purchaser with the instructions necessary to preserve the integrity of the original storage preparation when the equipment is received at the job site and prior to start-up. All equipment and materials should be stored free from direct ground contact and away from areas subject to collecting water. Indoor storage should be used whenever possible.

All carbon and low alloy steel surfaces should be protected from any contact with corrosive environments to prevent rust formation. All items with machined surfaces should be stored to facilitate periodic examination for damage or rust. Storage areas should be kept clean and free from such contaminants as concrete chipping, sanding, and painting.

Periodic rotation of equipment shafts should be performed in accordance with the equipment manufacturer's instructions for the specific equipment type and preservation methods used. When rotation is performed, determine first that all shipping blocks on rotating components have been removed and that there is adequate lubrication before rotation.

Certain preservatives and storage lubricants can affect safety and operating life of the equipment, especially if they react with the process fluid or operating lubricant. The

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installer should ensure that all preservative and storage lubricants are suitable for the specific application. Preservatives should not be used on surfaces where prohibited by the process or application.

Pump Location Working space must be checked to assure adequate accessibility for maintenance. Axially split casing horizontal pumps require sufficient headroom to lift the upper half of the casing free of the rotor. The inner assembly of radially split multistage centrifugal pumps is removed axially. Space must be provided so the assembly can be pulled out without canting it. For large pumps with heavy casings and rotors, a traveling crane or other facility for attaching a hoist should be provided over the pump location.

Pumps should be located as close as practicable to the source of liquid supply. Whenever possible, the pump centerline should be placed below the level of the liquid in the suction reservoir.

Foundations Foundations may consist of any structure heavy enough to afford permanent rigid support to the full area of the baseplate and to absorb any normal strains or shocks. Reinforced concrete foundations built up from solid ground are the most satisfactory. Although most pumping units are mounted on baseplates, very large equipment may be mounted directly on the foundations by using soleplates under the pump and driver feet. Misalignment is corrected with shims.

The space required by the pumping unit and the location of the foundation bolts are determined from the drawings supplied by the manufacturer. Each foundation bolt (Figure 1) should be surrounded by a pipe sleeve three or four diameters larger than the bolt. After the concrete foundations are poured, the pipe is held solidly in place but the bolt may be moved to conform to the corresponding hole in the baseplate.

When a unit is mounted on steelwork or some other structure, it should be placed directly over or as near as possible to the main members, beams, or walls and should be supported so the baseplate cannot be distorted or the alignment disturbed by any yielding or springing of the structure or of the baseplate.

Mounting of Vertical Wet-Pit Pumps A curb ring or soleplate must be used as a bearing surface for the support flange of a vertical wet-pit pump. The mounting face must be machined because the curb ring or soleplate will be used in aligning the pump.

If the discharge pipe is located below the support flange of the pump (belowground discharge), the curb ring or soleplate must be large enough to pass the discharge elbow during assembly. A rectangular ring should be used (Figure 2). If the discharge pipe is located above the support flange (aboveground discharge), a round curb ring or soleplate should be provided with clearance on its inner diameter to pass all sections of the pump below the support flange (Figure 3). A typical method of arranging a grouted soleplate for vertical pumps is shown in Figure 4.

If the discharge is belowground and an expansion joint is used, it is necessary to determine the moment that may be imposed on the structure. The pump casing should be attached securely to some rigid structural members with tie rods. If vertical wet-pit pumps are very long, some steadying device is required irrespective of the location of the discharge or of the type of pipe connection. Tie rods can be used to connect the unit to a wall, or a small clearance around a flange can be used to prevent excessive displacement of the pump in the horizontal plane.

Alignment When a complete unit is assembled at the factory, the baseplate is placed on a flat, even surface; the pump and driver are mounted on the baseplate; and the coupling halves are accurately aligned, using shims under the driver mounting surfaces where necessary. The pump is usually doweled to the baseplate at the factory, but the driver is left to be doweled after installation at the site.

The unit should be supported over the foundation by short strips of steel plate or shim stock close to the foundation bolts, allowing a space of $\frac{3}{4}$ to 2 in (2 to 5 cm) between the bottom of the baseplate and the top of the foundation for grouting. The shim stock should extend fully across the supporting edge of the baseplate. The coupling bolts should be

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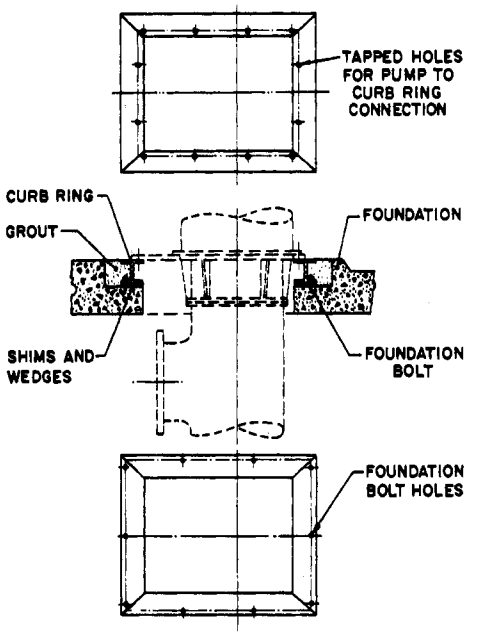


FIGURE 1 Foundation bolt

FIGURE 2 Rectangular curb ring for belowground discharge vertical pump

removed before the unit is leveled and the coupling halves are aligned. Where possible, it is preferable to place the level on some exposed part of the pump shaft, sleeve, or planed surface of the pump casing. The steel supporting strips or shim stock under the baseplate should be adjusted until the pump shaft is level, the suction and discharge flanges are vertical or horizontal as required, and the pump is at the specified height and location. When the baseplate has been leveled, the nuts on the foundation bolts should be made handtight.

During this leveling operation, accurate alignment of the unbolted coupling halves must be maintained. A straightedge should be placed across the top and sides of the coupling, and at the same time the faces of the coupling halves should be checked with a tapered thickness gage or with feeler gages (Figure 5) to see that they are parallel. For all alignment checks, including parallelism of coupling faces, both shafts should be pressed hard over to one side when taking readings.

When the peripheries of the coupling halves are true circles of equal diameter and the faces are flat and perpendicular to the shaft axes, exact alignment exists when the distance between the faces is the same at all points and when a straightedge lies squarely across the rims at any point. If the faces are not parallel, the thickness gage or feelers will show variation at different points. If one coupling is higher than the other, the amount may be determined by the straightedge and feeler gages.

Sometimes coupling halves are not true circles or are not of identical diameter because of manufacturing tolerances. To check the trueness of either coupling half, rotate it while holding the other coupling half stationary and check the alignment at each quarter turn of the half being rotated. Then the half previously held stationary should be revolved and the alignment checked. A variation within manufacturing limits may be found in either of the half-couplings, and proper allowance for this must be made when aligning the unit.

A more exact method for checking alignment that is recommended requires the use of a dial indicator. With the indicator bolted to the pump half of the coupling, both radial and

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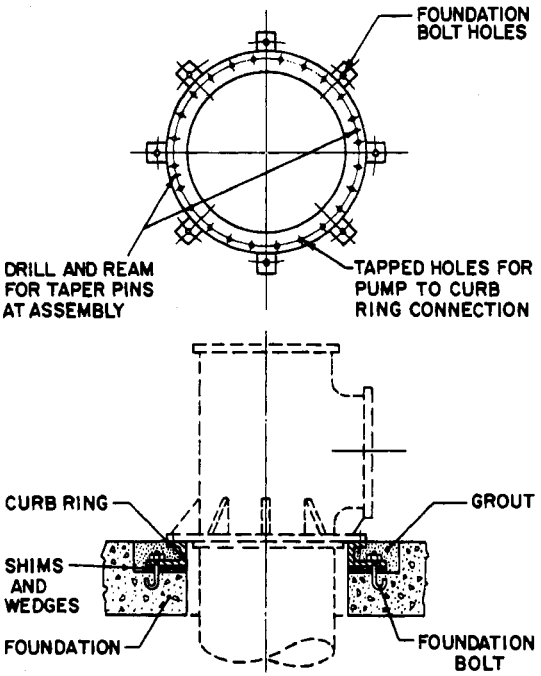


FIGURE 3 Round curb ring for aboveground discharge vertical pump

axial alignment can be checked, as illustrated in Figure 6. This method is called face-and-rim alignment. With the button resting on the periphery of the other coupling half, the dial should be set at zero and a mark chalked on the coupling half at the point where the button rests. For any check (top, bottom, or sides), both shafts should be rotated the same amount, that is, all readings on the dial should be made with the button on the chalk mark. The dial readings will indicate whether the driver must be raised, lowered, or moved to either side. After any movement, it is necessary to check that the coupling faces remain parallel to one another.

For example, if the dial reading at the starting point is set to zero and the diametrically opposite reading at the bottom or sides shows ±0.020 in (±0.508 mm), the driver must be raised or lowered by shimming or moved to one side or the other by half of this reading. The same procedure is used to align gear couplings, but the coupling covers must first be moved back out of the way and all measurements should be made on the coupling hubs.

When a spacer-type coupling connects the pump to its driver, a dial indicator should be used to check the alignment (Figure 7). The spacer between the coupling halves should be removed to expose the coupling hubs. The coupling nut on the end of the shaft may be used

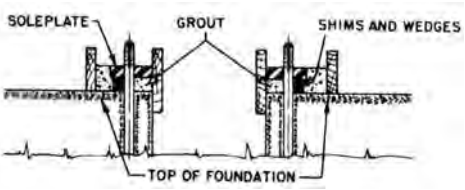


FIGURE 4 Grouting form for vertical pump soleplate (ANSI B-58.1 [AWWAE 101-61])

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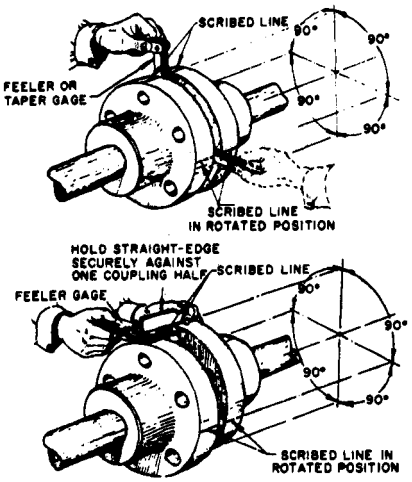


FIGURE 5 Coupling alignment using feeler gages

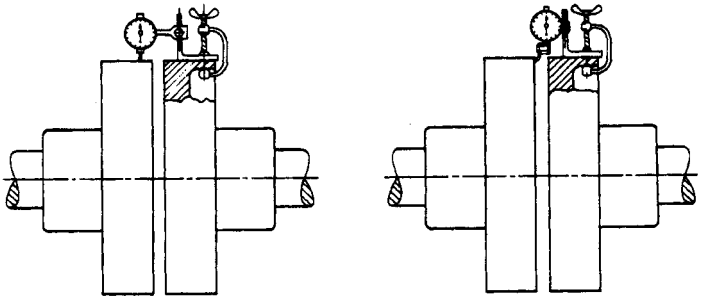


FIGURE 6 Use of dial indicator for face-and-rim alignment of standard coupling

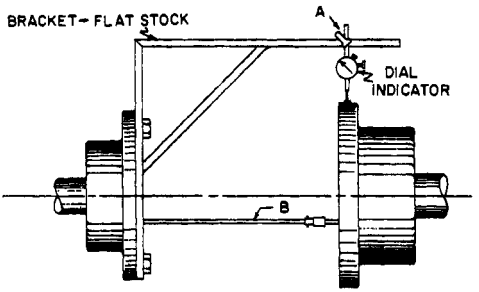


FIGURE 7 Use of dial indicator for face-and-rim alignment of spacer-type coupling

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to clamp a suitable extension arm or bracket long enough to extend across the space between the coupling hubs. The dial indicator is mounted on this arm, and alignment is checked for both concentricity of the hub diameters and parallelism of the hub faces. Changing the arm from one hub to the other provides an additional check. The dial extension bracket must be checked for sag, and readings must be corrected accordingly.

The clearance between the faces of the coupling hubs and the ends of the shafts should be such that they cannot touch, rub, or exert a pull on either pump or driver. The amount of this clearance may vary with the size and type of coupling used. Sufficient clearance will allow unhampered endwise movement of the shaft of the driving element to the limit of its bearing clearance. On motor-driven units, the magnetic center of the motor will determine the running position of the motor half-coupling. This position should be checked by running the motor uncoupled. This will also permit checking the direction of rotation of the motor. If current is not available at the time of installation, move the motor shaft in both directions as far as the bearings will permit and adjust the shaft centrally between these limits. The unit should then be assembled with the correct gap between the coupling halves.

Large horizontal sleeve-bearing motors are not generally equipped with thrust bearings. The motor rotor is permitted to float, and as it will seek its magnetic center, an axial force of rather small magnitude can cause it to move off this center. Sometimes it will move enough to cause the shaft collar to contact and possibly damage the bearing. To avoid this, a limited-end-float coupling is used between the pump and the motor on all large units to restrict the motor rotor (Section 9.3). The setting of axial clearances for such units should be given by the manufacturer in the instruction books and elevation drawings.

When the pump handles a liquid at other than ambient temperature or when it is driven by a steam turbine, the expansion of the pump or turbine at operating temperature will alter the vertical alignment. Alignment should be made at ambient temperature with suitable allowances for the changes in pump and driver centerlines after expansion. The final alignment must be made with the pump and driver at their normal temperatures and adjusted as required before the pump is placed into permanent service.

For large installations, particularly with steam-turbine-driven pumps, more sophisticated alignment methods are sometimes employed, using proximity probes and optical instruments. Such procedures permit checking the effect of temperature changes and machine strains caused by piping stresses while the unit is in operation. When such procedures are recommended, they are included with the manufacturer's instructions.

When the unit has been accurately leveled and aligned, the hold-down bolts should be gently and evenly tightened before grouting. The alignment must be rechecked after the suction and discharge piping has been bolted to the pump to test the effect of piping strains. This can be done by loosening the bolts and reading the movement of the pump, if any, with dial indicators.

The pump and driver alignment should be occasionally rechecked because misalignment may develop from piping strains after a unit has been operating for some time. This is especially true when the pump handles hot liquids because there may be a growth or change in the shape of the piping. Pipe flanges at the pump should be disconnected after a period of operation to check the effect of the expansion of the piping, and adjustments should be made to compensate for this.

For a further discussion of hot and cold alignment, face-and-rim versus reverse dial methods measurement of dial bracket sag, and graphical alignment plotting procedure, refer to Section 2.1.4.

Grouting Ordinarily, the baseplate is grouted before the piping connections are made and before the alignment of the coupling halves is finally rechecked. The purpose of grouting is to prevent lateral shifting of the baseplate, to increase the mass to reduce vibration, and to fill in irregularities in the foundation. Generally, it is recommended that all permanently installed pumping equipment be supported by a reinforced concrete foundation. Some pumps that require an elevated installation may be supported on structural steel structures, but care must be taken to ensure that such structures are of adequate stiffness and strength.

Foundation dimensions must consider the size and arrangement of the pump and driver, the piping arrangement and anticipated piping loads, anchor bolt placement, and minimum

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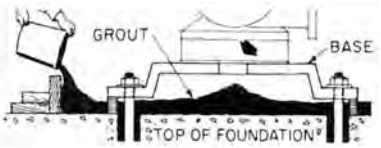


FIGURE 8 Application of grouting. Grout is added until the entire space under the base is filled. Holes in the base (arrow) allow air to escape and permit working of the grout to release air pockets.

dimensions required for servicing the equipment. Vertically suspended canned pump foundations should be designed so the pump can is directly attached to a mounting plate and is removable without disturbing the grout.

Foundation materials must be properly resistant to chemicals and oils. If the environment is aggressive, protective coatings or covers should be considered to protect the foundation and reinforcing steel. Two types of grout are commonly used: epoxy and cement-based. Epoxy grout, although usually more expensive, has significant advantages in higher bond strength and nonporous finished surface characteristics that generally make it the material of choice for most pump installations.

When the foundation has been prepared, the grout material has been selected, and the baseplate has been properly positioned on the foundation, a preliminary coupling alignment is made to ensure that final alignment is possible after the baseplate is finally grouted. After this alignment check is successfully accomplished, the grout is added through the holes in the baseplate. To retain the grout in place, a leak-tight form is built around the outside of the baseplate. Grout is added until the entire space under the baseplate is filled to the top of the underside (Figure 8). The grout holes and vent holes in the baseplate allow air to escape as the grout fills the cavity. A stiff wire may be used through the grout holes to work the grout and release any air pockets. It is usually best to start at one end and force the air out as the grout proceeds toward the other end of the baseplate. Leveling shims and wedges used to level the baseplate should be left in place after grouting.

When the grout has properly cured, voids have been filled and forms have been removed, the exposed surfaces of the grout and foundation can be properly finished. The foundation anchor/hold-down bolts should be finally torqued to the proper values and the coupling halves can be rechecked for alignment.

The pump and driver alignment must be rechecked thoroughly after the grout has hardened permanently, and at reasonable intervals thereafter.

Doweling of Pump and Driver When the pump handles hot liquids, doweling of both the pump and its driver should be delayed until the unit has been operated. A final recheck of alignment with the coupling bolts removed and with the pump and driver at operating temperature is advisable before doweling.

Large pumps handling hot liquids are usually doweled near the coupling end, allowing the pump to expand from that end out. Sometimes the other end is provided with a key and a keyway in the casing foot and the baseplate.

PIPING

Suction Piping The suction piping should be as direct and short as possible. If a long suction line is required, the pipe size should be increased to reduce frictional losses. (The exception to this recommendation is in the case of boiler-feed pumps, where difficulties may arise during transient conditions of load change if the suction piping volume is excessive. This is a special and complex subject, and the manufacturer should be consulted.) Where the pump must lift the liquid from a lower level, the suction piping should be laid out with a continual rise toward the pump, avoiding high spots in the line to prevent the formation of air pockets. Where a static suction head will exist, the pump suction piping should slope continuously downward to the pump.

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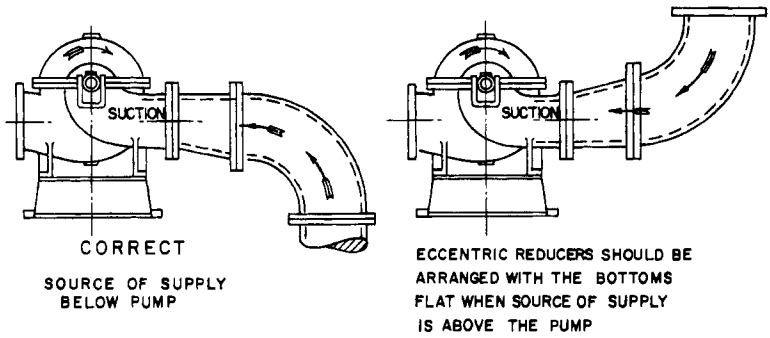


FIGURE 9 Recommended installation of reducers at pump suction

Generally, the suction line is larger than the pump suction nozzle and eccentric reducers should be used. If the source of supply is above the pump centerline, the reducer should be installed straight side up. If the source of supply is above the pump, the straight side of the reducer should be at the bottom (Figure 9). Installing eccentric reducers with a change in diameters greater than 4 in (10 cm) could disturb the suction flow. If such a change is necessary, it is advisable to use properly vented concentric reducers.

Elbows and other fittings next to the pump suction should be carefully arranged, or the flow into the pump impeller will be disturbed. Long-radius elbows are preferred for suction lines because they create less friction and provide a more uniform flow distribution than standard elbows.

It is extremely important to avoid the formation of vortices at the suction of both wet-pit and dry-pit pump installations. For a discussion of this and other suction conditions recommendations, see Sections 13.1 and 13.2.

Discharge Piping Generally both a check valve and a gate valve are installed in the discharge line. The check valve is placed between the pump and the gate valve and protects the pump from reverse flow in the event of unexpected driver failure or from reverse flow from another operating pump. The gate valve is used when priming the pump or when shutting it down for inspection and repairs. Manually operated valves that are difficult to reach should be fitted with a sprocket rim wheel and chain. In many cases, discharge gate valves are motorized and can be operated by remote control.

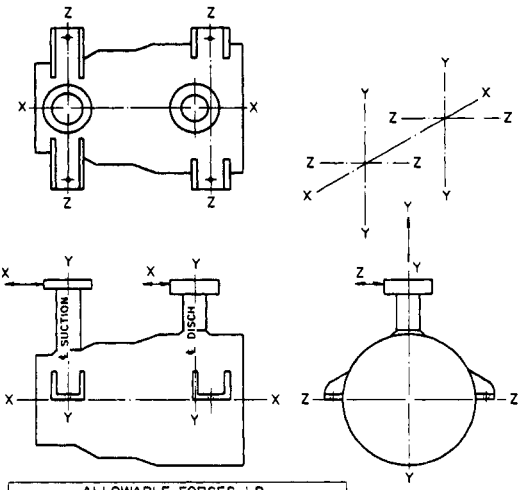
Piping Strains Cast iron pumps are never provided with raised face flanges. If steel suction or discharge piping is used, the pipe flanges should be of the flat-face and not the raised-face type. Full-face gaskets must be used with cast iron pumps.

Piping should not impose excessive forces and moments on the pump to which it is connected because these might spring the pump or pull it out of position. Piping flanges must be brought squarely together before the bolts are tightened. The suction and discharge piping and all associated valves, strainers, and so on should be supported and anchored near to but independent of the pump, so no strain will be transmitted to the pump casing.

There are four factors to be considered in determining the effect of nozzle loads: material stress in pump nozzles resulting from forces and bending moments, distortion of internal moving parts affecting critical clearances, stresses in pump hold-down bolts, and distortion in pump supports and baseplates resulting in driver coupling misalignment. API Standard 610 (Centrifugal Pumps for Refinery, Heavy Duty Chemical, and Gas Industry Services) provides guidelines for limiting the magnitude of nozzle loads and moments on pumps with suction nozzles 16 in (41 cm) and smaller and with casings constructed of steel or alloy steel.

With large pumps or when major temperature changes are expected, the pump manufacturer generally indicates to the user the maximum moments and forces that can be

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ALLOWABLE FORCES, LB			
	X	Y	Z
SUCTION	15,800	15,800	11,200
DISCHARGE	19,300	15,800	11,200
TOTAL	19,300	15,800	11,200

ALLOWABLE MOMENTS, FT-LB			
	X-X AXIS	Y-Y AXIS	Z-Z AXIS
SUCTION	22,900	45,400	22,900
DISCHARGE	21,600	43,200	21,600
TOTAL	22,900	45,400	22,900

FIGURE 10 Diagram of typical permissible pipe stresses and moments for a radially split double-casing multistage pump with top suction and discharge (1 N = 0.225 lb; 1 N · m = 0.737 ft · lb)

imposed on the pump by the piping. A typical diagram is illustrated in Figure 10 for a radially split double-casing multistage pump with top suction and discharge.

Expansion Joints Expansion joints are sometimes used in the discharge and suction piping to avoid transmitting any piping strains caused by misalignment or by expansion when hot liquids are handled. On occasion, expansion joints are formed by looping the pipe. More often, they are of the slip-joint or corrugated-diaphragm type. However, they transmit to the pump a force equal to the area of the expansion joint times the pressure in the pipe. These forces can be of very significant magnitude, and it is impractical to design the pump casings, baseplates, and so on to withstand them. Consequently, when expansion joints are used, a suitable pipe anchor must be installed between them and the pump proper. Alternately, tie rods can be used to prevent the forces from being transmitted to the pump.

Suction Strainers Except for certain special designs, pumps are not intended to handle liquid containing foreign matter. If the particles are sufficiently large, such foreign matter can clog the pump, reduce its capacity, or even render it altogether incapable of pumping. Small particles of foreign matter may cause damage by lodging between close running clearances. Therefore, proper suction strainers may be required in the suction lines of pumps not specially designed to handle foreign matter.

In such an installation, the piping must first be thoroughly cleaned and flushed. The recommended practice is to flush all piping to waste before connecting it to the pump. Then a temporary strainer of appropriate size should be installed in the suction line as close to the pump as possible. This temporary strainer may have a finer mesh than the

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permanent strainer installed after the piping has been thoroughly cleaned of all possible mill scale or other foreign matter. The size of the mesh is generally recommended by the pump manufacturer. For further details on strainers, see Sections 11.1 and 13.1.

Venting and Draining Vent valves are generally installed at one or more high points of the pump casing waterways to provide a means of escape for air or vapor trapped in the casing. These valves are used during the priming of the pump or during operation if the pump should become air- or vapor-bound. In most cases, these valves need not be piped up away from the pump because their use is infrequent, and the vented air or vapors can be allowed to escape into the surrounding atmosphere. On the other hand, vents from pumps handling flammable, toxic, or corrosive fluids must be connected in such a way that they endanger neither the operating personnel nor the installation. The suction vents of pumps taking liquids from closed vessels under vacuum must be piped to the source of the pump suction above the liquid level.

All drain and drip connections should be piped to a point where the leakage can be disposed of or collected for reuse if worth reclaiming.

Warm-Up Piping When it is necessary for a pump to come up to operating temperature before it is started up or to keep it ready to start at rated temperature, provision should be made for a warm-up flow to pass through the pump. There are a number of arrangements used to accomplish this. If the pump operates under positive pressure on the suction, the pumped liquid can be permitted to drain out through the pump casing drain connection to some point at a pressure lower than the suction pressure (Figure 11). Alternately, some liquid can be made to flow back from the discharge header through a jumper line around the check valve into the pump and out into the suction header (Figure 12). An orifice is provided in this jumper line to regulate the amount of warm-up flow. Care must be exercised in such an installation to maintain the suction valve open (unless the warm-up line valve is closed, as when the pump is to be dismantled) lest the entire pump, suction valve, and suction piping be subjected to full discharge pressure.

The manufacturer's recommendations should be sought in all cases as to the best means of providing an adequate warm-up procedure. Care must be taken to ensure that the pump is warmed uniformly. Stratification of the warm-up flow, or inadequate warm-up flow volume, can result in casing distortion or rotor bowing, or both (Heald and Penry, 1988).

RELIEF VALVES Positive displacement pumps, such as rotary and reciprocating pumps, can develop discharge pressures much in excess of their maximum design pressures. To protect these pumps against excessive pressures when the discharge is throttled or shut off, a pressure relief valve might be used. Some pumps are provided with internal integral relief valves, but unless operation against a closed discharge is both infrequent and of very

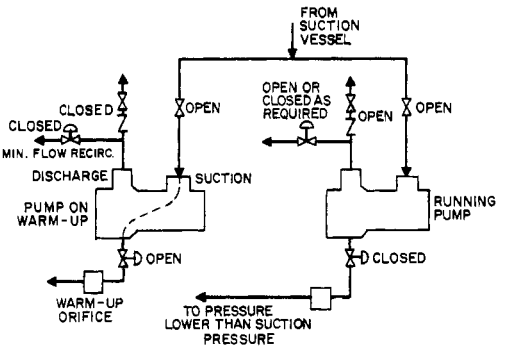


FIGURE 11 Arrangement for warm-up through pump casing drain connection

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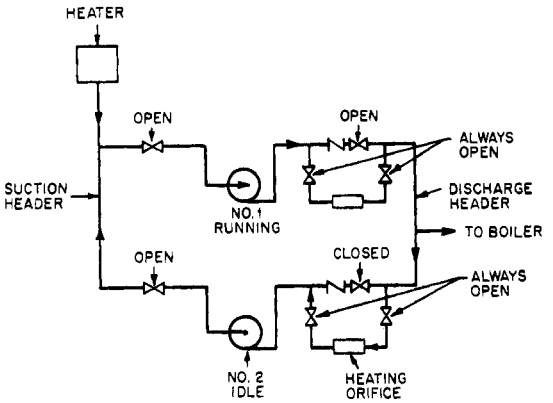


FIGURE 12 Arrangement for warm-up through jumper line around discharge check valve

short duration, a relief valve with an external return connection must be used and the liquid from the relief valve must be piped back to the source of supply.

Surge Chambers Generally, centrifugal pumps do not require surge chambers in their suction or discharge piping. Reciprocating pumps may have a suction and discharge piping layout that does not require compensation for variations in the flow velocity in the piping system. In many cases, however, reciprocating pump installations require surge chambers when the suction or discharge lines are of considerable length, when there is an appreciable static head on the discharge, when the liquid pumped is hot, or when it is desirable to smooth out variations in the discharge flow. The type, size, and arrangement of the surge chamber should be chosen on the basis of the manufacturer's recommendations. For more details, see Section 3.4.

Instrumentation There are a number of instruments that are essential to maintaining a close check on the performance and condition of a pump. A compound pressure gage should be connected to the suction of the pump, and a pressure gage should be connected to its discharge at the pressure taps that may be provided in the suction and discharge flanges. The gages should be mounted in a convenient location so they can be easily observed.

In addition, it is advisable to provide a flow-metering device. Depending upon the importance of the installation, indicating meters may be supplemented by recording attachments.

Whenever pumps incorporate various leakoff arrangements, such as a balancing device or pressure-reducing labyrinths, a check should be maintained on the quantity of these leakoffs by measuring orifices and differential gages installed in the leakoff lines.

Pumps operating in important or complex services or operating completely unattended by remote control may have additional instrumentation, such as speed indicators, vibration monitors, and bearing or casing temperature indicators. For more detail, see Sections 2.1.4 and 3.4.

OPERATION

Pumps are generally selected for a given capacity and total head when operating at rated speed. These characteristics are referred to as "rated conditions of service" and, with few exceptions, represent those conditions at or near which the pump will operate the greatest part of the time. Positive displacement pumps cannot operate at any greater flows than

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rated except by increasing their speed, nor can they operate at lower flows except by reducing their operating speed or bypassing some of the flow back to the source of supply. (See Section 3.5.)

On the other hand, centrifugal pumps can operate over a wide range of capacities, from near zero flow to well beyond the rated capacity. Because a centrifugal pump will always operate at the intersection of its head-capacity and system-head curves, the pump operating capacity may be altered either by throttling the pump discharge (hence altering the system-head curve) or by varying the pump speed (changing the pump head-capacity curve). This makes the centrifugal pump very flexible in a wide range of services and applications that require the pump to operate at capacities and heads differing considerably from the rated conditions. There are, however, some limitations imposed upon such operation by hydraulic, mechanical, or thermodynamic considerations (Section 2.1.3).

Operation of Centrifugal Pumps at Reduced Flows There are certain minimum operating flows that must be imposed on centrifugal pumps for either hydraulic or mechanical reasons. Four limiting factors must be considered: radial thrust, temperature rise, internal recirculation, and shape of the brake horsepower curve.

Radial thrust is discussed in Sections 2.1.3 and 2.2.1. For sustained operation, it is imperative to adhere to the minimum flow limits recommended by the pump manufacturer, which depend on the specific design of the pump casing and impeller.

The thermodynamic problem that arises when a centrifugal pump is operated at extremely reduced flows is caused by the heating up of the liquid handled. The difference between the brake horsepower consumed and the water horsepower developed represents the power losses in the pump, except for a small amount lost in the pump bearings. These power losses are converted to heat and transferred to the liquid passing through the pump.

If the pump were to operate against a completely closed valve, the power losses would be equal to the shutoff brake horsepower, and because there would be no flow through the pump, all this power would go into heating the small quantity of liquid contained in the pump casing. The pump casing would heat up, and a certain amount of heat would be dissipated by radiation and convection to the atmosphere. However, because the temperature rise in the liquid pumped could be quite rapid, it is generally safer to ignore the dissipation of heat through radiation and the absorption of heat by the casing. Calculations for determining the temperature rise in the pumped liquid are given in Section 2.1.3. The maximum permissible temperature rise in a centrifugal pump varies over a wide range, depending on the type of service and installation. For hot-water pumps, as on boiler-feed service, it is generally advisable to limit the temperature rise to about 15°F (8°C). As a general rule, the minimum permissible flow to hold the temperature rise in boiler-feed pumps to this value is 30 gpm for each 100 bhp (9.13 m^3/h per 100 kW) at shutoff. When the pump handles cold water, the temperature rise may be permitted to reach 50 or even 100°F (28 or 56°C). The minimum capacity based on thermodynamic considerations is then established as the capacity at which the temperature rise is the maximum permitted. Means and controls used to provide the necessary minimum flows are described in Section 10.

There are also hydraulic considerations that may affect the minimum flow at which a centrifugal pump can operate. In recent years, correlation has been developed between operation at low flows and the appearance of hydraulic pulsations both in the suction and in the discharge of centrifugal impellers. It has been proved that these pulsations are caused by the development of an internal recirculation at the inlet and discharge of an impeller at certain flows below the best-efficiency capacity. This subject is treated in Sections 2.1.3 and 2.1.4. The pump manufacturer's recommendations on minimum flows dictated by these considerations must be followed.

Priming With very few exceptions, no centrifugal pump should ever be started until it is fully primed; that is, until it has been filled with the liquid pumped and all the air contained in the pump has been allowed to escape. The exceptions involve self-priming pumps

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and some special large-capacity, low-head, and low-speed installations where it is not practical to prime the pump prior to starting; the priming takes place almost simultaneously with the starting in these cases. For further details, see Section 2.2.2.

Reciprocating pumps of the piston or plunger type are in principle self-priming. However, if quick starting is required, priming connections should be piped to a supply above the pump.

Positive displacement pumps of the rotating type, such as rotary or screw pumps, have clearances that allow the liquid in the pump to drain back to the suction. When pumping low-viscosity liquids, the pump may completely dry out when it is idle. In such cases a foot valve may be used to help keep the pump primed. Alternately, a vacuum device may be used to prime the pump. When handling liquids of higher viscosity, foot valves are usually not required because liquid is retained in the clearances and acts as a seal when the pump is restarted. However, before the initial start of a rotating positive displacement pump, some of the liquid to be pumped should be introduced through the discharge side of the pump to wet the rotating element.

The various methods and arrangements used for priming pumps are described in Section 2.2.2.

Final Checks Before Start-Up A few final checks are recommended before a pump is placed into service for its initial start. For pumps with journal bearings, the bearing covers should be removed, and the bearings should be flushed and thoroughly cleaned. They should then be filled with new lubricant in accordance with the manufacturer's recommendations.

With the coupling disconnected, the driver should be tested again for correct direction of rotation. Generally an arrow on the pump casing indicates the correct rotation.

It must be possible to rotate the rotor of a centrifugal pump by hand, and in the case of a pump handling hot liquids, the rotor must be free to rotate with the pump cold or hot. If the rotor is bound or even drags slightly, do not operate the pump until the cause of the trouble is determined or corrected.

Starting and Stopping Procedures The steps necessary to start a centrifugal pump depend upon its type and upon the service on which it is installed. For example, standby pumps are generally held ready for immediate starting. The suction and discharge gate valves are held open, and reverse flow through the pump is prevented by the check valve in the discharge line.

The methods followed in starting are greatly influenced by the shape of the power-capacity curve of the pump. High- and medium-head pumps (low and medium specific speeds) have power curves that rise from zero flow to the normal capacity condition. Such pumps should be started against a closed discharge valve to reduce the starting load on the driver. A check valve is equivalent to a closed valve for this purpose, as long as another pump is already on the line. The check valve will not lift until the pump being started comes up to a speed sufficient to generate a head high enough to lift the check valve from its seat. If a pump is started with a closed discharge valve, the recirculation bypass line must be open to prevent overheating.

Low-head pumps (high specific speed) of the mixed-flow and propeller type have power curves that rise sharply with a reduction in capacity; they should be started with the discharge valve wide open against a check valve, if required, to prevent backflow.

Assuming that the pump in question is motor-driven, that its shutoff power does not exceed the safe motor power, and that it is to be started against a closed gate valve, the starting procedure is as follows:

1. Prime the pump, opening the suction valve, closing the drains, and so on, to prepare the pump for operation.
2. Open the valve in the cooling supply to the bearings, where applicable.
3. Open the valve in the cooling supply if the seal chambers are liquid-cooled.
4. Open the valve in the sealing liquid supply if the pump is so fitted.

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5. Open the warm-up valve of a pump handling hot liquids if the pump is not normally kept at operating temperature. When the pump is warmed up, close the valve.
6. Open the valve in the recirculating line if the pump should not be operated against dead shutoff.
7. Start the motor.
8. Open the discharge valve slowly.
9. For pumps equipped with mechanical seals, check for seal leakage: there should be none.
10. For pump with shelf packing, observe the leakage from the stuffing boxes and adjust the sealing liquid valve for proper flow to ensure the lubrication of the packing. If the packing is new, do not tighten up on the gland immediately, but let the packing run in before reducing the leakage through the stuffing boxes.
11. Check the general mechanical operation of the pump and motor.
12. Close the valve in the recirculating line when there is sufficient flow through the pump to prevent overheating.

If the pump is to be started against a closed check valve with the discharge gate valve open, the steps are the same, except that the discharge gate valve is opened some time before the motor is started.

In certain cases, cooling to the bearings and flush liquid to the mechanical seals or to the packing seal cages is provided by the pump. This, of course, eliminates the need for the steps listed for the cooling and sealing supply.

Just as in starting a pump, the stopping procedure depends upon the type and service of the pump. Generally, the steps followed to stop a pump that can operate against a closed gate valve are

1. Open the valve in the recirculating line.
2. Close the gate valve.
3. Stop the motor.
4. Open the warm-up valve if the pump is to be kept at operating temperature.
5. Close the valve in the cooling supply to the bearings and seal chambers.
6. If the sealing liquid supply is not required while the pump is idle, close the valve in this supply line.
7. Close the suction valve, open the drain valves, and so on, as required by the particular installation or if the pump is to be opened up for inspection.

If the pump is of a type that does not permit operation against a closed gate valve, steps 2 and 3 are reversed.

In general, the starting and stopping of steam-turbine-driven pumps require the same steps and sequence prescribed for a motor-driven pump. As a rule, steam turbines have various drains and seals that must be opened or closed before and after operation. Similarly, many turbines require warming up before starting. Finally, some turbines require turning gear operation if they are kept on the line ready to start up. The operator should therefore follow the steps outlined by the turbine manufacturer in starting and stopping the turbine.

Most of the steps listed for starting and stopping centrifugal pumps are equally applicable to positive displacement pumps. There are, however, two notable exceptions:

1. Never operate a positive displacement pump against a closed discharge. If the gate valve on the discharge must be closed, always start the pump with the recirculation bypass valve open.
2. Always open the steam cylinder drain cocks of a steam reciprocating pump before starting, to allow condensate to escape and to prevent damage to the cylinder heads.

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Auxiliary Services on Standby Pumps Standby pumps are frequently started up from a remote location, and several methods of operation are available for the auxiliary services, such as the cooling supply to bearings or seal chambers:

1. A constant flow may be kept through jackets or oil coolers and through stuffing box lantern rings or seal chambers whether the pump is running or on standby service.
2. The service connections may be opened automatically whenever the pump is started up.
3. The service connections may be kept closed while the pump is idle, and the operator may be instructed to open them shortly after the pump has been put on the line automatically.

The choice among these methods must be dictated by the specific circumstances surrounding each case. There are, however, certain cases where sealing liquid supply to the pump seal chambers must be maintained whether the pump is running or not. This is the case when the pump handles a liquid that is corrosive or that may crystallize and deposit on the sealing components. It is also the case when the sealing supply is used to prevent air infiltration into a pump when it is operating under a vacuum.

Restarting Motor-Driven Pumps After Power Failure Assuming that power failure will not cause the pump to go into reverse rotation; that is, that a check valve will protect the pump against reverse flow, there is generally no reason why the pump should not be permitted to restart after current has been re-established. Whether the pump will start again automatically when power is restored will depend on the type of motor control used. (Section 2.1.3 and Section 11.1 give reasons why some pumps should not be started in reverse.)

Because pumps operating on a suction lift may lose their prime during the time that power is off, it is preferable to use starters with low load protection for such installations to prevent an automatic restart. This does not apply, of course, if the pumps are automatically primed or if some protection device is incorporated so the pump cannot run unless it is primed.

MAINTENANCE

Because of the wide variation in pump types, sizes, designs, and materials of construction, these comments on maintenance are restricted to those types of pumps most commonly encountered. The manufacturer's instruction books must be carefully studied before any attempt is made to service a particular pump.

Daily Observation of Pump Operation When operators are on constant duty, hourly and daily inspections should be made and any irregularities in the operation of a pump should be recorded and reported immediately. This applies particularly to changes in sound of a running pump, abrupt changes in bearing temperatures, and seal chamber leakage. A check of pressure gages and of flowmeters, if installed, and vibration should be made routinely during the day. If recording instruments are provided, a daily check should be made to determine whether the current capacity, pressure, power consumption or vibration level indicates that further inspection is required. If these readings are taken electronically, trending charts should be produced to allow observation of changes as a function of time. Certain trends may allow for scheduled outages to address deterioration of specific performance values.

Semiannual Inspection The following should be done at least every 6 months:

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1. For pumps equipped with shaft packing, the free movement of stuffing box glands should be checked, gland bolts should be cleaned and lubricated, and the packing should be inspected to determine whether it requires replacement.
2. The pump and driver alignment should be checked and corrected if necessary.
3. Housings for oil-lubricated bearings should be drained, flushed, and refilled with fresh oil.
4. Grease-lubricated bearings should be checked to see that they contain the correct amount of grease and that it is still of suitable consistency.

Annual Inspection A very thorough inspection should be performed once a year. In addition to the semiannual procedure, the following items should be considered:

1. Vibration trends should be reviewed. If the pump is trending toward unacceptable vibration levels,
   a. The bearings should be removed, cleaned, and examined for flaws and wear.
   b. The bearing housings should be carefully cleaned.
   c. Rolling element bearings should be examined for scratches and wear.
   d. Immediately after cleaning, rolling element bearings that are considered acceptable for reinstallation should be coated with oil or grease. *Note:* If there is any sign of damage, or if the bearings were damaged during removal, they should be replaced with new bearings of the correct size and type per the manufacturer's instruction book.
   e. The assembled rotor—or major rotor components if the rotor is not assembled of shrink-fit components—should be checked for balance prior to reassembly in the pump.
2. For pumps equipped with shaft packing, the packing should be removed and the shaft sleeves—or shaft, if no sleeves are used—should be examined for wear.
3. For pumps equipped with mechanical seals, if the seals were indicating signs of leaking, they should be removed and returned to the seal manufacturer for inspection, possible bench testing, and refurbishment.
4. When coupling halves are disconnected for an alignment check, the vertical shaft movement of a pump with sleeve (journal) bearings should be checked at both ends with packing or seals removed. Any movement exceeding 150% of the original design clearance should be investigated to determine the cause. Endplay allowed by the bearings should also be checked. If it exceeds that recommended by the manufacturer, the cause should be determined and corrected.
5. All auxiliary piping, such as drains, sealing water piping, and cooling water piping, should be checked and flushed, as necessary. Auxiliary coolers should also be flushed and cleaned.
6. Pump equipped with stuffing boxes should be repacked, and the pump and driver should be realigned and reconnected.
7. All instruments and flow-metering devices should be recalibrated, whenever feasible, and—whenever possible—the pump should be tested to determine whether proper performance is being obtained. If internal repairs are made, the pump should again be tested after completion of the repairs.

Complete Overhaul It is difficult to make general rules about the frequency of complete pump overhauls as it depends on the pump service, the pump construction and materials, the liquid handled, and the economic evaluation of overhaul costs versus the cost of power losses resulting from increased clearances or of unscheduled downtime. Some pumps on very severe service may need a complete overhaul monthly, whereas other applications require overhauls only every 2 to 4 years or even less frequently.

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A pump should not be opened for inspection unless either factual or circumstantial evidence indicates that overhaul is necessary. Factual evidence implies that the pump performance has fallen off significantly or that the noise or driver load indicates trouble. Circumstantial evidence refers to past experience with the pump in question or with similar equipment on similar service.

In order to ensure rapid restoration to service in the event of an unexpected overhaul, an adequate store of spare parts should be maintained at all times.

The relative complexity of the repairs, the facilities available at the site, and many other factors enter into the decision whether the necessary repairs will be carried out at the installation site or at the pump manufacturer's plant.

Spare and Repair Parts The severity of the service in which the pump is used will determine, to a great extent, the minimum number of spare parts that should be carried in stock at the installation site. Unless prior experience is available, the pump manufacturer should be consulted on this subject. As an insurance against delays, spare parts should be purchased when the pump is purchased. Depending on the contemplated method of overhaul, certain replacement parts may have to be supplied either oversized or undersized instead of the size used in the original unit.

API Standard 610, 8th edition, shows recommended spare parts as a function of the number of identical pumps installed at a site. It also lists parts usually associated with start-up and with normal maintenance. A form based on the recommendations of API Standard 610 is shown in Table 1.

When ordering spare parts after a pump has been in service, the manufacturer should always be given the pump serial number and size (stamped on the nameplate). This information is essential in identifying the pump exactly and in furnishing repair parts of correct size and material.

Records of Inspections and Repairs The working schedule of the semiannual and annual inspections should be entered into a log that tracks individual pump maintenance history. This log should then contain a complete record of all the items requiring attention. This log, usually electronic, should also contain comments and observations on the conditions of the parts to be repaired or replaced, on the rate and appearance of wear, and on the repair methods followed. In many cases, it is advisable to photograph badly worn parts before they are repaired.

In all cases, complete records of the cost of maintenance and repairs should be kept for each pump, together with a record of its operating hours. A study of these records will generally reveal whether a change in materials or even construction may be the most economical course of action to improve pump performance, reliability, and life.

Diagnosis of Pump Problems Pump operating problems may be either hydraulic or mechanical. In the first category, a pump may fail to deliver liquid, it may deliver an insufficient capacity or develop insufficient pressure, or it may lose its prime after starting. In the second category, it may consume excessive power, or symptoms of mechanical difficulties may develop at the seal chambers or at the bearings, or vibration, noise, or breakage of some pump parts may occur.

There is a definite interdependence between some difficulties of both categories. For example, increased wear at the running clearances must be classified as a mechanical trouble, but it will result in a reduction of the net pump capacity—a hydraulic symptom—without necessarily causing a mechanical breakdown or even excessive vibration. As a result, it is most useful to classify symptoms and causes separately and to list for each symptom a schedule of potential contributory causes. Such a diagnostic analysis is presented in Tables 2 through 5. Additionally, the following parts of this handbook should also be referred to for assistance in diagnosing pump hydraulic and mechanical problems and possible solutions: Sections 2.1.4, 3.4, 5.1, 5.2, and Chap. 10.

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TABLE 1 Recommended spare parts

Part	See Note	Spares Recommended: Start-up			Normal Maintenance			
Number of identical pumps		1–3	4–6	7+	1–3	4–6	7–9	10+
Cartridge	(2) (5)				1	1	1	1
Element	(2) (6)				1	1	1	1
Rotor	(3) (7)				1	1	1	1
Case	(1)							1
Head (case cover and seal chamber)								1
Bearing bracket	(1)							1
Shaft (w/key)					1	1	2	N/3
Impeller					1	1	2	N/3
Wear rings (set)	(8)	1	1	1	1	1	2	N/3
Bearings complete (rolling element, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearings complete (rolling element, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearings complete (hydrodynamic, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing pads only (hydrodynamic, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing complete (hydrodynamic, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing pads only (hydrodynamic, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Mechanical seal/packing	(4) (8) (9)	1	2	N/3	1	2	N/3	N/3
Shaft sleeve	(8)	1	2	N/3	1	2	N/3	N/3
Gaskets, shims, O-rings (set)	(8)	1	2	N/3	1	2	N/3	N/3
Add for vertical pump								
Bowls							N/3	N/3
Spiders (set)				1	1	1	N/3	N/3
Bearings, bushings (set)		1	1	2	1	1	N/3	N/3
Add for high speed integral gear								
Gear box			1	1	1	1	1	N/3
Diffuser and cover		1	1	1	1	1	1	N/3
Splined shaft		1	1	1	1	1	1	N/3
Gear box housing					1	1	1	N/3
Oil pump, internal			1	1	1	1	1	N/3
Oil pump, external			1	1	1	1	1	N/3
Oil filter		1	2	N/3	1	2	3	N/3

Notes:
N = Number of identical pumps
(1) Horizontal pumps only
(2) Vital service pumps are generally unspared, partially spared, or multistage. When a vital machine is down, production loss or violation of environmental permits results.
(3) Essential service pumps are required for operation and have an installed spare. A production loss will occur only if main and spare fail simultaneously.
(4) Cartridge type mechanical seals shall include sleeve and gland.
(5) Cartridge consists of assembled element plus discharge head, seal(s), and bearing housing(s).
(6) Element consists of assembled rotor plus stationary hydraulic parts (diffuser(s) or volute(s)).
(7) Rotor consists of all rotating parts attached to the shaft.
(8) Normal wear parts
(9) Per pump set

Source: (American Petroleum Institute, 2004)

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TABLE 2A Check chart for centrifugal pump problems

Symptoms	Possible cause of trouble (each number is defined in Table 2B)
1. Pump does not deliver liquid	1, 2, 3, 5, 10, 12, 13, 14, 16, 21, 22, 25, 30, 32, 38, 40
2. Insufficient capacity delivered	2, 3, 4, 5, 6, 7, 7a, 10, 11, 12, 13, 14, 15, 16, 17, 18, 21, 22, 23, 24, 25, 31, 32, 40, 41, 44, 63, 64
3. Insufficient pressure developed	4, 6, 7, 7a, 10, 11, 12, 13, 14, 15, 16, 18, 21, 22, 23, 24, 25, 34, 39, 40, 41, 63, 64
4. Pump loses prime after starting	2, 4, 6, 7, 7a, 8, 9, 10, 11
5. Pump requires excessive power	20, 22, 23, 24, 26, 32, 33, 34, 35, 39, 40, 41, 44, 45, 61, 69, 70, 71
6. Pump vibrates or is noisy at all flows	2, 16, 37, 43, 44, 45, 46, 47, 48, 49, 50, 51, 52, 53, 54, 55, 56, 57, 58, 59, 60, 61, 67, 78, 79, 80, 81, 82, 83, 84, 85
7. Pump vibrates or is noisy at low flows	2, 3, 17, 19, 27, 28, 29, 35, 38, 77
8. Pump vibrates or is noisy at high flows	2, 3, 10, 11, 12, 13, 14, 15, 16, 17, 18, 33, 34, 41
9. Shaft oscillates axially	17, 18, 19, 27, 29, 35, 38
10. Impeller vanes are eroded on visible side	3, 12, 13, 14, 15, 17, 41
11. Impeller vanes are eroded on invisible side	12, 17, 19, 29
12. Impeller vanes are eroded at discharge near center	37
13. Impeller vanes are eroded at discharge near shrouds or at shroud/vane fillets	27, 29
14. Impeller shrouds bowed out or fractured	27, 29
15. Pump overheats and seizes	1, 3, 12, 28, 29, 38, 42, 43, 45, 50, 51, 52, 53, 54, 55, 57, 58, 59, 60, 61, 62, 77, 78, 82
16. Internal parts are corroded prematurely	66
17. Internal clearances wear too rapidly	3, 28, 29, 45, 50, 51, 52, 53, 54, 55, 57, 59, 61, 62, 66, 77
18. Axially-split casing is cut through wire-drawing	63, 64, 65
19. Internal stationary joints are cut through wire-drawing	53, 63, 64, 65
20. Packed box leaks excessively or packing has short life	8, 9, 45, 55, 57, 68, 69, 70, 71, 72, 73, 74
21. Packed box: sleeve scored	8, 9
22. Mechanical seal leaks excessively	45, 54, 55, 57, 58, 62, 75, 76
23. Mechanical seal: damaged faces, sleeve, bellows	45, 54, 55, 57, 58, 62, 75, 76
24. Bearings have short life	3, 29, 41, 42, 45, 50, 51, 54, 55, 58, 77, 78, 79, 80, 81, 82, 83, 84, 85
25. Coupling fails	45, 50, 51, 54, 67

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TABLE 2B Possible causes of problems

Suction Problems

1. Pump not primed
2. Pump suction pipe not completely filled with liquid
3. Insufficient available NPSH
4. Excessive amount of air or gas in liquid
5. Air pocket in suction line
6. Air leaks into suction line
7. Air leaks into pump through stuffing boxes or through mechanical seal

7a. Air in source of sealing liquid

8. Water seal pipe plugged
9. Seal cage improperly mounted in stuffing box
10. Inlet of suction pipe insufficiently submerged
11. Vortex formation at suction
12. Pump operated with closed or partially closed suction valve
13. Clogged suction strainer
14. Obstruction in suction line
15. Excessive friction losses in suction line
16. Clogged impeller
17. Suction elbow in plane parallel to the shaft (for double-suction pumps)
18. Two elbows in suction piping at 90° to each other, creating swirl and prerotation
19. Selection of pump with too high a suction specific speed

Other Hydraulic Problems

20. Speed of pump too high
21. Speed of pump too low
22. Wrong direction of rotation
23. Reverse mounting of double-suction impeller
24. Uncalibrated instruments
25. Impeller diameter smaller than specified
26. Impeller diameter larger than specified
27. Impeller selection with abnormally high head coefficient
28. Running the pump against a closed discharge valve without opening a by-pass
29. Operating pump below recommended minimum flow
30. Static head higher than shut-off head

Other Hydraulic Problems (continued)

31. Friction losses in discharge higher than calculated
32. Total head of system higher than design of pump
33. Total head of system lower than design of pump
34. Running pump at too high a flow (for low specific speed pumps)
35. Running pump at too low a flow (for high specific speed pumps)
36. Leak of stuck check valve
37. Too close a gap between impeller vanes and volute tongue or diffuser vanes
38. Parallel operation of pumps unsuitable for the purpose
39. Specific gravity of liquid differs from design conditions
40. Viscosity of liquid differs from design conditions
41. Excessive wear at internal running clearances
42. Obstruction in balancing device leak-off line
43. Transients at suction source (imbalance between pressure at surface of liquid and vapor pressure at suction flange)

Mechanical Problems—general

44. Foreign matter in impellers
45. Misalignment
46. Foundation insufficiently rigid
47. Loose foundation bolts
48. Loose pump or motor bolts
49. Inadequate grouting of baseplate
50. Excessive piping forces and moments on pump nozzles
51. Improperly mounted expansion joints
52. Starting the pump without proper warm-up
53. Mounting surfaces of internal fits (at wearing rings, impellers, shaft sleeves, shaft nuts, bearing housings, and so on) not perpendicular to shaft axis
54. Bent shaft
55. Rotor out of balance
56. Parts loose on the shaft
57. Shaft running off-center because of worn bearings

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TABLE 2B Possible causes of problems (*Continued*)

Mechanical Problems—general (continued)

58. Pump running at or near critical speed
59. Too long a shaft span or too small a shaft diameter
60. Resonance between operating speed and natural frequency of foundation, baseplate, or piping
61. Rotating part rubbing on stationary part
62. Incursion of hard solid particles into running clearances
63. Improper casing gasket material
64. Inadequate installation of gasket
65. Inadequate tightening of casing bolts
66. Pump materials not suitable for liquid handled
67. Certain couplings lack lubrication

Mechanical Problems—sealing area

68. Shaft or shaft sleeves worn or scored at packing
69. Incorrect type of packing for operating conditions
70. Packing improperly installed
71. Gland too tight, prevents flow of liquid to lubricate packing
72. Excessive clearance at bottom of stuffing box allows packing to be forced into pump interior

Mechanical Problems—sealing area (continued)

73. Dirt or grit in sealing liquid
74. Failure to provide adequate cooling liquid to water-cooled stuffing boxes
75. Incorrect type of mechanical seal for prevailing conditions
76. Mechanical seal improperly installed

Mechanical Problems—bearings

77. Excessive radial thrust in single-volute pumps
78. Excessive axial thrust caused by excessive wear at internal clearances or, if used, failure or excessive wear of balancing drive
79. Wrong grade of grease or oil
80. Excessive grease or oil in rolling element bearing housings
81. Lack of lubrication
82. Improper installation of rolling element bearings such as damage during installation, incorrect assembly of stacked bearings, use of unmatched bearings as a pair, and so on
83. Dirt getting into bearings
84. Moisture contaminating lubricant
85. Excessive cooling of water-cooled bearings

TABLE 2C Diagnosis from appearance of stuffing box packing in centrifugal pumps

Symptom	Cause
Wear on one or two rings next to packing gland; other rings OK	Improper packing installation
Wear on O.D. of packing rings	Packing rings rotating with shaft sleeve or leakage between rings and I.D. of box. Wrong packing size or incorrectly cut rings
Charring or glazing of inner circumference of rings	Excessive heating. Insufficient leakage to lubricate packing or unsuitable packing
I.D. of rings excessively increased or heavily worn on part of inner circumference	Rotation eccentric

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TABLE 2D Vibration symptoms and causes in centrifugal pumps

Vibration frequency	Cause
Several times pump RPM	Bad rolling element bearings
Twice pump RPM	Loose parts on rotor, axial misalignment of coupling, influence of twin-volute when gap is insufficient
Running Speed	Imbalance of rotor, clogged impeller, coupling misaligned or loose
Running speed times number of impeller vanes	Vane passing syndrome—insufficient gap between impeller vanes and collector vanes. This is also sometimes seen during operating with suction recirculation.
One-half running speed	Oil whirl in bearing
Random low frequency	Internal circulation in impeller or cavitation
Random high frequency	Usually resonance
Subsynchronous frequency at 70% to 90% of running speed	Hydraulic excitation of resonance

TABLE 3 Check chart for rotary pump problems

Symptom	Possible cause of trouble (each number is defined in the list below)
Pump fails to discharge	1, 2, 3, 4, 5, 6, 8, 9, 16
Pump is noisy	6, 10, 11, 17, 18, 19
Pump wears rapidly	11, 12, 13, 20, 24
Pump not up to capacity	3, 5, 6, 7, 9, 16, 21, 22
Pump starts, then loses suction	1, 2, 6, 7, 10
Pump takes excessive power	14, 15, 17, 20, 23

Suction problems

1. Pump not properly primed
2. Suction pipe not submerged
3. Strainer clogged
4. Foot valve leaking
5. Suction lift too high
6. Air leaking into suction
7. Suction pipe too small

System problems

8. Wrong direction of rotation
9. Low speed
10. Insufficient liquid supply
11. Excessive pressure
12. Grit or dirt in liquid

System problems (continued)

13. Pump runs dry
14. Viscosity higher than specified
15. Obstruction in discharge line

Mechanical troubles

16. Pump worn
17. Drive shaft bent
18. Coupling out of balance or alignment
19. Relief valve chatter
20. Pipe strain on pump casing
21. Air leak at packing or seal
22. Relief valve improperly seated
23. Packing too tight
24. Corrosion

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TABLE 4 Check chart for reciprocating pump problems

Symptom	Possible cause of problem (each number is defined in the list below)
Liquid end noise	1, 2, 7, 8, 9, 10, 14, 15, 16
Power end noise	17, 18, 19, 20
Overheated power end	10, 19, 21, 22, 23, 24
Water in crankcase	25
Oil leak from crankcase	26, 27
Rapid packing or plunger wear	11, 12, 28, 29
Pitted valves or seats	3, 11, 30
Valves hanging up	31, 32
Leak at cylinder valve hole plugs	10, 13, 33, 34
Loss of prime	1, 4, 5, 6

Suction problems

1. Insufficient suction pressure
2. Partial loss of prime
3. Cavitation
4. Lift too high
5. Leaking suction at foot valve
6. Acceleration head requirement too high

System problems

7. System shocks
8. Poorly supported piping, abrupt turns in piping, pipe too small, piping misaligned
9. Air in liquid
10. Overpressure or overspeed
11. Dirty liquid
12. Dirty environment
13. Water hammer

Mechanical problems

14. Valves broken or badly worn
15. Packing worn
16. Obstruction under valve
17. Main bearings loose
18. Bearings worn
19. Oil level low
20. Plunger loose
21. Main bearings tight
22. Ventilation inadequate
23. Belts too tight
24. Driver misaligned
25. Condensation
26. Seals worn
27. Oil level too high
28. Pump not level and rigid
29. Packing loose
30. Corrosion
31. Valve binding
32. Valve spring broken
33. Cylinder plug loose
34. O-ring seal damaged

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TABLE 5 Check chart for steam-pump problems

Symptoms	Possible cause of problem (each number is defined in the list below)
Pump does not develop rated pressure	4, 5, 7, 8
Pump loses capacity after starting	1, 2, 6
Pump vibrates	9, 10, 11, 14
Pump has short strokes	12, 13, 14
Pump operation is erratic	1, 2, 3, 6

Suction problems
1. Suction line leaks
2. Suction lift too high
3. Cavitation

System problems
4. Low steam pressure
5. High exhaust pressure
6. Entrained air or vapors in liquid

Mechanical problems
7. Worn piston rings in steam end
8. Binding piston rings in liquid end
9. Misalignment
10. Foundation not rigid
11. Piping not supported
12. Excessive steam cushioning
13. Steam valves out of adjustment
14. Liquid piston packing too tight

FURTHER READING

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C • H • A • P • T • E • R • 16

PUMP TESTING

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Since the first time a device was used to pump or lift water, pump testing of one sort or another has occurred. Each improvement in pumping devices was accepted only after being tested, which was the proof of its worthiness. As pumping equipment has become more refined, so has the art of pump testing, both in the shop or laboratory and in the field. For very large pumps, model testing is being used to develop the optimum refinement in prototype design.

Every pump, regardless of size or classification, should be tested in some way before final acceptance by the purchaser. If not, the user does not have any way of knowing that all requirements have been fulfilled. What tests to run and what methods to use depend on the ultimate purpose of the tests, which normally have one of two objectives:

1. To check improvement in design or operation
2. To determine if contractual commitments have been met, thus making possible the comparison of specified, predicted, and actual performance

In most cases, the manufacturer supplies a test report and certifies the characteristics of the pump being furnished. Even these can be given a cursory check by the customer from time to time to give a record of performance or an indication of the need for replacement or overhaul. If at all possible, the pump should be tested as installed, with repeat tests from time to time to check operation.

The main object of this chapter is to present a set of procedures and rules for conducting, computing, and reporting on tests of pumping units and for obtaining the head, capacity, power, efficiency, and suction requirements of a pump.

CLASSIFICATION OF TESTS

Pump tests should be classified as follows:

- *Shop tests* are also called laboratory, manufacturer's, or factory acceptance tests. They are conducted in the pump manufacturer's plant under geometrically similar, ideal, and controlled conditions and are usually assumed to be the most accurate tests.

- *Field tests* are made with the pumping unit installed in its exact environment and operating under existing field or ultimate conditions. The accuracy and reliability of field testing depend on the instrumentation used, installation, and advance planning during the design stages of the installation. By mutual agreement, field tests can be used as acceptance tests.

- *Index tests* are a form of field testing usually made to serve as a standard of comparison for wear, changing conditions, or overhaul evaluation. Index tests should always be run using the same procedures, instruments, and personnel where possible, and a very accurate record and log of events should be kept to give as complete and comparable a history of the results as possible.

- *Model tests* precede the design of the prototype and are usually quite accurate. They supplement or complement field tests of the prototype for which the model was made. The role of the model test must be clearly established as early in the design as possible, preferably in the specification or invitation to bid. Model tests may be used when very large units are involved, when the performances of several models must be compared, and when an advance indication of prototype design is required.

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DEFINITIONS, SYMBOLS, AND UNITS

For a detailed discussion of letter symbols, definitions, description of terms, and table of letter symbols in general use, the user is referred to ASME Power Test Code[1] and to the Technical Data section of this handbook.

Standard Units Used in Pump Testing The following definitions and quantities from the Hydraulic Institute standards[2] are used throughout the industry in pump testing.

VOLUME The standard units of volume are the U.S. gallon and the cubic foot (cubic meter). The standard U.S. gallon contains 231.0 in^3 (0.00379 m^3), and 1 ft^3 = 7.4805 gal (0.028 m^3). Rate of flow is expressed in gallons per minute (cubic meters per hour), cubic feet per second, or million gallons per 24-h day. The specific weight (mass) w of pure water at 68°F (20°C), at sea level and 40° latitude, is 62.315 lb/ft^3 (0.998 kg/liter). For other temperatures or locations, proper specific weight corrections should be made. See Table 1 and the appropriate ASME power test codes.

TABLE 1 Specific weight of water in air

	Temperature, °F (°C)							
Latitude	32 (0)	40 (4.4)	50 (10)	60 (16)	70 (21)	80 (27)	90 (32)	100 (38)
				At sea level				
0°	62.1741	62.1823	62.1654	62.1227	62.0578	61.9729	61.8701	61.7514
10	62.1840	62.1921	62.1753	62.1325	62.0677	61.9828	61.8800	61.7612
20	62.2125	62.2206	62.2038	62.1610	62.0961	62.0112	61.9083	61.7895
30	62.2562	62.2643	62.2475	62.2046	62.1397	62.0547	61.9518	61.8328
40	62.3098	62.3179	62.3011	62.2582	62.1932	62.1082	62.0051	61.8861
50	62.3670	62.3751	62.3582	62.3153	62.2503	62.1652	62.0620	61.9429
60	62.4208	62.4289	62.4120	62.3691	62.3040	62.2188	62.1156	61.9963
70	62.4647	62.4728	62.4559	62.4130	62.3478	62.2626	62.1593	62.0399
				At 6000 ft				
0°	62.1508	62.1589	62.1421	62.0993	62.0345	61.9497	61.8469	61.7282
10	62.1607	62.1688	62.1520	62.1092	62.0444	61.9595	61.8568	61.7380
20	62.1891	62.1972	62.1804	62.1377	62.0728	61.9879	61.8851	61.7663
30	62.2328	62.2409	62.2241	62.1813	62.1164	62.0315	61.9286	61.8097
40	62.2864	62.2946	62.2777	62.2349	62.1699	62.0849	61.9819	61.8630
50	62.3436	62.3517	62.3349	62.2920	62.2270	62.1419	62.0388	61.9198
60	62.3974	62.4055	62.3887	62.3458	62.2807	62.1955	62.0924	61.9732
70	62.4413	62.4495	62.4326	62.3896	62.3245	62.2393	62.1361	61.0168

Note: All values are in pounds per cubic foot, 1 lb/ft^3 = 16 kg/m^3; 1 ft = 0.048 m.
Density of water from Source 1, p. 296
Gravity formula from Source 4, p. 488
$G_\phi = 980.616\,(1 - 0.0026373 \cos 2\phi + 0.0000059 \cos^2 2\phi)$
Altitude correction = 0.0003086 × altitude in meters.
Standard gravity = 980.665 cm/s^2
Density of air from Source 2:
Density = $0.001225(1 - 0.0065H \mid 288.16)^{4.2561}$
where H is in meters
Sources:

1. *Smithsonian Physical Tables,* 9th rev. ed.
2. National Advisory Committee for Aeronautics, TN 3182.
3. American Society of Mechanical Engineers, PTC 2-1971.
4. *Smithsonian Meteorological Tables,* 6th rev. ed.

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HEAD The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{head, ft} = \text{lb/in}^2 \times \frac{144}{w}$$

in SI units $$\text{Head, m} = \text{kPa} \times \frac{0.102}{w}$$

where w = specific weight of liquid being pumped under pumping conditions, lb/ft^3 (kg/l)

All pressure readings must be converted to feet (meters) of the liquid being pumped, referenced to a datum elevation, which is defined as follows. For a horizontal shaft unit, the datum elevation is the centerline of the pump shaft (Figure 1). For vertical-shaft single-suction pumps, it is the entrance eye to the first-stage impeller (Figure 2). For vertical-shaft double-suction pumps, it is the impeller-discharge horizontal centerline (Figure 3).

VELOCITY HEAD The velocity head h_v is computed from the average velocity V, obtained by dividing the flow by the pipe cross-sectional area. Velocity head is determined at the point of gage connection and is expressed by the formula

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

where V = velocity in the pipe, ft/s (m/s)

g = acceleration due to gravity = 32.17 ft/s^2 (9.81 m/s^2) (Table 2)

FLOODED SUCTION Flooded suction implies that the liquid must flow from an atmospherically vented source to the pump without the average or minimum pressure at the pump datum dropping below atmospheric pressure with the pump operating at specified capacity.

TOTAL SUCTION LIFT Suction lift exists where the total suction head is below atmospheric pressure. Total suction lift h_s is the reading of a liquid manometer or pressure gage at the

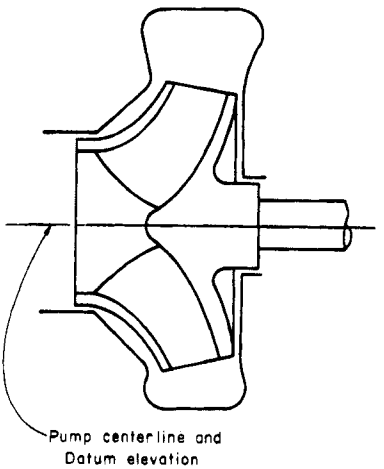


FIGURE 1 Horizontal pump

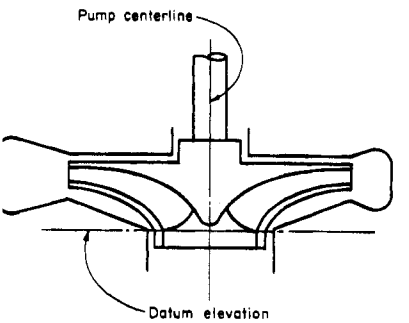


FIGURE 2 Vertical single-suction pump

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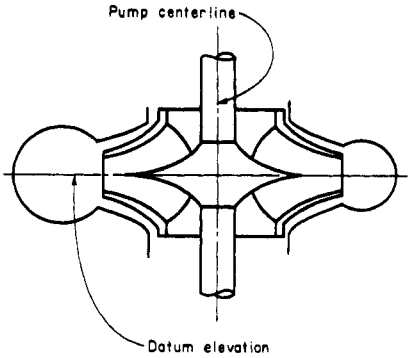


FIGURE 3 Vertical double-suction pump

TABLE 2 Variation of acceleration of gravity with latitude and altitude

	Altitude above mean sea level, ft						
Latitude	0	2000	4000	6000	8000	10,000	12,000
0°	32.0878	32.0816	32.0754	32.0693	32.0631	32.0569	32.0508
10	32.0929	32.0867	32.0805	32.0744	32.0682	32.0620	32.0558
20	32.1076	32.1014	32.0952	32.0890	32.0829	32.0767	32.0705
30	32.1301	32.1239	32.1177	32.1115	32.1054	32.0992	32.0930
40	32.1577	32.1515	32.1454	32.1392	32.1330	32.1269	32.1207
50	32.1872	32.1810	32.1748	32.1687	32.1625	32.1563	32.1501
60	32.2149	32.2087	32.2026	32.1964	32.1902	32.1841	32.1779
70	32.2375	32.2314	32.2252	32.2190	32.2129	32.2067	32.2005

Note: All values are in feet per second per second; 1 ft/s^2 = 0.3048 m/s^2; 1 ft = 0.3048 m

Gravity = $980.616 (1 - 0.0026373 \cos 2\phi + 0.0000059 \cos^2 2\phi)(1.0/30.48)$

Correction for altitude = -0.003086 ft/s^2/1,000 ft

The international standard value of gravity adopted by the International Commission on Weights and Measures is 980.665 cm/s^2 (32.17405 ft/s^2) at sea level and approximately latitude 45°.

Sample computation

Given:

Altitude = 12,000 ft (3657.60 m)

Latitude = 70°

Water temperature = 40°F (4.4°C)

Altitude correction for gravity:

$0.0003086 \times 3657.60 = 1.128785$ cm/s^2

Gravity corrected for latitude and altitude:

Gravity = $980.616 (1 - 0.0026373 \cos 140° - 0.0000059 \cos^2 140°) - 1.128735 = 981.471784$ cm/s^2

Density corrected for gravity:

Density = $(0.9999988 \times 981.471784/980.6650 \times 1.000028) = 1.0007930$ g/cm^3

Correcting for buoyancy of air:

Density = $1.0007930 - 0.0008491 = 0.9999438$ g/cm^3

Density in USCS units:

Density = $0.9999438 \times 62.4279606 = 62.4244543$ lb/ft^3

Sources:

1. "Smithsonian Physical Tables," 9th rev. ed.

2. American Society of Mechanical Engineers, PTC 2—1971.

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suction nozzle of the pump, converted to feet (meters) of liquid, and referred to datum minus the velocity head at the point of gage attachment.

TOTAL SUCTION HEAD Suction head exists when the total suction head h_s is above atmospheric pressure. Total suction head is the reading of a gage at the suction of the pump converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL DISCHARGE HEAD Total discharge head h_d is the reading of a pressure gage at the discharge of the pump, converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL HEAD Total head H is the measure of the work increase per pound (kilogram) of liquid, imparted to the liquid by the pump, and is therefore the algebraic difference between the total discharge head and the total suction head. Total head, as determined on test where suction lift exists, is the sum of the total discharge head and total suction lift. Where positive suction head exists, the total head is the total discharge head minus the total suction head.

NET POSITIVE SUCTION HEAD The net positive suction head (NPSH) h_{sv} is the total suction head in feet (meters) of liquid absolute determined at the suction nozzle and referred to datum less the vapor pressure of the liquid in feet (meters) absolute.

DRIVER INPUT The driver input ehp is the input to the driver expressed in horsepower (kilowatts). Usually this is electric input horsepower (kilowatts).

PUMP INPUT Pump input bhp is the power delivered to the pump shaft and is designated as brake horsepower (brake kilowatts).

LIQUID OR WATER POWER Water power whp (wkW) is the useful work delivered by the pump and is usually expressed by the formula

in USCS units $$\text{whp} = \frac{(\text{sp. gr.})QH}{3960}$$

in SI units $$\text{wkW} = 9.8QH(\text{sp. gr.})$$

where sp. gr. = specific gravity of liquid, referred to water at 68°F (20°C)
Q = flow rate, gpm(m^3/h)
H = total head, ft (m)

This formula is derived on p. 16.36.

EFFICIENCY Pump efficiency E_p is the ratio of the power delivered by the pump to the power supplied to the pump shaft; that is, the ratio of the liquid power (also known as water power) to the brake power expressed in percent:

In USCS units $$E_p = \frac{\text{whp}}{\text{bhp}} \times 100$$

In SI units $$E_p = \frac{\text{wkW}}{\text{kW}} \times 100$$

Overall efficiency E_o is the ratio of the power delivered by the pump to the power supplied to the input side of the pump driver; that is, the ratio of the output power to the input power to the driver.

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In USCS units $$E_o = \frac{\text{whp}}{\text{ehp}} \times 100$$

In SI units $$E_o = \frac{\text{wkW}}{\text{kW}} \times 100$$

PRIME MOVER RATINGS The prime movers for driving pumps are rated according to established standards. For example, for electric motor drivers, see the standards of the latest edition of the *National Electrical Manufacturers Association*.

ACCURACY AND TOLERANCES

Accuracy The accuracy to which tests can be made depends on the instruments used, their proper installation, the skill of the test engineer, and the shop tests for the simulation of field conditions. The test engineer must have sufficient knowledge of the characteristics and limitations of the test instruments to obtain maximum accuracy when using them, along with a thorough understanding of the pumps, prime movers, controls, and installation peculiarities to interpret the results. For shop testing, the acceptable deviations and fluctuations of the instrumented test readings are given in Table 1.11 of the *ASME Performance Test Code for Centrifugal Pumps* (ASME PTC 8.2-1990). These deviations are not to be misconstrued as tolerances, which must be spelled out in the specifications. The limits of accuracy of pump test measuring devices for use in field testing are shown in Figure 4. Using these limits, the combined accuracy of the efficiency is the square root of the quantity [square of the head accuracy plus square of the flow rate accuracy plus square of the power input accuracy]:

$$A_c = \sqrt{(\pm H^2) + (\pm Q^2) + (\pm \text{ehp}^2)} \text{ (percent)}$$

Pump speed and voltage are not required for efficiency computations, and so the values for these are not included in this formula.

Instrumentation All instruments should be calibrated before the tests, and all calibration and correction data or curves should be prepared in advance. Where required, a certified calibration curve showing the calibration of the instrument, including any procedures for establishing a coefficient, should be furnished before testing begins. The specifications should be explicit in regard to a waiving of these calibration requirements. After testing, all instruments should be recalibrated. Any differences between before and after calibration values must be resolved either by retest or by acceptable variations being spelled out in the specifications.

Tolerances The tolerances in pump performance permitted are usually given in the specifications. The user can and should make these requirements known before the order for pumping apparatus is placed. The test tolerances permitted by the many organizations are commonly used.

TEST REQUISITES

Operating Conditions The primary factors affecting the operation of a pump are the inlet (suction), outlet (discharge or total head), and speed. The secondary factors are physical and climatic variables, such as the temperature, viscosity, specific weight, and turbidity of the liquid being pumped and the elevation of the pumping system above sea level. In some installations, it is impossible to measure discharge or even head accurately. In these instances, good shop tests are essential. It follows then that, in order for the shop test to predict the field performance of a pump, the field operating, installation, and suction conditions should be simulated.

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Quantity to be measured	Measuring device	Calibrated limit of accuracy plus or minus, %
Capacity	Venturi meter	1/2
	Nozzle	1
	Pitot tube	1 1/2
	Orifice	1 1/4
	Disk	2
	Magnetic flowmeter	1/2
	Piston	1/4
	Volume or weight-tank	1
	Propeller meter	4
Head	Electric sounding line	1/4
	Air line	1/2
	Instrument grade pressure transducer:	
	1/4–1/2 scale	1 %
	1/2–3/4 scale	1/2 %
	3/4–1 scale	1/4 %
	Liquid manometer, 3- to 5-in (75- to 127-mm) deflections	3/4
	Liquid manometer, over 5-in (127-mm) deflections	1/2
	Bourdon gage—5 in (127-mm) min dial:	
	1/4–1/2 full scale	1
	1/2–3/4 full scale	3/4
	Over 3/4 scale	1/2
Power input	Watt-hour meter and stopwatch	1 1/2
	Portable recording wattmeter	1 1/2
	Test precision wattmeter:	
	1/4–1/2 scale	3/4
	1/2–3/4 scale	1/2
	Over 3/4 scale	1/4
	Clamp on ammeter	4
Speed	Revolution counter and stopwatch	1 1/4
	Handheld tachometer	1 1/4
	Stroboscope	1 1/2
	Automatic counter and stopwatch	1/2
	Digital tachometer	1/8
Voltage	Test meter	
	1/4–1/2	1
	1/2–3/4	%
	3/4-full	%
	Rectifier voltmeter	5

Note: Instruments with accuracy significantly different than shown are available. Refer to the specific instruction manual.

Source: ANSI B-58.1 (AWWAE 101–61).

FIGURE 4 Typical Limits of accuracy of pump test measuring devices in field use

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The inlet passages are critical, and the sump where used on the suction lift must be duplicated as closely as possible. During the shop tests, no total suction head less than specified should be permitted, nor should the suction head exceed the specified amount in cases where cavitation or possibly operating "in the break" could occur.

For field installations above sea level, the difference in elevation between the shop test site and the field installation must be taken into account by reducing to the barometric pressure at the specified elevation. This is especially true if a suction lift or negative suction head is involved. Standard tables of barometric pressures are available for use in computing the data, and the tables to be used should be acceptable to all interested parties.

Flow Control The most common means of adjusting pump flow during a test is by using a valve to provide the required change in system head. The valve should be selected so that the minimum and maximum head conditions may be attained. Since valve cavitation is a common problem in tests using butterfly, gate, or globe valves, special noncavitating valves may be needed for high power tests, or tests requiring a significant head breakdown.

Cavitation Tests Cavitation tests should be run if required by the specifications (provided such tests are needed and have not been previously conducted on similar pumps and certified by the manufacturer) or if needed to assume a successful pump installation.

The suction requirements that must be met by the pump are usually defined by the cavitation coefficient σ. Plant σ is defined as NPSHA (net positive suction head available) divided by total pump head per stage:

$$\sigma = \frac{NPSHA}{H}$$

Three typical arrangements for determining the cavitation characteristics of pumps are illustrated in Figure 5. In Figure 5A, the suction is taken from a sump with a constant-level surface. The liquid is drawn first through a valve (throttle) and then through a section of pipe containing screens and straightening devices, such as vanes and baffles. This setup will dissipate the turbulence created by the suction valve and will also straighten the flow so the pump suction flow will be relatively free from undue turbulence. In Figure 5B, the suction is taken from a relatively deep sump or well in which the water surface can be varied over a fairly large range to provide the designed variation in suction lift. In Figure 5C, the suction is taken from a closed vessel in a closed loop in which the pressure level can be varied by a gas pressure over the liquid, by temperature of the liquid, or by a combination of these.

By using one of these cavitation test arrangements, the critical value of σ (that is, the value at which cavitation will begin) can be found by one of the following two methods:

1. Constant speed and capacity vary the suction lift. Run the pump at constant speed and capacity with the suction lift varied to produce cavitation conditions. Plots of the head, efficiency, and power input against σ as shown in Figure 6.

   When the values of σ are held high, the values of head, efficiency, and power should remain relatively constant. As σ is reduced, a point is reached when the curves break from the normal, indicating an unstable condition. This breakaway condition may and usually does impair the operation of the pump. The extent of impairment depends on the size, specific speed, and service of the pump and on the characteristics of the pumped fluid. A variation of this method is to plot results using capacities both greater and less than normal, as shown in Figure 7.
2. Constant speed and suction lift vary the capacity. Run the pump at constant speed and suction lift and vary the capacity. For a given suction lift, the pumping head is plotted against capacity. A series of such tests will result in a family of curves, as shown in Figure 8.

Where the plotted curve for any suction condition breaks away from the normal, cavitation has occurred. The value of σ may be calculated at the breakaway points by dividing NPSHA by total head H at the point under consideration.

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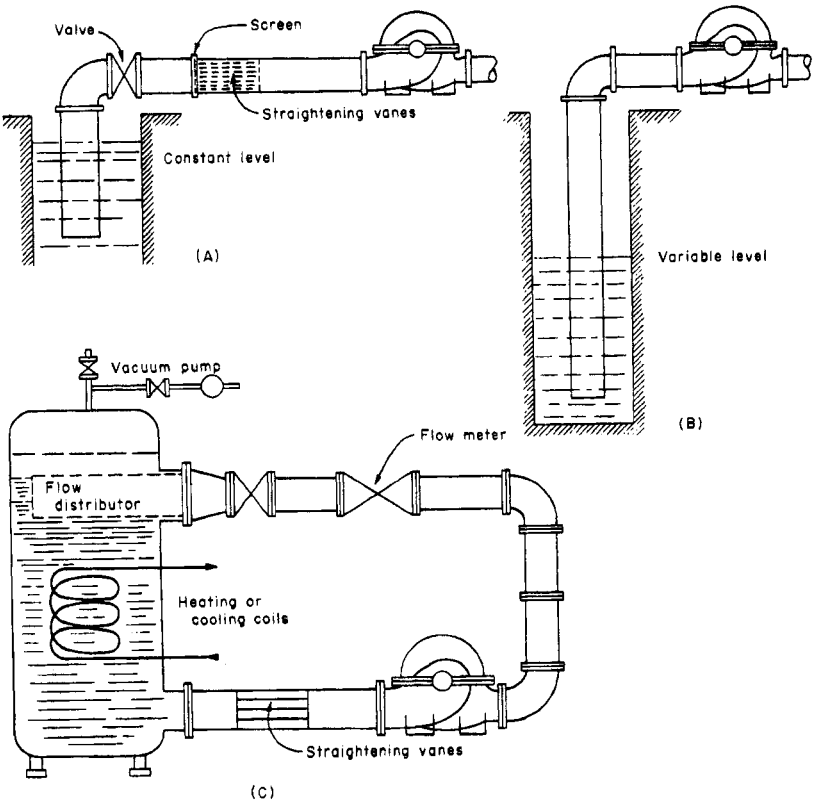


FIGURE 5A through C Typical arrangement for determining cavitation characteristics (Hydraulic Institute Standards, 13th Edition—out of print[2])

PUMP TESTING

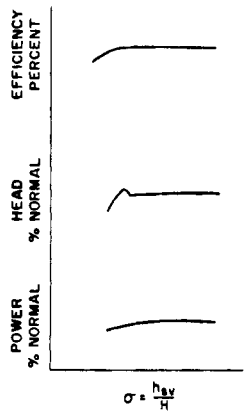


FIGURE 6 Functions of sigma at constant capacity and speed; suction pressure varied

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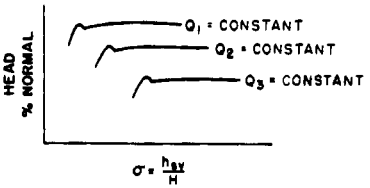


FIGURE 7 Sigma capacities above and below normal; suction pressure varied

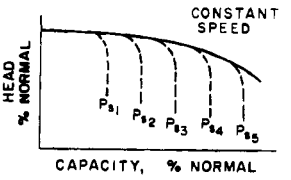


FIGURE 8 Typical cavitation curves at constant speed and suction pressure

Testing at Zero Flow In some cases, pumps can be damaged by testing at zero flow (shut-off). As pump power increases, and casing water volume decreases, this problem becomes acute. Prior to test, all parties should agree on the minimum flow to be demonstrated.

TEST PROCEDURE

Agreements The specifications and contract should be very clear on any special points that must be covered by the pump testing. All interested parties shall be represented and given equal rights in regard to test date, setup, conditions, instrumentation, calibration of instruments, examination of pump and test setup, and accuracy of results and computations. Any controversial points or methods not provided for in the specifications should be resolved to the satisfaction of all interested parties before testing is begun.

In some special instances and by agreement between parties, an independent test expert may be engaged to take over full responsibility of the test stand and apparatus. This person will make all decisions after consultation with the interested parties and should be used only where an impasse is reached.

Normally, the manufacturer will establish the time and date for the pump tests. In some cases, the specifications will cover such items as duration, limiting date, and notification time. Reasonable notice must be given to all official witnesses or representatives. A 30-day notice is preferred, and 1 week should be considered a bare minimum.

Any time limitation regarding correction of mechanical defects or equipment malfunctions that arise during testing must be resolved by mutual agreement.

Observers and Witnesses Representatives from each party to the contract shall have equal opportunity to attend the testing. Where more than one representative from one of the parties is present, their function as observers, official witnesses, or representatives must be made clear before the start of testing. The number of representatives present from any one party shall not be a deciding factor when disagreements are being resolved. Any comments or constructive criticism from the observers and witnesses should be duly considered.

Inspection and Preliminary Operation All interested parties shall make as complete an inspection as possible before, during, and after the test to determine compliance with specifications and correct connection of all instrumentation. The following items should be inspected before or during the test:

- Impeller and casing passages
- Pump and driver alignment
- Piezometer openings
- Electrical connections
- Lubricating devices and system
- Wearing ring and other clearances
- Stuffing box or mechanical seal adjustment and leakage

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This is not a complete list of items and should be taken as only a guide for the interested parties. Instruments installed on the pump to obtain the necessary test information shall not affect the pump operation or performance. If a question arises as to the effect an instrument has on the operation, it should be resolved by all parties. Where necessary, comparative preliminary tests can be conducted with the disputed equipment removed and then reinstalled. The dimensions at the piezometer connections on both the suction and discharge sides must be accurately determined to permit accurate determination of the velocity head correction.

On satisfactory completion of the preliminary inspection, the pump may be started. The pump and all instrumentation should then be checked for proper operation, scale readings, or evidence of malfunction. When all equipment and apparatus are functioning properly, a preliminary test run should be made. If possible, this run should be made at or near the rated condition. The correct procedures for observing and recording the data should be established during this run. Also, the time it takes to obtain steady test conditions is determined for use in the pump test runs. In test systems with relatively long circuits (like pipelines), the time needed to reach steady-state conditions increases dramatically. The acceptable deviations and fluctuations for test readings are given in the section named "Accuracy."

Suggested Test Procedure

TIME AND DATE After tests have been decided upon, it is to the best interests of all parties to conduct them with the least delay. The contract normally will not give a time or date for the test, but will specify a completion date for submission or approval of a final test report.

PERSONNEL Pump testing, regardless of classification, should be carried out by personnel specially trained in the operation of the test equipment used. Representatives from each party to the contract shall be given equal opportunity to witness the test or tests and shall also have equal voice in commenting on the conduct of the tests or on compliance with specifications or code requirements where applicable.

SCHEDULE A schedule should be agreed upon by all parties in advance of the test. The schedule should be as complete a program as possible and give some particulars on the range of test heads, discharge rates, and speed to be used. This schedule should be flexible and subject to change, especially after the preliminary runs have been made.

INSPECTION The pump and test setup should be thoroughly inspected both before and after the tests. Special attention should be given to the hydraulic passages and pressure taps near the suction and discharge sections. Also, the discharge measuring device should be inspected.

CALIBRATION OF INSTRUMENTS While the setup is being inspected, all measuring devices should be calibrated and adjusted as explained under the section named "Instrumentation."

PRELIMINARY TESTS After it has been determined that the test setup complies with the installation and specification requirements and that the instrumentation is properly installed, the pump is started. A sufficient number of preliminary test runs should be made to check the functioning of the test stand and all control and measuring devices. These preliminary tests also give the test personnel and representatives an opportunity to check and adjust the entire setup and serve as a basis for agreements on accuracy and compliance. Each test point is held until satisfactory stable conditions exist. The acceptable fluctuations in test readings is covered under "Accuracy." It is suggested that preliminary computations be made, plotted, and analyzed prior to actual test runs. (*Note:* Most pump manufacturers will conduct a complete preliminary shop test to be sure of specifications and contractual compliance before inviting the purchaser's representatives to witness the official test.)

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OFFICIAL TEST RUNS The test points for the official test runs must be sufficient in number to establish the head-discharge curve over the specified range and to provide any other data needed to compute or plot the information required by the specifications. It is suggested that one test run be as near the rated condition as possible and that at least three runs be in the specified operating range of the pump.

LOGGING OF EVENTS In most instances, the official pump test is conducted by only two, three, or four test engineers, and no record other than test data is needed. In more complicated or important tests, it is sometimes very desirable to assign someone the task of recording and logging events as they occur. This is most important if reruns are required. Complete records, including any notes or comments on inspection and calibration, shall be kept of all data, readings, observations, and information relevant to the test. A suggested form for shop and field tests is shown in Figure 28.

PRELIMINARY COMPUTATIONS Sufficient preliminary computations should be made to determine whether all specification requirements have been met and whether reruns will be necessary.

RERUNS When the preliminary computations indicate that reruns are necessary, they should be run immediately or as soon as possible after the official test runs and with the same personnel, instruments, and devices. Sometimes mechanical or electrical faults will necessitate a rerun. If after correction of these faults several reruns indicate a change, a complete retesting may be required. Any official representatives shall have the right to ask for a rerun or be shown to their satisfaction that a rerun is not required.

COMPUTATIONS, PLOTTING REPORTS These topics are discussed later in the text.

TEST MEASUREMENTS

DISCHARGE The choice of which method of discharge measurement to use should be made by an agreement between all parties concerned. Some test codes and procedures in regular use permit or even recommend certain methods for model or shop testing, but restrict their use in field or index testing. Some methods are more adaptable to the site conditions than others, and so the test engineers and interested parties should be completely familiar with the several methods applicable before settling on the one to be used.

The most commonly used methods of discharge measurement are those that use quantity meters and those that use rate-of-flow meters. Both meters are usually classified as liquid meters, and their functions are listed in Table 3.

QUANTITY METERS The term *quantity* is here used to designate those meters in which the fluid passes through the primary element in successive and more or less completely isolated

TABLE 3 Liquid meters and their functions

Quantity meters	Rate-of-flow meters
Weighing meters	Differential pressure meters
Weighing tank	Venturi
Tilting trap	Nozzle
Volumetric meters	Orifice plate
Tank	Pitot tube
Reciprocating piston	Head area meters
Rotary piston	Weir
Nutating disk	Flume
	Current meters
	Velocity meters (magnetic, ultrasonic)

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quantities, either weights or volumes, by alternately filling and emptying containers of known capacities. The secondary element of a quantity meter consists of a counter with suitably graduated dials for registering the total quantity that has passed through. Quantity meters are classified into two groups: weighing meters and volumetric meters.

WEIGHING METERS There are two types of weighing meters: weighing tank and tilting trap. In the tilting trap meter, the equilibrium of a container is upset by a rise of the center of gravity as the container is filled. Weighing tank meters employ a container suspended from a counterbalanced scale beam. The weighing tank and the tilting trap are affected slightly by the temperature of the liquid but not enough to cause concern in normal testing.

VOLUMETRIC METERS Volumetric meters measure volumes instead of weights. There are four types: tank, reciprocating piston, rotary piston, and nutating disk.

Tank meters are a very elementary form of meter of limited commercial importance. As the name implies, they consist of one or more tanks that are alternately filled and emptied. The height to which they are filled can be regulated manually or automatically. In some cases, the rising liquid operates a float that controls the inflow and outflow; in others, it starts a siphon. Occasionally, some tank meters have been erroneously classified as weighing meters.

Reciprocating piston meters use one or more members that have a reciprocating motion and operate in one or more fixed chambers. The quantity per cycle can be adjusted either by varying the magnitude of movement of one or more of the reciprocating members or by varying the relation between the primary and secondary elements.

Rotary (or oscillating) piston meters have one or more vanes that serve as pistons or movable partitions for separating the fluid segments. These vanes may be either flat or cylindrical and rotate within a cylindrical metering chamber. The axis of rotation of the vanes may or may not coincide with that of the chamber. The portion of the chamber in which the fluid is measured usually includes about 270°. In the remaining 90°, the vanes are returned to the starting position for closing off another segment of fluid. This may be accomplished by the use of an idle rotor or gear, a cam, or a radial partition. The vanes must make almost a wiping contact with the walls of the measuring chamber. The rotation of the vanes operates the counter.

Nutating disk meters have the disk mounted in a circular chamber with a conical roof and either a flat or conical floor. When in operation, the motion of the disk is such that the shaft on which it is mounted generates a cone with the apex down. However, the disk does not rotate about its own axis; this is prevented by a radial slot that fits about a radial partition extending in from the chamber sidewall nearly to the center. The peculiar motion of the disk is called *nutating*. The inlet and outlet openings are in the sidewall of the chamber on either side of the partition. These meters are usually adjusted by changing the relation between the primary and secondary elements.

RATE-OF-FLOW METERS The term *rate of flow* is applied to all meters through which the fluid passes not in isolated quantities but in a continuous stream. The movement of this fluid stream through the primary element is directly or indirectly utilized to actuate the secondary element. The quantity of flow per unit time is derived from the interactions of the stream and the primary element, using physical laws supplemented by empirical relations.

In rate-of-flow meters, the functioning of the primary element depends upon some property of the fluid other than, or in addition to, volume or mass. This property may be kinetic energy (head meters), inertia (gate meters), specific heat (thermal meters), or the like. The secondary element senses a change in the property concerned and usually embodies some device that draws the necessary inferences automatically, so the observer can read the rate of flow from a dial or chart. In some cases, the secondary element records pressures, such as static and differential, from which the rate of flow and time-quantity flow must be computed. In others, the secondary element not only indicates the rate of flow but also integrates it with respect to time and records the total quantity that has passed through the meter. In some cases, the indications of the secondary element are transmitted to a point some distance from the primary element.

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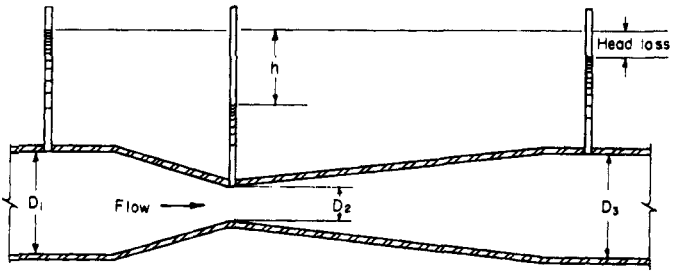


FIGURE 9 Diagram of venturi meter

DIFFERENTIAL PRESSURE METERS With this group of meters the stream of fluid creates a pressure difference as it flows through the primary element. The magnitude of this pressure difference depends upon the speed and density of the fluid and features of the primary element.[3]

Flow in a pipeline, or closed pressure conduit, can be measured by a wide variety of methods, and the choice of method for a particular installation will depend upon prevailing conditions. The accuracy of flow measurements in pressure conduits made with properly selected, installed, and maintained measuring equipment, such as venturi meters, flow nozzles, orifice meters, and pitot tubes, can be very high.

The venturi meter (Figure 9) is perhaps the most accurate flow measuring device that can be used in a water supply system. It contains no moving parts, requires very little maintenance, and causes very little head loss. Venturi meters operate on the principle that flow in a closed conduit system is faster through areas of small cross section (D_2 in Figure 9) than through areas of large cross action (D_1). The total energy in the flow, consisting primarily of velocity head and pressure head, is essentially the same at D_1 and D_2. Thus the pressure must decrease in the constricted throat D_2, where the velocity is higher, and conversely must increase at D_1 upstream from the throat, where the velocity is lower. This reduction in pressure from the meter entrance to the meter throat is directly related to the rate of flow through the meter and is the measurement used to determine flow rate.

The coefficient of discharge for the venturi meter ranges from 0.935 for small-throat velocities and diameters to 0.99 for large-throat velocities and diameters. Equations for the venturi meter are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = 3.118\,\frac{CA_2'\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for meter
A_2 = area of throat section, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = differential head of liquid between meter inlet and throat, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A'_2 = area of throat section, in²

Flow nozzles operate on the same basic principle as venturi meters. In effect, the flow nozzle is a venturi meter that has been simplified and shortened by omission of the long

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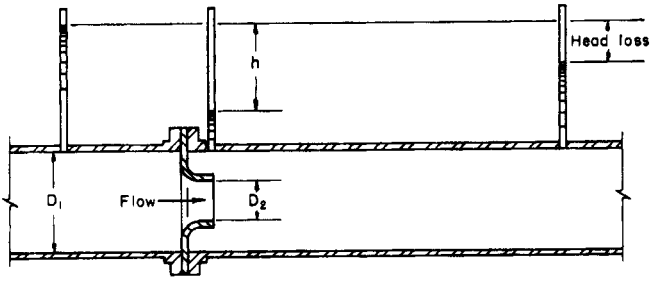


FIGURE 10 Diagram of flow nozzle

diffuser on the outlet side (Figure 10). The streamlined entrance of the nozzle provides a straight cylindrical jet without contraction, so the coefficient of discharge is almost the same as that for the venturi meter. In the flow nozzle, the jet is allowed to expand of its own accord, and the high degree of turbulence created downstream from the nozzle causes a greater loss of head than occurs in the venturi meter, where the diffuser suppresses turbulence. The relationship of flow rate to head and flow nozzle dimensions is

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = \frac{3.118CA_2'\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for nozzle
A_2 = area of nozzle throat, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = head at or across the nozzle, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A_2' = area of nozzle throat, in²

In an orifice meter, a thin plate orifice inserted across a pipeline is used for measuring flow in much the same manner as a flow nozzle (Figure 11). The upstream pressure connection is often located about one pipe diameter upstream from the orifice plate. The

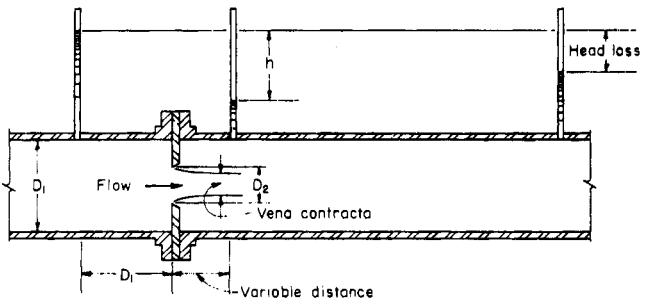


FIGURE 11 Diagram of orifice meter

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pressure of the jet ranges from a minimum at the *vena contracta* (the smallest cross section of the jet) to a maximum at about four or five conduit diameters downstream from the orifice plate. The downstream pressure connection (the center connection in Figure 11) is usually made at the *vena contracta* to obtain a large pressure differential across the orifice.

The pressure tap openings should be free from burrs and flush with the interior surfaces of the pipe. Equations for the orifice plate are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = \frac{3.118CA_2'\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for orifice plate
A_2 = area of orifice, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = head across the orifice plate, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A'_2 = area of orifice, in²

The principal disadvantage of orifice meters, compared with venturi meters and flow nozzles, is their greater loss of head. On the other hand, they are inexpensive and capable of producing accurate flow measurements.

It should be noted that the relationship of flow rate to head and dimensions of the metering section is identical for the venturi meter, flow nozzle, and orifice meter except that the coefficients of discharge vary.

Where it is impossible to employ one of the methods previously described, the pitot tube is often used. A pitot tube in its simplest form consists of a tube with a right-angle bend which, when partly immersed with the bent part under water and pointed directly into the flow, indicates flow velocity by the distance water rises in the vertical stem. The pitot tube makes use of the difference between the static and total pressures at a single point.

The height of rise h of the water column above the water surface, expressed in feet (meters) and tenths of feet (millimeters), equals the velocity head $v^2/2g$. The velocity of flow v in feet (meters) per second may thus be determined from the relation $v = \sqrt{2gh}$.

In a more complete form known as the pitot static tube, the instrument consists of two separate, essentially parallel parts, one for indicating the sum of the pressure and velocity heads (total head) and the other for indicating only the pressure head. Manometers are commonly used to measure these heads, and the velocity head is obtained by subtracting the static head from the total head. A pressure transducer may be used instead of the manometer to measure the differential head. Oscillographic or digital recording of the electric signal from the transducer provides a continuous record of the changes in head.

The simple form of the pitot tube has little practical value for measuring discharges in open channels handling low-velocity flows because the distance the water in the manometer tube rises is difficult to measure. This limitation is overcome to a large extent by using a pressure transducer for the measurement and precise electronic equipment for the data readings.

The pitot static tube, on the other hand, works very well for this purpose if the tube is used with a differential manometer of the suction lift type (Figure 12). In this manometer, the two legs are joined at the top by a T that connects to a third line, in which a partial vacuum can be created. After the pitot tube has been bled to remove all air, water flows up

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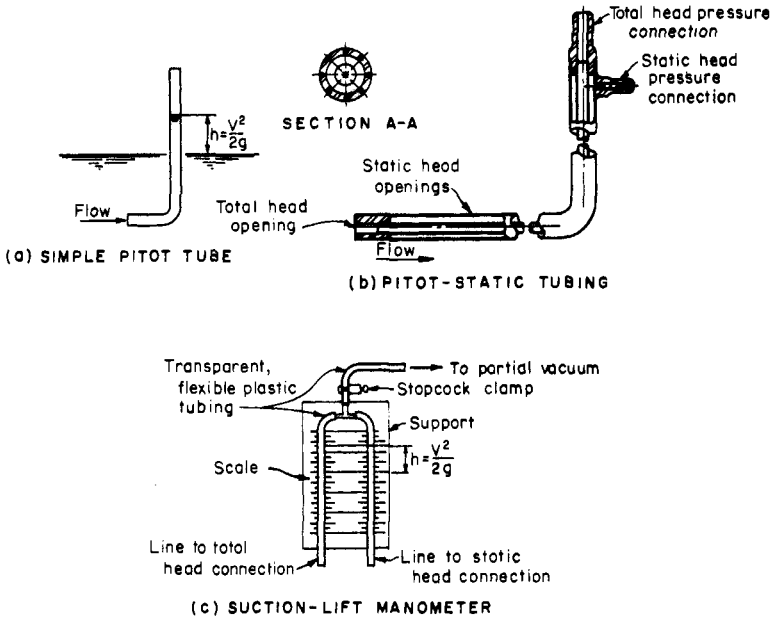


FIGURE 12A through C Pilot tubes and manometer

through it into the manometer to the height desired for easy reading. Then the stopcock or clamp on the vacuum line is closed. The partial vacuum acts equally on the two legs and does not change the differential head. The velocity head h is then the difference between the total head reading and the static head reading. If desired, a pressure transducer can also be used for the head measurement.

Pitot tubes can be used to measure relatively high velocities in canals, and it is often possible to make satisfactory discharge measurements at drops, chutes, overfall crests, or other stations where the water flows rapidly and fairly large velocity heads occur. At low velocities, values of h become quite small. The pitot tube head error for a low velocity will lead to a much larger inaccuracy in discharge computation than the same error when the velocity is high. The velocity traverse with a pitot tube may be made in the same manner as with a current meter (discussed later).

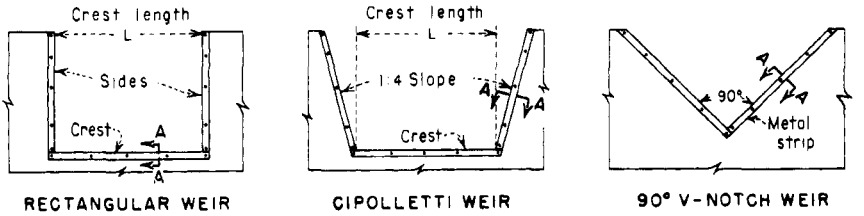
HEAD AREA METERS The instruments used to measure flow in open conduits are normally classified as head area meters, the most common of which are weirs and flumes.

A weir is an overflow structure built across an open channel. Weirs are one of the oldest, simplest, and most reliable structures for measuring the flow of water in canals and ditches. These structures can be easily inspected, and any improper operations can be quickly detected and corrected.

The discharge rates are determined by measuring the vertical distance from the crest of the overflow portion of the weir to the water surface in the pool upstream from the crest and referring to computation tables that apply to the size and shape of the weir. For standard tables to apply, the weir must have a regular shape and definite dimensions and must be placed in a bulkhead and pool of adequate size so the system performs in a standard manner.

Weirs may be termed rectangular, trapezoidal, or triangular, depending upon the shape of the opening. In rectangular and trapezoidal weirs, the bottom edge of the opening is the crest and the side edges are called *sides* or *weir ends* (Figures 13 and 14). The sheet of

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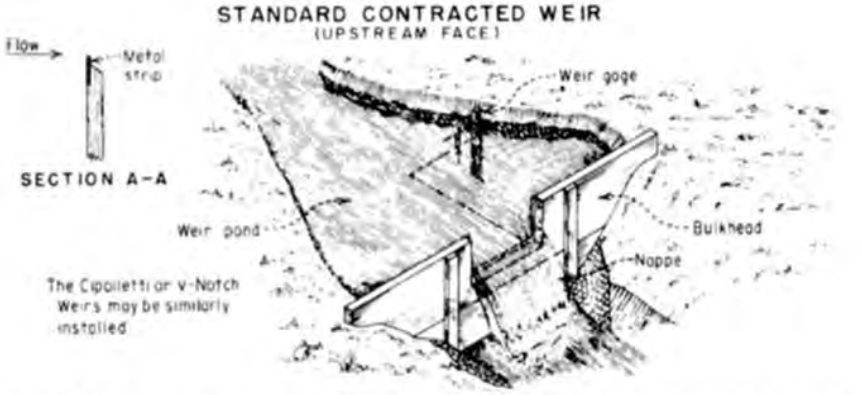


FIGURE 13 Standard contracted weirs and temporary bulkhead with contracted rectangular weir discharging at free flow

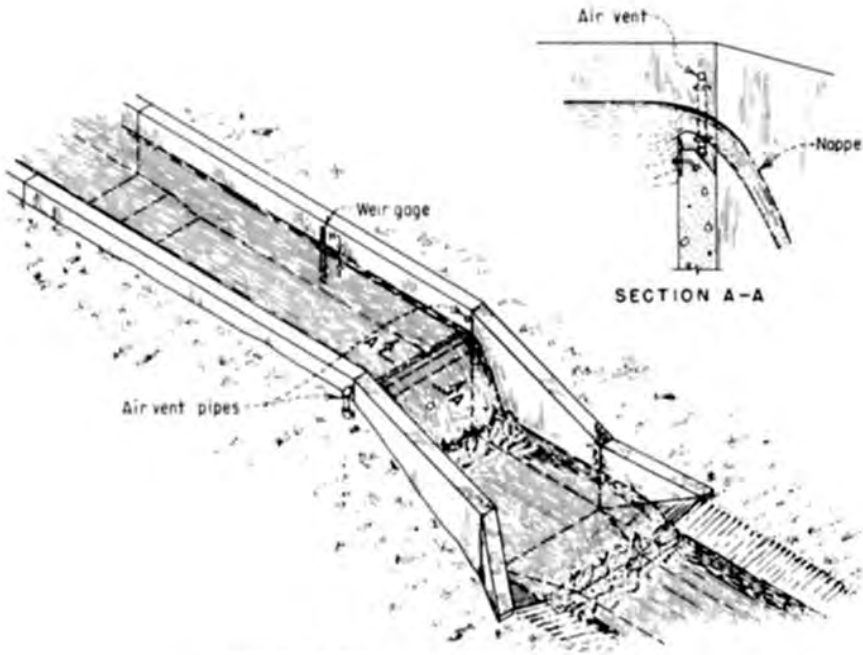


FIGURE 14 Typical suppressed weir in a flume drop

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water leaving the weir crest is called the *nappe*. In certain submerged conditions, the under-nappe airspace must be ventilated to maintain near-atmospheric pressure.

The types of weirs most commonly used to measure water are

- Sharp-crested and sharp-sided Cipolletti weirs
- Sharp-sided 90° V-notch weirs
- Sharp-crested contracted rectangular weirs
- Sharp-crested suppressed rectangular weirs

For measuring water flow, the type of weir used has characteristics that make it suitable for a particular operating condition. In general, for best accuracy, a rectangular suppressed weir or a 90° V-notch weir should be used.

The discharge in second-feet (cubic meters per second) over the crest of a contracted rectangular weir, a suppressed rectangular weir, or a Cipolletti weir is determined by the head H in feet (meters) and the crest length L in feet (meters). The discharge of the standard 90° V-notch weir is determined directly by the head on the bottom of the V notch.

As the stream passes over the weir, the top surface curves downward. This curved surface, or drawdown, extends upstream a short distance from the weir notch. The head H must be measured at a point on the water surface in the weir pond beyond the effect of the drawdown. This distance should be at least four times the maximum head on the weir, and the same gage point should be used for lesser discharges. A staff gage (Figure 15) having a graduated scele with the zero placed at the same elevation as the weir crest is usually provided for the head measurements.

Two widely used sets of formulas for computing discharge over standard contracted rectangular weirs are those of Smith[4] and Francis.[5] The formulas proposed by Smith require the use of coefficients of discharge that vary with the head of water on the weir and with the length of the weir. Consequently, the Smith formulas are somewhat inconvenient to use, although they are accurate for the ranges of coefficients usually given. For this type of weir operating under favorable conditions as prescribed in preceding paragraphs, the Francis formula when velocity of approach is neglected is

in USCS units $$Q = 3.33H^{3/2}(L - 0.2H) \tag{1}$$

in SI units $$Q = 1.837H^{3/2}(L - 0.2H)$$

and the formula when velocity of approach is included is

in USCS units $$Q' = 3.33[(H + h)^{3/2} - h^{3/2}](L - 0.2H)$$

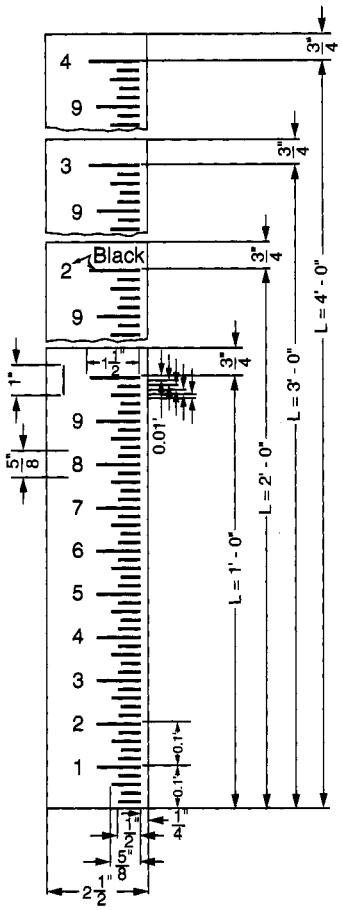
in SI units $$Q' = 1.837[(H + h)^{3/2} - h^{3/2})(L - 0.2H) \tag{2}$$

where Q = discharge neglecting velocity of approach, ft³/s (m³/s)
Q' = discharge considering velocity of approach, ft³/s (m³/s)
H = head on weir, ft (m)
L = length of weir, ft (m)
h = head due to velocity of approach ($v^2/2g$), ft (m)

Note that the Francis formulas contain constant discharge coefficients that allow computation without the use of tables. A table of $\frac{3}{2}$ powers that provides values of $H^{3/2}$, $h^{3/2}$, and $(H + h)^{3/2}$ for convenience in computing discharge with the Francis formulas may be found in hydraulic handbooks.

The principal formulas used for computing the discharge of the standard suppressed rectangular weir were also proposed by Smith and Francis. In the Smith formulas for suppressed weirs, as for contracted weirs, coefficients of discharge vary with weir head and length; therefore, these formulas are not convenient for use in computations without tables or coefficients.

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NOTES

Material of 18 gage (U.S. Standard) metal coated with substantial thickness of porcelain enamel. Face of gage is white. Numerals and graduations are black. Graduations are sharp and accurate to dimensions shown.

Length "L" represents gate limits.

Gages may be made in any length desired using similar details.

FIGURE 15 Standard weir, or staff, gage

The Francis formula for the standard suppressed rectangular weir neglecting velocity of approach in

in USCS units $$Q = 3.33LH^{3/2}$$

in SI units $$Q = 1.857LH^{3/2} \quad (3)$$

and that including velocity of approach is

in USCS units $$Q' = 3.33L[(H + h)^{3/2} - h^{3/2}]$$

in SI units $$Q' = 1.837L[(H + h)^{3/2} - h^{3/2}] \quad (4)$$

In these formulas, the letters have the same significance as in the formulas for contracted rectangular weirs. The coefficient of discharge was obtained by Francis from the same general set of experiments as those used for the contracted rectangular weir. No tests have been made to determine the applicability of these formulas to weirs less than 4 ft (1.2 m) in length.

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The Cipolletti weir is a contracted weir and must be installed as such to obtain reasonably correct and consistent discharge measurements. However, Cipolletti has compensated for the reduction in discharge due to end contractions by sloping the sides of the weir sufficiently to overcome the effect of contraction. The Cipolletti formula, in which the Francis coefficient is increased by about 1% and velocity of approach is neglected, is

in USCS units $$Q = 3.367LH^{3/2} \quad (5)$$

in SI units $$Q = 1.858LH^{3/2}$$

The formula including velocity of approach is

in USCS units $$Q' = 3.367L(H + 1.5h)^{3/2} \quad (6)$$

in SI units $$Q' = 1.858L(H + 1.5h)^{3/2}$$

where Q = discharge neglecting velocity of approach, ft³/s (m³/s)
Q' = discharge considering velocity of approach, ft³/s (m³/s)
H = head on weir, ft (m)
L = length of weir, ft (m)
h = head due to velocity of approach ($v^2/2g$), ft (m)

The accuracy of measurements obtained with Cipolletti weirs and these formulas is inherently not as great as that obtained with suppressed rectangular or V-notch weirs.[6] It is, however, acceptable where no great precision is required.

There are several well-known formulas used to compute the discharge over 90° V-notch weirs. The most commonly used in the field of irrigation are the Cone formula and the Thomson formula. The Cone formula, considered by authorities to be more reliable for small weirs and for conditions generally encountered in measuring water for open channels, is

in USCS units $$Q = 2.49H^{2.48}$$

in SI units $$Q = 1.34H^{2.48} \quad (7)$$

where Q = discharge over weir, ft³/s (m³/s)
H = head on weir, ft (m)

Ordinarily V-notch weirs are not appreciably affected by velocity of approach. If the weir is installed with complete contraction, the velocity of approach will be low.

Flumes have a measuring section that is produced by contraction of the channel sidewalls or by raising of the bottom to form a hump, or by both. The Parshall flume[7] is the most common and best known measuring flume, especially in irrigation canals. It is a specially shaped open-channel flow section that may be installed in a canal, lateral, or ditch to measure the rate of flow of water. The flume has four significant advantages: (1) it can operate with relatively small head loss, (2) it is relatively insensitive to velocity of approach, (3) it has the capability of making good measurements with no submergence, moderate submergence, or even with considerable submergence downstream, and (4) its velocity of flow is sufficiently high to virtually eliminate sediment deposition in the structure during operation.

Discharge through a Parshall flume can occur for two conditions of flow. The first, free flow, occurs when there is insufficient backwater depth to reduce the discharge rate. The second, submerged flow, occurs when the water surface downstream from the flume is far enough above the elevation of the flume crest to reduce the discharge. For free flow, only the flume head H_a at the upstream gage location is needed to determine the discharge from a standard table. The free-flow range includes some of the range that might ordinarily be considered submerged flow because Parshall flumes tolerate 50 to 80% submergence before the free-flow rate is measurably reduced. For submerged flows (when submergence is greater than 50 to 80%, depending upon flume size), both the upstream and downstream heads H_a and H_b are needed to determine the discharge (Figure 16).

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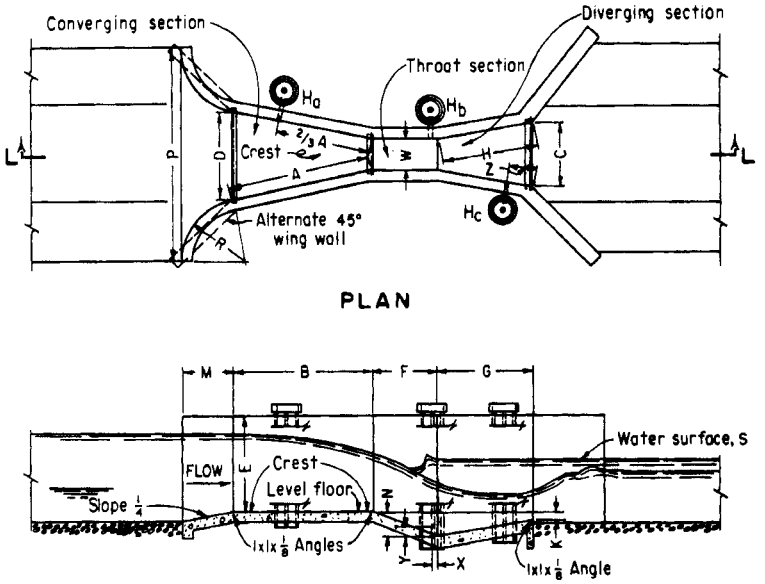


FIGURE 16A Parshall flume dimensions (sheet 1 of 2)

A distinct advantage of the Parshall flume is its ability to function as a flowmeter over a wide operating range with minimum head loss while requiring but a single head measurement for each discharge. The head loss is only about one-fourth of that needed to operate a weir having the same crest length. Another advantage is that the velocity of approach is automatically controlled if the correct size of flume is chosen and if the flume is used as it should be, that is, as an in-line structure.

Flumes are widely used because there is no easy way to alter the dimensions of flumes that have been constructed or to change the device or channel to obtain an unfair proportion of water.

The main disadvantages of Parshall flumes are (1) they cannot be used in close-coupled combination structures consisting of turnout, control, and measuring device; (2) they are usually more expensive than weirs or submerged orifices; (3) they require a solid, watertight foundation; and (4) they require accurate workmanship for satisfactory construction and performance.

Parshall flume sizes are designated by the throat width W, and sizes range from 1 in (25.4 mm) for discharges as small as 0.01 ft^3/s (2.83×10^{-2} m^3/s) up to 50 ft (15 m) for discharges as large as 3000 ft^3/s (85 m^3/s).[8] Flumes may be built of wood, concrete, galvanized sheet metal, or other desired materials. Large flumes are usually constructed on the site, but smaller flumes may be purchased as prefabricated structures to be installed in one piece. Some flumes are available as lightweight shells, which are made rigid and immobile by placing concrete outside the walls and beneath the bottom. Larger flumes are used in rivers and large canals and streams; smaller ones are used for measuring farm deliveries or for row requirements in the farmer's field.

Flumes can operate in two modes: free flow and submerged flow. In free flow, the discharge depends solely upon the width of the throat W and the depth of water H_a at the gage point in the converging section (Figures 16 and 17). Free-flow conditions in the flume are similar to those that occur at a weir or spillway crest in that water passing over the crest is not slowed by downstream conditions.

In submerged flow, other factors are operative. In most installations, when the discharge is increased above a critical value, the resistance to flow in the downstream

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	W		A		⅔ A		B		C		D		E		F		G		H		K		M		N		P		R		X		Y		Z		FREE-FLOW CAPACITY	
																																					MINIMUM	MAXIMUM
	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	SEC.-FT.	SEC.-FT.
2/	0	1 1/2	1	2 9/32	0	9 17/32	1	2	0	3 21/32	0	6 25/32	0	6 TO 9	0	3	0	8	0	8 1/8	0	3/4	–	–	0	1 1/8		–	–		0	5/16	0	1/2	0	1/8	.01	0.19
		2 1/2	1	4 5/16		10 7/8	1	4		5 5/16		8 33/32		6 TO 10		4 1/2		10		10 1/8		7/8		–		1 11/16		–	–			5/8		1		1/4	.02	.47
		3 1/2	1	6 3/8	1	1/4	1	6		7		10 31/32		1 TO 1 1/2		6	1	0	1	5/32		1	–	–		2 1/4		–	–			1		1 1/2		1/2	.03	1.13
3/	0	6	2	7/16	1	4 5/16	2	0	1	3 1/2	1	3 5/8	2	0	1	0	2	0	–		0	3	1	0	0	4 1/2	2	11 1/2	1	4	0	2	0	3	–		.05	3.9
		9	2	10 3/8	1	11 7/8	2	10	1	3	1	10 3/8	2	6	1	0	1	6	–			3	1	0		4 1/2	3	6 1/2	1	4		2		3	–		.09	8.9
	1	0	4	6	3	0	4	4 7/8	2	0	2	9 1/4	3	0	2	0	3	0	–			3	1	3		9	4	10 3/4	1	8		2		3	–		.11	16.1
	1	6	4	9	3	2	4	7 7/8	2	6	3	4 3/8	3	0	2	0	3	0	–			3	1	3		9	5	6	1	8		2		3	–		.15	24.6
	2	0	5	0	3	4	4	10 7/8	3	0	3	11 1/2	3	0	2	0	3	0	–			3	1	3		9	6	1	1	8		2		3	–		.42	33.1
3/	3	0	5	6	3	8	5	4 3/4	4	0	5	1 7/8	3	0	2	0	3	0	–			3	1	3		9	7	3 1/2	1	8		2		3	–		.61	50.4
	4	0	6	0	4	0	5	10 3/8	5	0	6	4 1/4	3	0	2	0	3	0	–			3	1	6		9	8	10 3/4	2	0		2		3	–		1.3	67.9
	5	0	6	6	4	4	6	4 1/2	6	0	7	6 3/8	3	0	2	0	3	0	–			3	1	6		9	10	11 1/4	2	0		2		3	–		1.6	85.6
	6	0	7	0	4	8	6	10 3/8	7	0	8	9	3	0	2	0	3	0	–			3	1	6		9	11	3 1/2	2	0		2		3	–		2.6	103.5
	7	0	7	6	5	0	7	4 1/4	8	0	9	11 3/8	3	0	2	0	3	0	–			3	1	6		9	12	6	2	0		2		3	–		3.0	121.4
	8	0	8	0	5	4	7	10 1/8	9	0	11	1 3/4	3	0	2	0	3	0	–			3	1	6		9	13	8 1/4	2	0		2		3	–		3.5	139.5
4/	10	0	–		6	0	14	0	12	0	15	7 1/4	4	0	3	0	6	0	–		0	6	–		1	1 1/2	–		–		0	9	1	0	–		6	200
	12	0	–		6	8	16	0	14	8	18	4 3/4	5	0	3	0	8	0	–			6	–		1	1 1/2	–		–			9	1	0	–		8	350
	15	0	–		7	8	25	0	18	4	25	0	6	0	4	0	10	0	–			9	–		1	6	–		–			9	1	0	–		8	600
	20	0	–		9	4	25	0	24	0	30	0	7	0	6	0	12	0	–		1	0	–		2	3	–		–			9	1	0	–		10	1000
	25	0	–		11	0	25	0	29	4	35	0	7	0	6	0	13	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1200
	30	0	–		12	8	26	0	34	8	40	4 3/4	7	0	6	0	14	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1500
	40	0	–		16	0	27	0	45	4	50	9 1/2	7	0	6	0	16	0	–		1	0	–		2	3	–		–			9	1	0	–		20	2000
	50	0	–		19	4	27	0	56	8	60	9 1/2	7	0	6	0	20	0	–		1	0	–		2	3	–		–			9	1	0	–		25	3000

1/ Tolerance on throat width (w) ± 1/64 inch; tolerance on other dimensions ± 1/32 inch. Sidewalls of throat must be parallel and vertical.

2/ From Colorado State University Technical Bulletin No. 61.

3/ From U.S. Department of Agriculture Soil Conservation Circular No. 843.

4/ From Colorado State University Bulletin No. 426-A.

FIGURE 16B Parshall flume dimensions (sheet 2 of 2.) (1 ft = 0.3048 m; ft³/s = 0.0253 m³/s)

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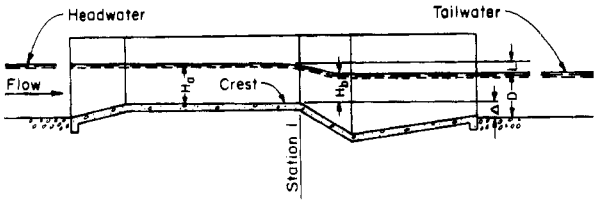


FIGURE 17 Relationships of flow depth to flume crest elevation

channel becomes sufficient to reduce the velocity, increase the flow depth, and cause a backwater effect at the Parshall flume. It might be expected that the discharge would begin to be reduced as soon as the backwater level H_b exceeds the elevation of the flume crest; however, this is not the case. Calibration tests show that the discharge is not reduced until the submergence ratio H_b/H_a (expressed in percent) exceeds the following values:

50% for flumes 1, 2, and 3 in (25, 50, and 75 mm) wide
60% for flumes 6 and 9 in (152 and 229 mm) wide
70% for flumes 1 to 8 ft (0.3 to 2.4 m) wide
80% for flumes 8 to 50 ft (2.4 to 15.2 m) wide

The discharge equations for free flow over flumes are as follows. The equation which expresses the relationship between upstream head H_a and discharge Q for widths W from 1 to 8 ft (0.3 to 2.4 m) is

in USCS units $$Q = 4WH_a^{1.522W^{0.026}}$$

in SI units $$Q = 0.371W\left(\frac{Ha}{0.305}\right)^{1.57W^{0.026}}$$

If this equation is used to compute discharges through flumes ranging from 10 to 50 ft (3 to 15.7 m) wide, the computed discharges are always larger than actual discharges. Therefore, a more accurate equation was developed for the large flumes:

In USCS units $$Q = (3.6875W + 2.5)H_a^{1.6}$$

In SI units $$Q = (2.293W + 0.474)H_a^{1.6}$$

This difference in computed discharges obtained by using the two equations for an 8-ft (2.4-m) flume is normally less than 1%; however, the difference becomes greater as the flume size increases. Because of the difficulties in regularly using these equations, discharge tables have been prepared for use with flumes 1 to 50 ft (0.3 to 15.2 m) wide.

CURRENT METERS The essential features of a conventional current meter are a wheel that rotates when immersed in flowing water and a device for determining the number of revolutions of the wheel. For open-channel flow measurement, a type generally used is the Price meter with five or six conical cups. The relationship between the velocity of the water and the number of revolutions of the wheel per unit time is determined experimentally for each instrument for various velocities. Also, the operator must have considerable skill to obtain consistent satisfactory results.

A detailed explanation of the use of current meters is contained in Chap. 5 of Reference 9.

OTHER METHODS The discharge measurement methods previously given are the ones in common use; however, a number of other methods, some newer and more sophisticated,

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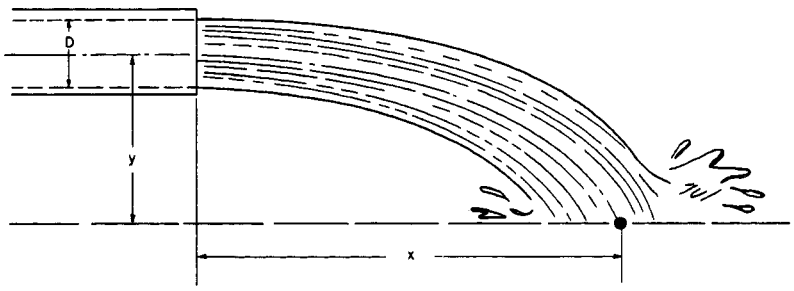


FIGURE 18 Approximating flow from a horizontal pipe

are well established. The use of these special methods is acceptable provided their limitations are recognized and all parties to the testing program are in agreement on their use. A few of these methods, not in any particular order, are

Salt velocity	Deflection meters	Radioisotope	Color dilution
Salt dilution	Propeller meters	Gates and sluices	Slope area
Color velocity	Float movement	Acoustic flowmeters	Magnetic meters

FIELD APPROXIMATING Often a field approximation of water flow from a pump discharge becomes necessary, especially if no other methods are practical or readily available. One of the accepted methods is by trajectory. The discharge from the pipe may be either vertical or horizontal, the principal difficulty being in measuring the coordinates of the flowing stream accurately. The pipes must be flowing full, and the accuracy of this method varies from 85 to 100%. Figure 18 illustrates the approximation from a horizontal pipe. This method can be further simplified by measuring to the top of the flowing stream and always measuring so y equals 12 in (300 mm) and measuring the horizontal distance X in inches (millimeters), as illustrated in Figure 19.

Figure 20 illustrates a method of measuring discharge from a vertical pipe.

Head Measurements Head is the quantity used to express the energy content of a liquid per unit weight of the liquid referred to any arbitrary datum. In terms of foot-pounds of energy per pound of liquid, all head quantities have the dimensions of feet of liquid. The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{Head, ft} = \text{lb/in}^2 \times \frac{2.31 \times 62.3}{W} = \text{lb/in}^2 \times \frac{2.31}{\text{sp. gr.}}$$

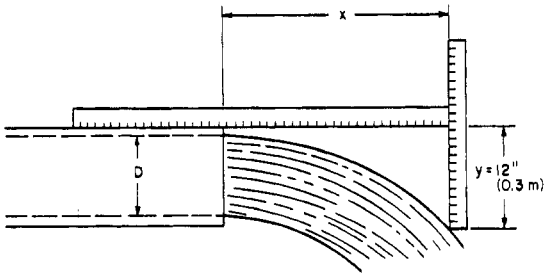
in SI units $$\text{Head, m} = \frac{0.102 \times \text{kPa}}{W}$$

where = specific weight (mass), lb/ft^3 (kg/l)

sp. gr. = specific gravity of the liquid

The following exerpt from the *Hydraulic Institute Standard* is used by the Bureau of Reclamation throughout its test program:

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CAPACITY, GPM $= 0.818\ D^2X$

CAPACITY, $m^3/h = 1.1336 \times 10^{-6} D^2X$

D = in (mm)

X = in (mm)

APPROXIMATE CAPACITY, GPM,
FOR FULL FLOWING HORIZONTAL PIPES

STD. WT. STEEL PIPE, INSIDE DIA., IN.		DISTANCE X, IN., WHEN Y = 12"										
NOMINAL	ACTUAL	12	14	16	18	20	22	24	26	28	30	32
2	2.067	42	49	56	63	70	77	84	91	98	105	112
2½	2.469	60	70	80	90	100	110	120	130	140	150	160
3	3.068	93	108	123	139	154	169	185	200	216	231	246
4	4.026	159	186	212	239	266	292	318	345	372	398	425
5	5.047	250	292	334	376	417	459	501	543	585	627	868
6	6.065	362	422	482	542	602	662	722	782	842	902	962
8	7.981	627	732	837	942	1047	1150	1255	1360	1465	1570	1675
10	10.020	980	1145	1310	1475	1635	1800	1965	2130	2290	2455	2620
12	12.000	1415	1650	1890	2125	2360	2595	2830	3065	3300	3540	3775

FIGURE 19 Simplified method for approximating flow from a horizontal pipe (1 in = 25.4 mm; 1 gpm = 0.227 m^3/h)

It is important that steady flow conditions exist at the point of instrument connection. For this reason, it is necessary that pressure or head measurement be taken on a section of pipe where the cross section is constant and straight. Five to ten diameters of straight pipe of unvarying cross section following any elbow or curved member, valve, or other obstruction, are necessary to insure steady flow conditions.

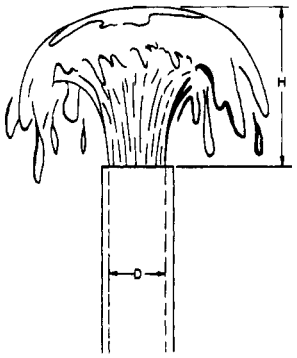
The following precautions should be taken in forming orifices for pressure-measuring instruments and for making connections. The orifice in the pipe should be flush with and normal to the wall of the water passage. The wall of the water passage should be smooth and of unvarying cross section. For a distance of at least 12 in (0.8 m) preceding the orifice, all tubercles and roughness should be removed with a file or emery cloth, if necessary. The orifice should be from $\frac{1}{8}$ to $\frac{1}{4}$ in (3–6 mm) in diameter and two diameters long.

The edges of the orifice should be provided with a suitable radius tangential to the wall of the water passage and should be free from burrs or irregularities. Two pressure tap arrangements shown in Figure 27 indicate taps in conformity with the above. Where more than one tap is required at a given measuring section, separate properly valved connections should be made. As an alternative, separate instruments should be provided.

Multiple orifices should not be connected to one instrument except on those metering devices, such as venturi meters, where proper calibrations have been made.

All leads from the orifice should be tight and as short and direct as possible. For dry-tube leads, suitable drain pots should be provided and a loop of sufficient height to keep

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Capacity, Gpm = $5.68\ KD^2\ H1/2$
Capacity, $m^3/h = 7.872 \times 10^{-5} KD^2\ H1/2$
D = I.D. of Pipe, in. (mm)
H = Vertical Height of water jets, in. (mm)
K = a constant, varying from .87 to .97 for pipes 2 to 6 in. (51 to 152 mm)
H = 6 to 24 in. (152 to 610 mm)

APPROXIMATE CAPACITY, GPM, FOR FLOW FROM VERTICAL PIPES

NOMINAL	VERTICAL HEIGHT, H, OF WATER JET, IN.										
I.D. PIPE, IN.	3	3.5	4	4.5	5	5.5	6	7	8	10	12
2	38	41	44	47	50	53	56	61	65	74	82
3	81	89	96	103	109	114	120	132	141	160	177
4	137	151	163	174	185	195	205	222	240	269	299
6	318	349	378	405	430	455	480	520	560	635	700
8	567	623	684	730	776	821	868	945	1020	1150	1270
10	950	1055	1115	1200	1280	1350	1415	1530	1640	1840	2010

FIGURE 20 Approximating flow from a vertical pipe (1 in = 25.4 mm; 1 gpm = 0.227 m^3/h)

the pumped liquid from entering the leads should be formed. For wet-tube leads, vent cocks for flushing should be provided at any high point or loop crest to assure that tubes do not become airbound. All instrument hose, piping, and fittings should be checked under pressure prior to test to assure that there are no leaks. Suitable damping devices may be used in the leads.

If these conditions cannot be satisfied at the point of measurement, it is recommended that four separate pressure taps be installed, equally spaced about the pipe, and that the pressure at that section be taken as the average of the four separate values. If the separate readings show a difference of static pressure that might affect the head beyond the contract tolerances, the installation should be corrected or an acceptable tolerance determined.

Figures 21 to 27 show suitable arrangements for various types of instruments and formulas for translating instrument readings into feet (meters) of liquid pumped, for expressing instrument head as elevation over a common datum, and for correcting these formulas for the velocity head existing in the suction and discharge pipes.

The datum is taken as the centerline of the pump for horizontal-shaft pumps and as the entrance eye of the impeller for vertical-shaft pumps.

The instruments, when practicable, are water columns or manometers for normal pressures and mercury manometers, bourdon gages, electric pressure transducers, or deadweight gage testers for high pressure. When water columns are used, care should be taken to avoid errors due to the difference between the temperature of the water in the gage and that of the water in the pump.

DEFINITIONS AND SYMBOLS The symbols used throughout this section for expressing and computing head are those used by the ASME Power Test Code[1] for pumps and the

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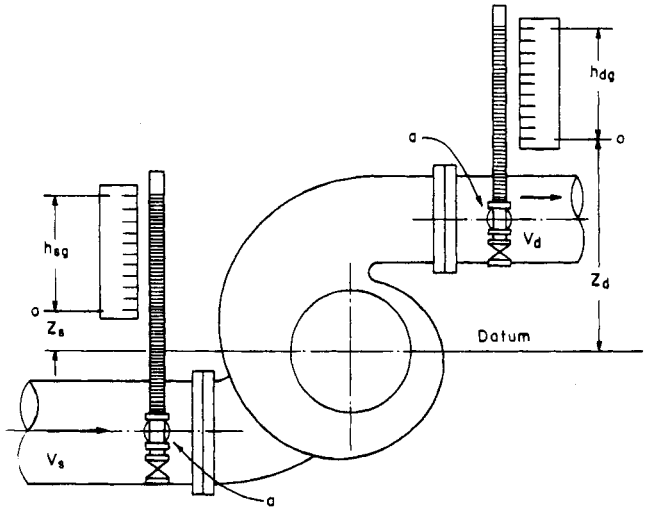


FIGURE 21 Head measurement

Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13). The following symbols apply to Figures 21 to 27 where temperature effects are negligible:

H = total head or dynamic head, ft (m) = the energy increase per pound (kilogram) of liquid imparted to the liquid by the pump and therefore the algebraic difference between total discharge head and total suction head ($H = h_d - h_s$); h_d and h_s are negative if the corresponding values at the datum elevation are below atmospheric pressure

h_{gd} = discharge gage reading, ft (m) H_2O

h_{gs} = suction gage reading, ft (m) H_2O

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

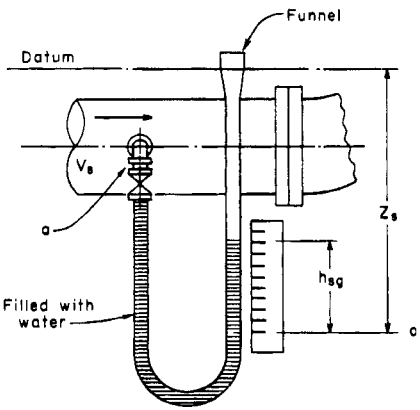


FIGURE 22 Head measurement

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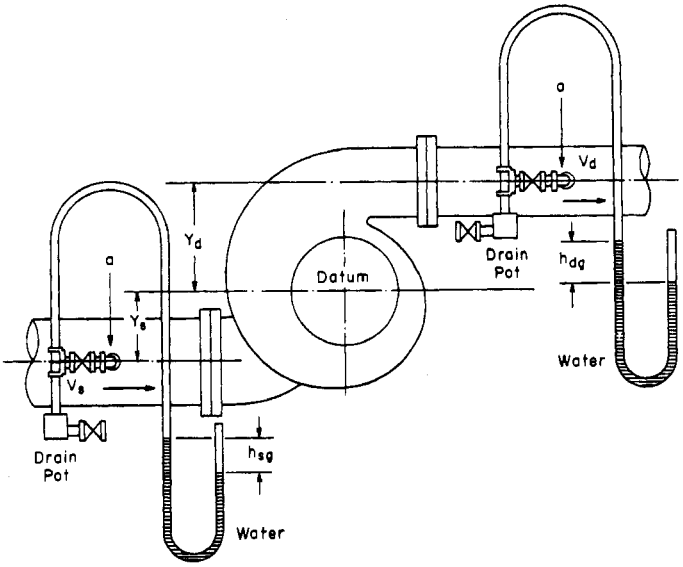


FIGURE 23 Head measurement

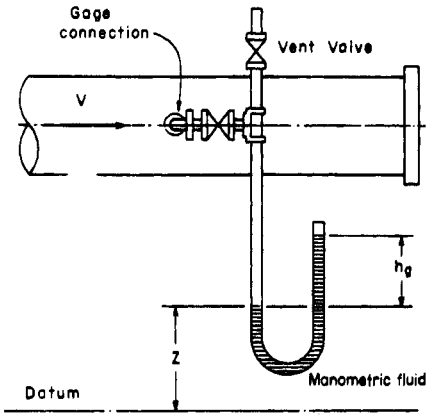


FIGURE 24 Head measurement

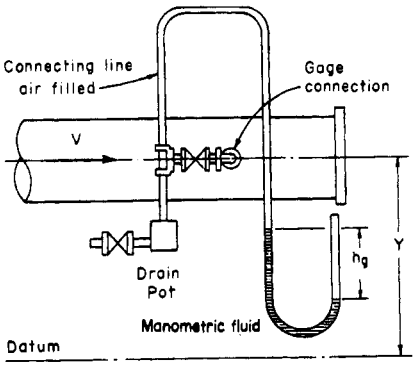


FIGURE 25 Head measurement

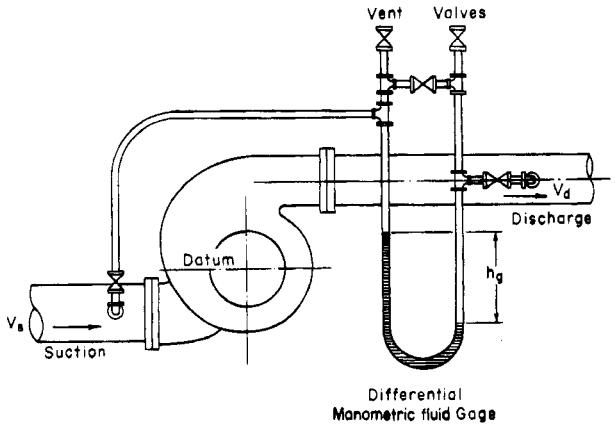


FIGURE 26 Head measurement

Both the discharge gage and the suction gage can be direct-reading water manometers, converted mercury manometers, or calibrated bourdon pressure gages.

Z_d = elevation of discharge gage, zero above datum elevation, ft (m)

Z_s = elevation of suction gage, zero above datum elevation, ft (m)

The quantities Z_d and Z_s are negative if the gage zero is below the datum elevation.

Y_d = elevation of discharge gage connection to discharge pipe above datum elevation, ft (m)

Y_s = elevation of suction gage connection to suction pipe above datum elevation, ft (m)

The quantities Y_d and Y_s are negative if the gage connection to the pipe lies below the datum elevation.

V_d = average water velocity in discharge pipe at discharge gage connection, ft/s (m/s)

V_s = average water velocity in suction pipe at suction gage connection, ft/s (m/s)

h_d = total discharge head above atmospheric pressure at datum elevation, ft (m)

h_s = total suction head above atmospheric pressure at datum elevation, ft (m)

h_{vs} = velocity head in suction pipe ($V_s^2/2g$), ft (m)

h_{vd} = velocity head in discharge pipe ($V_d^2/2g$), ft (m)

NPSHA = net positive suction head available = total suction head in feet (meters) of liquid absolute, determined at suction nozzle and referred to datum less absolute vapor pressure of the liquid in feet (meters) of liquid pumped (*NPSHA* = $h_a - H_{vpa} + h_s$)

h_{sa} = total suction head absolute ($h_a + h_s$)

H_{vpa} = vapor pressure of liquid, ft (m) abs

h_a = atmosphere pressure, ft (m) abs

MEASURING HEAD WITH WATER GAGES The following examples illustrate how to calculate the head in a centrifugal pump arrangement with gages either above or below atmospheric pressure.

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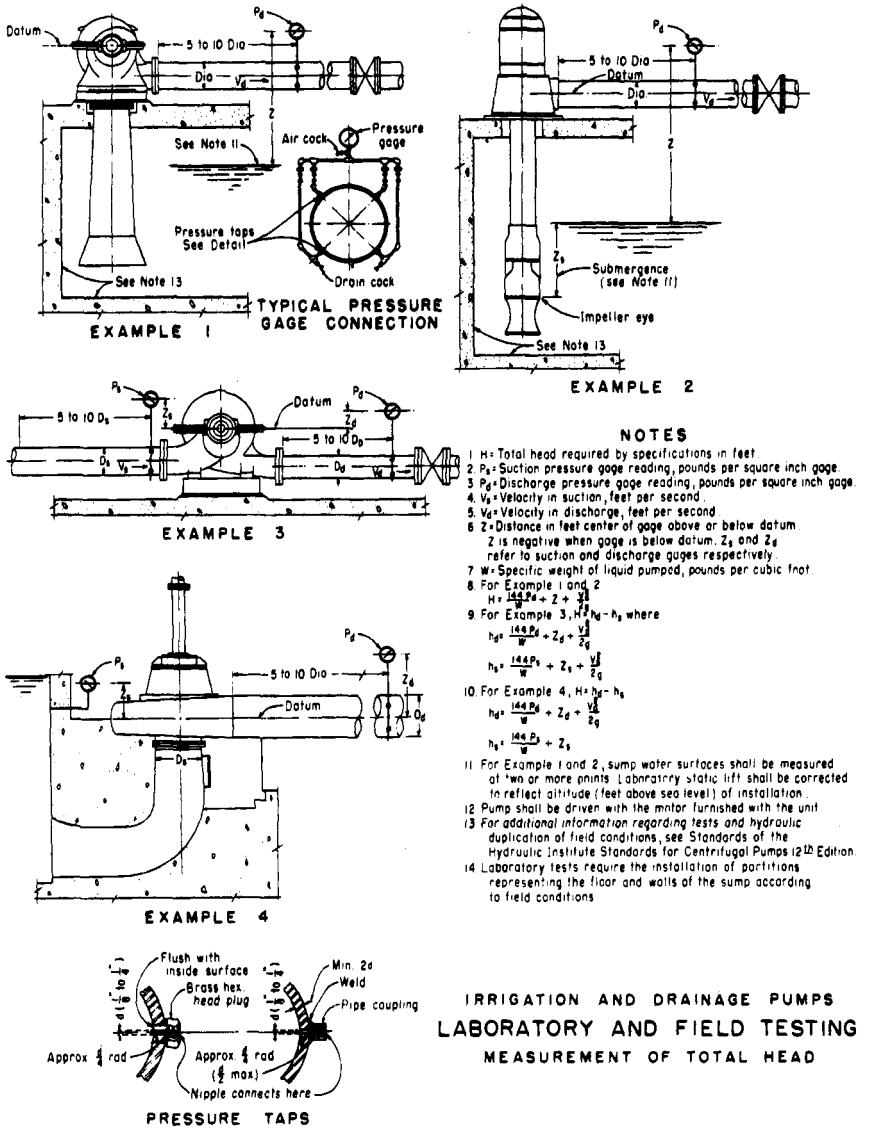


FIGURE 27 Head measurement. In SI units, $hd = [(0.102\,P_d/W) + Z_d + (V_d^2/2g)]$; $h_s = [(0.102\,P_s/W) + Z_s + (V_s^2/2g)]$. (ft × 0.3048 = m; psi × 6.894 = kPa; pcf × .016 = kg/liter)

EXAMPLE The pressure at gage connection a in Figure 21 is above atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with water. The following equations apply:

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

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$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

Note: The word *water* is used to represent the liquid being pumped. The provisions are applicable to the pumping of other liquids, provided the gages and connecting lines contain the liquid being pumped.

EXAMPLE The pressure at gage connection a in Figure 22 is below atmospheric pressure. The following equation applies:

$$h_s = h_{gs} - Z_s + \frac{V_s^2}{2g}$$

The negative sign of Z_s indicates that the gage zero is located below the datum.

EXAMPLE The pressure at gage connection a in Figure 23 is below atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with air. The following equations apply:

$$h_d = -h_{gd} + Y_d + \frac{V_d^2}{2g}$$

$$h_s = -h_{gs} - Y_s + \frac{V_s^2}{2g}$$

Note: If a connecting pipe is air-filled, it must be drained before a reading is made. Water cannot be used in the U tube if either h_{dg} or h_{ds} exceeds the height of the rising loop.

MEASURING HEAD WITH MANOMETRIC FLUID GAGES The following examples illustrate the use of manometric fluid gages for measuring head in a centrifugal pump arrangement.

EXAMPLE In Figure 24, the gage pressure is above atmospheric pressure and the connecting line is completely filled with water. The following equation applies:

$$h = \frac{W_m}{W} h_g + Z + \frac{V^2}{2g}$$

where W_m = specific weight (mass) of manometric fluid, lb/ft^3 (kg/liter)
W = specific weight (mass) of liquid pumped, lb/ft^3 (kg/liter)
h_g = suction or discharge gage reading, ft (m) Hg

The quantities h, Z, Y, and V without subscripts apply equally to suction and discharge head measurements.

EXAMPLE The gage pressure in Figure 25 is below atmospheric pressure, and the connecting line is completely filled with air, with a rising loop to prevent water from passing to the manometric fluid column. The following equation applies

$$h = \frac{W_m}{W} h_g = Y + \frac{V^2}{2g}$$

MEASURING HEAD WITH DIFFERENTIAL MANOMETRIC FLUID GAGES Figure 26 indicates a centrifugal pump arrangement in which a differential manometric fluid gage is used to measure head. When this type of gage is used and the connecting lines are completely filled with water, the correct equation is

$$H = \left(\frac{W_m}{W} - 1\right) h_g + \frac{V_d^2}{2g} - \frac{V_s^2}{2g}$$

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In addition to the differential gage, a separate suction gage can be used, as shown in Figures 22 and 25. The equation in this case is

$$h_s = \frac{W_m}{W} h_{gs} - Z + \frac{V_s^2}{2g}$$

MEASURING HEAD WITH BOURDON GAGES AND PRESSURE TRANSDUCERS An example of a centrifugal pump arrangement that uses calibrated bourdon gages for head measurement is shown in Example 3 of Figure 27, with the gage pressure above atmospheric pressure. The distances Z_s and Z_d are measured to the center of the gage and are negative if the center of the gage lies below the datum line.

MEASURING HEAD ON VERTICAL SUCTION PUMPS IN SUMPS AND CHANNELS In vertical-shaft pumps drawing water from large open sumps and having inlet passages whose length does not exceed about three inlet opening diameters, such inlet pieces having been furnished as part of the pump, the total head should be the reading of the discharge connection in feet (meters) plus the vertical distance from the gage centerline to the free water level in the sump in feet (meters) (Example 2 of Figure 27).

Power Measurement The pump input power may be determined with a calibrated motor, a transmission dynamometer, or a torsion dynamometer. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13) are generally used as the basis for most power measurement procedures.

CALIBRATED MOTORS When pump input power is to be determined with a calibrated motor, the power input should be measured at the terminals of the motor to exclude any line losses that may occur between the switchboard and the driver. Certified calibration curves of the motor must be obtained. The calibration should be conducted on the motor in question and not on a similar machine. Such calibrations must indicate the true input-output value of motor efficiency and not some conventional method of determining an arbitrary efficiency. Calibrated laboratory-type electric meters and transformers should be used to measure power input to all motors.

TRANSMISSION DYNAMOMETERS The transmission, or torque-reaction, dynamometer consists of a cradled electric motor with its frame and field windings on one set of bearings and the rotating element on another set, so the frame is free to rotate but is restrained by means of some weighting or measuring device.

When pump input power is to be determined with a transmission dynamometer, the unloaded and unlocked dynamometer must be properly balanced prior to the test at the same speed at which the test is to be run. The balance should be checked against standard weights. After the test the balance must be rechecked to assure that no change has taken place. In the event of an appreciable change, the test should be rerun. An accurate measurement of speed is essential and should not vary from the pump rated speed by more than 1%. Power input is calculated as shown later in this section under "Computations."

TORSION DYNAMOMETERS The torsion dynamometer consists of a length of shafting whose torsional strain when rotating at a given speed and delivering a given torque is measured by some standard method. When pump input power is to be determined with a torsion dynamometer, the unloaded dynamometer should be statically calibrated prior to the test. This is done by restraining one end of the dynamometer shaft and applying a known force at a known radius. Loads should be applied in intervals of 10% of the dynamometer's full scale rating from zero to 100% of full scale. If the dynamometer is to be used in an overload condition, the calibration range should be increased to include the overload force. When the force is removed, the dynamometer must return to within 0.25% of zero. The no-load (zero) output of the dynamometer should be recorded before and after the test. The temperature of the torsion dynamometer during the test runs must be within 10°F (6°C) of the temperature when the dynamic calibrations were made unless the dynamometer is temperature compensated. All torsion dynamometer calibrations should be witnessed and

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approved by all parties to the test. In the event of a variation greater than allowed, a rerun of the test must be made. Power input calculations are shown in the following text.

Speed Measurement The speed of the pump under test is determined by one of the following methods:

Revolution counter (manual or automatic)
Tachometer
Stroboscopic device
Electronic counter

In all cases, the instruments used must be carefully calibrated before the test to demonstrate that they will produce the required speed readout to within the desired accuracy. Accepted accuracy is usually ±0.1%. Should cyclic speed change result in power fluctuations, at least five equally spaced, timed readings should be taken to give a satisfactory mean speed.

COMPUTATIONS

Pump Power

POWER OUTPUT The water power, or useful work, done by the pump is found by the formula

in USCS units $$\text{whp} = \frac{\text{lb of liquid pumped/min} \times \text{total head in ft of liquid}}{33{,}000}$$

in SI units $$\text{wkW} = \frac{\text{kg of liquid pumped/min} \times \text{total head in m of liquid}}{6131}$$

If the liquid has a specific gravity of 1 and the specific weight of the liquid is 62.3 lb/ft^3 (1.0 kg/liter) at 68°F (20°C), the formula is

in USCS units $$\text{whp} = \frac{\text{gpm} \times \text{head in ft}}{3960}$$

in SI units $$\text{wkW} = 9.8\ \text{m}^3/\text{h} \times \text{head in m}$$

POWER INPUT The brake power required to drive the pump is found by the formula

in USCS units $$\text{bhp} = \frac{\text{gpm} \times \text{total head in ft}}{3960 \times \text{pump efficiency}}$$

in SI units $$\text{bkW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{total head in m}}{\text{pump efficiency}}$$

where pump efficiency is obtained from the formula

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

(used as a decimal)

The electric power input to the motor is

in USCS units $$\text{ehp} = \frac{\text{bhp}}{\text{motor efficiency}} = \frac{\text{gpm} \times \text{head in ft}}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

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in SI units $$\text{kW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{head in m}}{\text{pump efficiency} \times \text{motor efficiency}}$$

The kilowatt input to the motor is

$$\text{kW input} = \frac{\text{bhp} \times 0.746}{\text{motor efficiency}}$$

$$= \frac{\text{gpm} \times \text{head} \times 0.746}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

Pump Efficiency The pump efficiency is

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

For an electric-motor-driven pumping unit, the overall efficiency is

$$\text{Overall efficiency} = \text{pump efficiency} \times \text{motor efficiency}$$

In many specifications, it is required that the actual job motor be used to drive the pump during shop or field testing. Using this test setup, the overall efficiency then becomes what is commonly called "wire-to-water" efficiency, which is expressed by the formula

$$\text{Overall efficiency} = \frac{\text{water power}}{\text{electric power input}} = \frac{\text{whp}}{\text{ehp}} = \frac{\text{wkW}}{\text{kW}}$$

Speed Adjustments The best and standard practice in testing pump speed is to use the actual job motor to drive the pump under test. However, for purposes of plotting test results, it becomes necessary to correct the test values at test speed to rated pump speed.

To adjust pump flow rate, head, power, and NPSH from the values recorded during test to another speed, the following formulas[10] should be used:

Capacity: $$Q_2 = \frac{N_2}{N_1} \times Q_1$$

where Q_1 = flow rate at test speed, gpm (m^3/h)
Q_2 = flow rate at rated speed, gpm (m^3/h)
N_1 = test speed, rpm
N_5 = rated speed, rpm

Head: $$H_2 = \left(\frac{N_2}{N_1}\right)^2 \times H_1$$

where H_1 = head at test speed, ft (m)
H_2 = head at rated speed, ft (m)

Power:

in USCS units $$\text{hp}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{hp}_1$$

in SI units $$\text{kW}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{kW}_1$$

where hp_1 = power at test speed, hp
hp_2 = power at rated speed, hp

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kW_1 = power at test speed, kW
kW_2 = power at rated speed, kW

Net positive suction head: $$NPSH_2 = \left(\frac{N_2}{N_1}\right)^2 \times NPSH_1$$

where $NPSH_1$ = net positive suction head at test speed, ft (m)
$NPSH_2$ = net positive suction head at rated speed, ft (m)

Shop or field testing at either reduced or increased speed should be permitted only when absolutely no alternatives are available. It is recommended, if reduced or increased speed tests are used as official performance tests, that the specifications state the test head and speed and that the performance warranties be based on the specified head and speed conditions.

If reduced or increased speed tests are considered, certain affinity laws must be observed to maintain hydraulic similarity between actual and test conditions. These affinity law relationships can be expressed by

$$\frac{Q_1}{Q} = \frac{N_1}{N} = \left(\frac{H_1}{H}\right)^{1/2}$$

where test = Q_1 = flow rate and H_1 = head at N_1 = rpm
actual = Q = flow rate and H = head at N = rpm

RECORDS

Data There probably are as many test data forms as there are test laboratories. Each may or may not have an advantage for its particular application. A form for recording pump performance data is shown in Figure 28.

The manufacturer's serial number, type, and size or other means of identification of each pump and driver involved in the test should be carefully recorded in order that mistakes in identity may be avoided. The dimensions and physical conditions not only of the machine tested but of all associated parts of the plant which have any important bearing on the outcome of a test, should be noted.

Normal practice suggests that one test run be either at or as near as possible to the rated condition and that at least three runs be within the specified range of heads.

Automatic Data Collection Computerized data collection is possible and desirable as long as the required test accuracy can be demonstrated. Multiple sets of data should be recorded at each point to prove that steady state conditions exist. Before and after test, zero (speed) conditions should also be recorded.

Plotting Test Results In plotting curves from the test results, it should be kept in mind that any one point may be in error but that all the points should establish a trend.

Unless some external factor is introduced to cause an abrupt change, a smooth curve can be drawn for the points plotted, not necessarily through each and every one. Figure 29 is a graphic representation showing the determination of pump performance with the total head, power input, and efficiency in percent, all plotted on the same graph with the capacity as the abscissa.

Reports In some instances, a preliminary report may be issued as part of a contract agreement. However, normally a final report is all that is required.

On shop tests of relatively small pumps, the test log and plotted results constitute the entire report. Reports get progressively more involved as pumps become larger. Where model

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RECORD OF PUMP PERFORMANCE TEST

DATE OF TEST_____ TEST NO._____ CUSTOMER_____ CUST. ORDER NO._____
MANUFACTURER'S ORDER NO._____ PLANT_____ UNIT NO._____
PROJECT_____
RATED CONDITIONS:
CAPACITY, G.P.M. (m^3/h)_____ TOTAL HEAD, FEET (m)_____ R.P.M._____
OVERALL EFFICIENCY PERCENT_____ RANGE OF HEAD_____
DRIVER:
TYPE_____ HORSEPOWER (kW)_____
MANUFACTURER_____ SERIAL NO._____ TEST VOLTAGE_____
TEST EQUIPMENT:
DISCHARGE MEASUREMENT METHOD_____ CONVERSION FACTOR_____
DISCHARGE GAGE_____ CORRECTION_____ SUCTION GAGE_____ CORRECTION_____
DIFFERENTIAL BETWEEN GAGES_____ INSIDE DIAMETER SUCTION_____ INSIDE DIAMETER DISCHARGE_____
PUMP DATA:
TYPE OF PUMP_____ SIZE_____ NO. STAGES_____
MANUFACTURER_____ SERIAL NO._____ SUCTION SIZE_____ DISCHARGE SIZE_____

	RUN NO.	1	2	3	4	5	6	7	8	9	10
HEAD	PRESSURE, P.S.I. (kPa)										
	HEAD, FEET (m)										
	GAGE ℄ TO WATER LEVEL, FEET (m)										
	VELOCITY HEAD, FEET (m)										
	TOTAL HEAD, FEET (m)										
CAPACITY	READING										
	CONVERSION										
	FLOW, G.P.M. (m^3/h)										
POWER DATA	MOTOR VOLTAGE										
	AMPERES										
	KILOWATTS										
	TOTAL HORSEPOWER INPUT										
	MOTOR EFFICIENCY, %										
	SPEED, R.P.M.										
	DYNAMOMETER										
	BRAKE HORSEPOWER (kW)										
	WATER HORSEPOWER (kW)										
	PUMP EFFICIENCY, %										
	OVERALL EFFICIENCY, %										

TESTED BY_____ WITNESSED BY_____
TYPE OF TEST_____
(FIELD OR SHOP)
REMARKS:

FIGURE 28 Record of pump performance test

testing is used, the final report is a complete record of the agreements, inspections, personnel, calibration data, sample computations, tabulations, descriptions, discussions, etc.

MODEL TESTING

Models are tested for one or more of the following purposes:

1. To develop new ideas and new designs
2. To give the manufacturer a range of warranties for performance and efficiency

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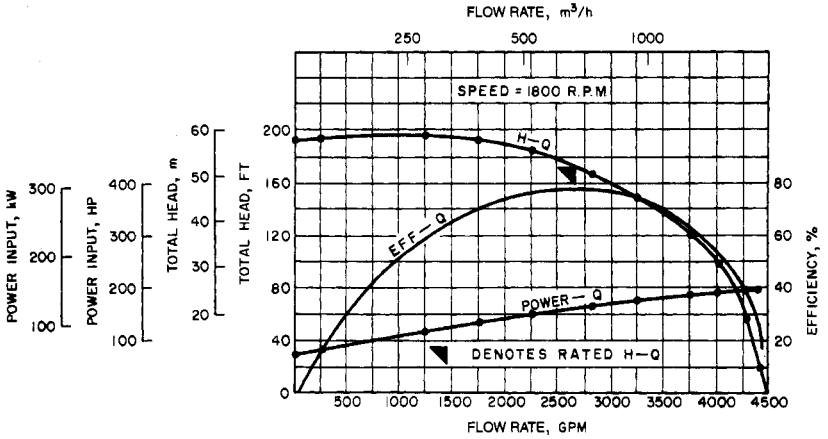


FIGURE 29 Plotted pump performance

3. To give the buyer some assurance that requirements are being met
4. To replace or supplement field testing of a prototype
5. To compare performances of several models

Model testing in advance of final design and installation of a large unit not only provides advance assurance of satisfactory performance but allows for alterations in time for incorporation into the prototype.

Testing Procedures The model should have complete geometric similarity to the prototype in all wetted parts between the intake and discharge sections of the pump. The model should be tested in the same horizontal- or vertical-shaft position as the prototype. The speed of the model should be such that, at the test head, the specific speed for each test run is the same as that of the installed unit or prototype. Unless otherwise specified, the suction head or suction lift should give the same (cavitation factor) value.

If model and prototype diameters are D_1 and D, respectively, then the model speed N_1 and capacity Q_1 under the test head H_1 must agree with the relations

$$\frac{N_1}{N} = \frac{D}{D_1}\sqrt{\frac{H_1}{H}} \quad \text{and} \quad \frac{Q_1}{Q} = \left(\frac{D_1}{D}\right)^2\sqrt{\frac{H_1}{H}}$$

In testing a model of reduced size under the previous conditions, complete hydraulic similarity will not be secured unless the relative roughness of the impeller and pump casing surfaces are the same. With the same surface texture in model and prototype, the model efficiency will be lower than that of the prototype, and greater relative clearances and shaft friction in the model will also reduce its efficiency.

The efficiency of a pump model can conveniently be stepped up to match the prototype efficiency by applying a formula of the same general form as the Moody formula used for hydraulic turbines:

$$\frac{1 - e_1}{1 - e} = \left(\frac{D}{D_1}\right)^n$$

The exponent n should be determined for a given laboratory and given type of pump on the basis of an adequate number of comparisons of the efficiencies of models and prototypes, with consistent surface finish in model and prototype. The *Hydraulic Institute Standards*[2] states that n has been found to vary from zero (when the surface roughness

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and clearances of the model and prototype are proportional to their size) to 0.26 (when the absolute roughness is the same in both).

When model tests are to serve as acceptance tests, it is generally recommended that the efficiency guarantees be stated in terms of model performance rather than in terms of calculated prototype performance. In the absence of such a provision, the efficiency stepup formula and the numerical value of its exponent should be clearly specified or agreed upon in advance of tests.

The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 10) give an example of model testing as follows:

EXAMPLE A single-stage pump to deliver 200 ft³/s (5.66 m³/s) against a head of 400 ft (122 m) at 450 rpm and with a positive suction head, including velocity head, of 10 ft (3 m) has an impeller diameter of 6.8 ft (2.lm). The pump being too large for a shop or laboratory test, a model with an 18-in (0.46-m) impeller is to be tested at a reduced head at 320 ft (97.5 m). At what speed, capacity, and suction head should the test be run?

Applying the above relations:

in USCS units $N_1 = N\dfrac{D}{D_1}\sqrt{\dfrac{H_1}{H}} = 450\left(\dfrac{6.8}{1.5}\right)\sqrt{\dfrac{320}{400}} = 1825$ rpm

$$Q_1 = Q\left(\frac{D_1}{D}\right)^2\sqrt{\frac{H_1}{H}} = 200\left(\frac{1.5}{6.8}\right)^2\sqrt{\frac{320}{400}} = 8.73 \text{ ft}^3/\text{s} = 3920 \text{ gpm}$$

in SI units $N_1 = 450\left(\dfrac{2.1}{0.46}\right)^2\sqrt{\dfrac{97.5}{122}} = 1825$ rpm

$$Q_1 = 5.66\left(\frac{0.46}{22.1}\right)^2\sqrt{\frac{97.5}{122}} = 0.247 \text{ m}^3/\text{s}$$

To check these results, the specific speed of the prototype is

in USCS units $N_s = N\dfrac{\sqrt{Q}}{H^{3/4}} = 450\dfrac{\sqrt{200}}{400^{3/4}} = 71.2$ in the ft³/s system

in SI units $N_s = \dfrac{450\sqrt{5.66}}{122^{3/4}} = 39$ in the m³/s system

and that of the model is

in USCS units $N_{s1} = 1.825\dfrac{\sqrt{8.73}}{320^{3/4}} = 71.2$ (or 1,510 in the gpm system)

in SI units $N_{s1} = 1825\dfrac{\sqrt{0.247}}{97.5^{314}} = 29$

The cavitation factor σ for the field installation, assuming a water temperature of 80°F (27°C) as a maximum probable value and $H_b = 32.8$ ft (10 m) as in the first example, will be

in USCS units $\sigma = \dfrac{H_b - H_s}{H} = \dfrac{32.8 - 10}{400} = 0.057$

in SI units $\sigma = \dfrac{10 - 3}{122} = 0.057$

where $H_b = h_{sa} - H_{vpa}$ (absolute atmospheric pressure minus absolute vapor pressure)

H_s = distance from datum to suction level

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which should be the same in the test. With the water temperature approximately the same,

$$\sigma = \frac{H_b - H_{s1}}{H_1}$$

in USCS units $H_{s1} = H_b - \sigma H = 32.8 - (0.057)(320) = 14.6$ ft

in SI units $H_{s1} = 10 - (0.057)(97.5) = 4.4$ m

Hence the model should be tested with a positive suction head of 14.6 ft (4.4 m) to reproduce the field conditions.

Normally, one of the requirements when using model tests as acceptance tests is to make sure true geometric similarity exists between model and installed prototype. True values of all required and specified dimension should be determined. The actual parts, areas, shape, clearances, and positions should be clearly understood by all parties. Also, the amount of permissible geometric deviation between prototype and model should be agreed to, in writing, before the test is begun.

OTHER OBSERVATIONS

Testing Noncentrifugal Pumps The next largest class of pumps after centrifugal are displacement pumps. This classification includes reciprocating, rotary, screw, and other miscellaneous displacement pumps. Testing of these closely parallels the centrifugal procedures. Normally the capacities are smaller and the heads higher, but the objectives, methods, and measurements are all about the same. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (References 11 and 12) thoroughly cover testing of rotary and reciprocating pumps.

The testing of pumps not falling into the two broad classifications of centrifugal and displacement is usually very special, and each case must be treated separately. The test procedures are normally spelled out in the specifications; otherwise an agreement between all parties must be made before testing is started. The testing of eduction or jet pumps falls under this special category of testing.

Other Test Phenomena When testing pumps, other phenomena of interest should also be checked and noted on the test record. The two phenomena normally reported on are vibration and noise. The acceptable limits of these plus instrumentation for measuring them are special and normally covered in the contract. If the pumps are to be installed in a special environment, this should also be taken into consideration during testing.

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A • P • P • E • N • D • I • X

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NOTE: For more units and conversions and wealth of general technical information, the following resource is recommended: www.nist.gov

TABLE 1A Base units of the International System of Units (SI) and their definitions

Symbol	Unit	Definition
m	meter[a]	Unit of length equal to 1,650,763.73 wavelengths in vacuum of the radiation corresponding to the transition between the $2p^{10}$ and $5d^5$ levels of the krypton-86 atom
kg	kilogram	Unit of mass equal to that of the international prototype of the kilogram
s	second	Unit of time equal to the duration of 9,192,631,770 periods of the radiation corresponding to the transition between two hyperfine levels of the ground state of the cesium-133 atom
A	ampere	Unit of electric current equal to that which, if maintained in two straight parallel conductors of infinite length and negligible cross-section and placed 1 meter (m) apart in vacuum, produces between those conductors a force equal to 2×10^{-7} newtons per meter (N/m) of length
K	kelvin	Unit of thermodynamic temperature equal to the fraction 1/273.16 of the thermodynamic temperature of the triple point of water
cd	candela	Unit of luminous intensity equal to that (in the perpendicular direction) of a surface of 1/600,000 m^2 of a blackbody at the temperature of freezing platinum under a pressure of 101,325 newtons per square meter (N/m^2)
mol	mole	Unit of substance equal to the moment of substance of a system that contains as many elementary entities[b] as there are atoms in 0.012 kg of carbon-12

[a]Spelled *metre* in countries other than the United States.

[b]When the mole is used, the elementary entities must be specified and may be atoms, molecules, ions, electrons, other particles, or specified groups of each particle.

TABLE 1B Supplementary SI units and their definitions

Symbol	Unit	Definition
rad	radian	Unit of measure of a planar angle with its vertex at the center of a circle subtended by an arc equal in length to the radius
sr	steradian	Unit of measure of a solid angle with its vertex at the center of a sphere and enclosing an area of the spherical surface equal to that of a square with sides equal in length to the radius

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TABLE 2 Derived units of the SI system

Quantity	Unit	Symbol	Formula
Acceleration	meters per second per second	m/s^2	—
Angular acceleration	radians per second per second	rad/s^2	—
Angular velocity	radians per second	rad/s	—
Area	square meters	m^2	—
Density	kilograms per cubic meter	kg/m^3	—
Energy	joules	J	$N \cdot m$
Force	newtons	N	$kg \cdot m/s^2$
Frequency	hertz	Hz	cycle/s
Power	watts	W	J/s
Pressure	pascals	Pa	N/m^2
Stress	newtons per square meter	N/m^2	—
Velocity	meters per second	m/s	—
Viscosity, dynamic	newton-seconds per square meter	$N \cdot s/m^2$	—
Viscosity, kinematic	square meters per second	m^2/s	—
Volume	cubic meters	m^3	—
Work	joules	J	$N \cdot m$

TABLE 3 Prefixes for SI multiple and submultiple units

Prefix	SI symbol	Multiplication factor
exa	E	10^{18}
peta	P	10^{15}
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
hecto	h	10^{2}
deka	da	10
deci	d	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}
femto	f	10^{-15}
atto	a	10^{-18}

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TABLE 4 Velocity and frictional head loss in old cast iron piping based on Hazen-Williams $C = 100$

gpm[a]	*v*[b]	*f*[c]	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	gpm
							3-in ID pipe[d]		4-in ID pipe										
30							1.36	0.534	0.77	0.131	5-in ID pipe								30
40							1.81	0.910	1.02	0.224									40
50							2.27	1.38	1.28	0.338	0.82	0.114	6-in ID pipe						50
60							2.72	1.92	1.53	0.475	0.98	0.160							60
70							3.18	2.56	1.79	0.631	1.14	0.213	0.79	0.088					70
80							3.63	3.28	2.04	0.808	1.31	0.273	0.91	0.112					80
90							4.08	4.08	2.30	1.01	1.47	0.339	1.02	0.139					90
100							4.54	4.96	2.55	1.22	1.63	0.412	1.14	0.170	8-in ID pipe				100
125							5.68	7.50	3.19	1.85	2.04	0.623	1.42	0.256					125
150							6.81	10.5	3.83	2.59	2.47	0.874	1.70	0.360	0.96	0.89			150
175							7.95	14.0	4.47	3.44	2.86	1.16	1.99	0.478	1.12	0.118			175
200							9.08	17.9	5.10	4.41	3.27	1.49	2.27	0.613	1.28	0.151	10-in ID pipe		200
225							10.2	22.3	5.74	5.48	3.68	1.85	2.55	0.762	1.44	0.188			225
250							11.3	27.1	6.38	6.67	4.08	2.25	2.84	0.926	1.60	0.228	1.02	0.077	250
275							12.5	32.3	7.02	7.96	4.50	2.68	3.12	1.11	1.76	0.272	1.12	0.092	275
	12-in ID pipe																		
300							13.6	37.9	7.65	9.34	4.90	3.13	3.41	1.30	1.91	0.320	1.23	0.108	300
350	0.99	0.059	14-in ID pipe				15.9	50.4	8.93	12.4	5.72	4.20	3.97	1.73	2.23	0.425	1.43	0.144	350
400	1.13	0.076					18.2	64.6	10.2	15.9	6.54	5.38	4.54	2.21	2.55	0.545	1.63	0.184	400
450	1.28	0.094	0.94	0.044					11.5	19.8	7.36	6.68	5.10	2.75	2.87	0.678	1.84	0.228	450
500	1.42	0.114	1.04	0.054				12.8	24.1	8.18	8.12	5.68	3.34	3.19	0.823	2.04	0.278		500
					16-in ID pipe														
550	1.56	0.136	1.15	0.064					14.0	28.7	8.99	9.69	6.24	3.99	3.51	0.982	2.24	0.331	550
600	1.70	0.160	1.25	0.076	0.96	0.039			15.3	33.7	9.81	11.4	6.81	4.68	3.82	1.15	2.45	0.389	600
650	1.84	0.186	1.36	0.088	1.04	0.046	18-in ID pipe		16.6	39.1	10.6	13.2	7.38	5.43	4.15	1.34	2.65	0.452	650
700	1.99	0.214	1.46	0.100	1.12	0.052			17.9	44.9	11.4	15.1	7.94	6.23	4.47	1.53	2.86	0.518	700
750	2.13	0.242	1.56	0.114	1.20	0.060	0.95	0.034			12.3	17.2	8.51	7.08	4.78	1.74	3.06	0.589	750
800	2.27	0.273	1.67	0.129	1.28	0.067	1.01	0.038	20-in ID pipe		13.1	19.4	9.08	7.98	5.10	1.97	3.26	0.666	800
900	2.55	0.339	1.88	0.160	1.44	0.084	1.13	0.047			14.7	24.1	10.2	9.92	5.74	2.44	3.67	0.825	900
1,000	2.83	0.412	2.08	0.195	1.60	0.102	1.26	0.057	1.02	0.034	16.3	29.3	11.4	12.1	6.38	2.97	4.08	1.00	1,000
1,100	3.12	0.492	2.29	0.232	1.76	0.121	1.39	0.068	1.12	0.041	18.0	35.0	12.5	14.4	7.02	3.55	4.50	1.20	1,100
1,200	3.40	0.578	2.50	0.273	1.92	0.143	1.51	0.080	1.23	0.048			13.6	16.9	7.66	4.17	4.90	1.41	1,200

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1,300	3.69	0.671	2.71	0.316	2.08	0.165	1.64	0.093	1.33	0.056	24-in ID pipe		14.8	19.6	8.30	4.83	5.31	1.63	1,300
1,400	3.97	0.770	2.92	0.363	2.24	0.190	1.76	0.107	1.43	0.064			15.9	22.5	8.93	5.54	5.72	1.87	1,400
1,500	4.25	0.875	3.12	0.413	2.40	0.215	1.89	0.121	1.53	0.073	1.06	0.030	17.0	25.5	9.55	6.30	6.12	2.13	1,500
1,600	4.54	0.985	3.33	0.465	2.55	0.243	2.02	0.137	1.63	0.082	1.13	0.034	18.2	28.8	10.2	7.10	6.53	2.39	1,600
1,800	5.11	1.22	3.75	0.578	2.87	0.302	2.27	0.170	1.84	0.102	1.28	0.042			11.5	8.83	7.35	2.98	1,800
2,000	5.67	1.49	4.17	0.703	3.19	0.367	2.52	0.207	2.04	0.124	1.42	0.051			12.8	10.7	8.17	3.62	2,000
2,500	7.09	2.25	5.21	1.06	3.99	0.555	3.15	0.312	2.55	0.187	1.77	0.077			16.0	16.2	10.2	5.48	2,500
3,000	8.51	3.16	6.25	1.49	4.78	0.778	3.78	0.438	3.06	0.262	2.13	0.108			19.1	22.8	12.3	7.67	3,000
3,500	9.93	4.20	7.29	1.98	5.59	1.04	4.41	0.583	3.57	0.349	2.48	0.143					14.3	10.2	3,500
4,000	11.3	5.38	8.33	2.54	6.39	1.33	5.04	0.746	4.08	0.447	2.83	0.184					16.3	13.1	4,000
4,500	12.8	6.68	9.38	3.15	7.18	1.65	5.67	0.928	4.59	0.555	3.19	0.228					18.4	16.3	4,500
5,000	14.2	8.13	10.4	3.83	7.98	2.00	6.30	1.13	5.10	0.675	3.54	0.278							5,000
5,500	15.6	9.70	11.5	4.58	8.78	2.39	6.93	1.35	5.61	0.806	3.90	0.332							5,500
6,000	17.0	11.4	12.5	5.38	9.68	2.81	7.56	1.58	6.12	0.947	4.25	0.390							6,000
6,500	18.4	13.2	13.6	6.24	10.4	3.26	8.19	1.83	6.73	1.10	4.61	0.452							6,500
7,000	19.9	15.2	14.6	7.16	11.2	3.74	8.82	2.11	7.15	1.26	4.96	0.518							7,000
7,500			15.6	8.13	12.0	4.24	9.45	2.39	7.66	1.43	5.32	0.589							7,000
8,000			16.7	9.16	12.8	4.79	10.1	2.69	8.17	1.61	5.66	0.664							8,000
9,000			18.8	11.4	14.4	5.95	11.3	3.39	9.18	2.01	6.38	0.825							9,000
10,000					16.0	7.24	12.6	4.07	10.2	2.44	7.09	1.00							10,000
11,000					17.6	8.63	13.9	4.86	11.2	2.91	7.80	1.20							11,000
12,000					19.2	10.1	15.1	5.71	12.3	3.42	8.51	1.41							12,000
13,000							16.4	6.62	13.3	3.96	9.12	1.63							13,000
14,000							17.6	7.59	14.3	4.54	9.93	1.87							14,000
15,000							18.9	8.63	15.3	5.27	10.6	2.13							15,000
16,000									16.3	5.82	11.3	2.40							16,000
18,000									18.4	7.24	12.8	2.98							18,000
20,000											14.2	3.62							20,000
25,000											17.7	5.48							25,000

[a] 1 gpm = 6.31×10^{-5} m³/s = 0.227 m³/h = 0.0631 liters/s

[b] Velocity, in feet per second. 1 ft/s = 0.305 m/s

[c] Frictional head loss, in feet of water per 100 feet of pipe. 1 ft = 0.305 m. Friction values apply to cast iron pipes after 15 years of service handling average water at 60°F (15.6°C). Based on Hazen-Williams formula with $C = 100$. See Figure 1 for other values of C.

[d] 1 in = 25.4 mm

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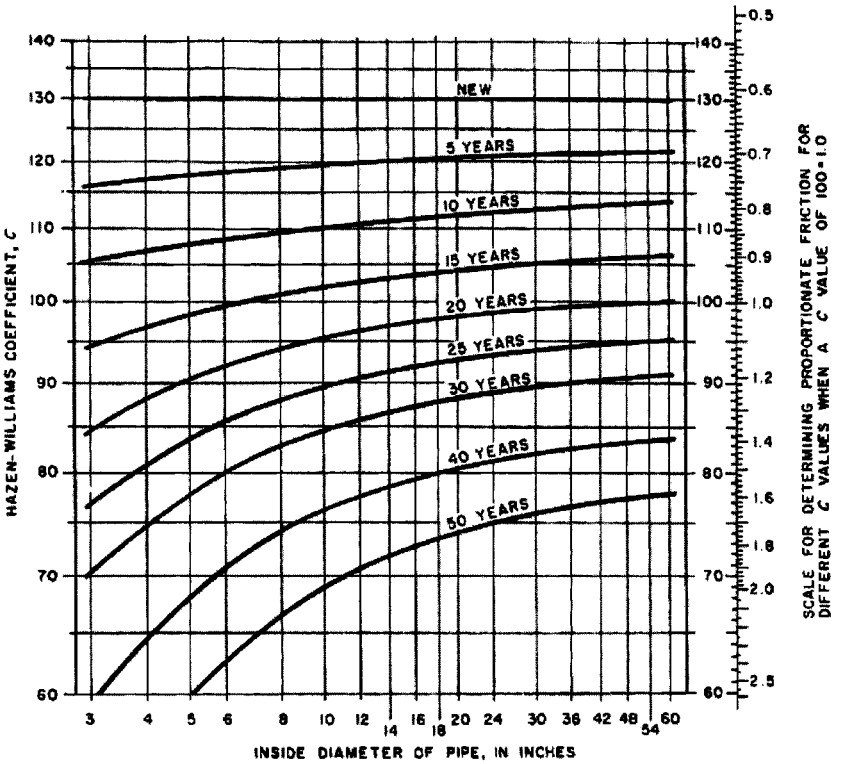


FIGURE 1 Change in Hazen-Williams coefficient C with years of service, for cast iron pipes handling soft, clear, unfiltered water at 60°F (15.6°C). To correct head loss from Table 4, which is based on $C = 100$, multiply by the conversion factor. For example, with a flow of 700 gpm through a 6-in pipe, the frictional head loss is 6.23 ft per 100 ft of pipe with $C = 100$. For $C = 130$, the conversion factor is 0.613; therefore, the frictional head loss is $6.23 \times 0.613 = 3.82$ ft per 100 ft of pipe (see also Section 8.1) (1 in = 25.4 mm).

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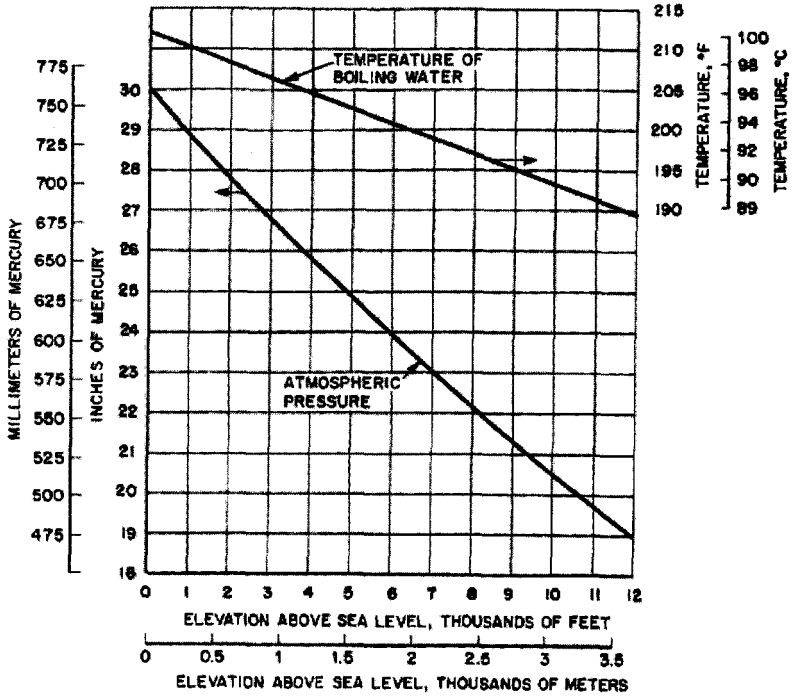


FIGURE 2 Atmospheric pressures for altitudes up to 12,000 ft (3600 m)

TABLE 5 Velocity and frictional head loss in new schedule 40 (standard weight) steel piping based on Darcy-Weisbach

gpm[a]	*v*[b]	*f*[c]	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	gpm
					1-in pipe[d] (1.049-in ID)		$1\frac{1}{4}$-in pipe (1.380-in ID)		$1\frac{1}{2}$-in pipe (1.610-in ID)								
1					0.37	0.11											1
2					0.74	0.39	0.43	0.10									2
3					1.11	0.82	0.64	0.21	0.47	0.10							3
4					1.49	1.37	0.86	0.36	0.63	0.17	2-in pipe						4
5					1.86	2.08	1.07	0.54	0.79	0.26	(2.067-in ID)						5
													$2\frac{1}{2}$-in pipe				
6					2.23	2.83	1.28	0.76	0.95	0.35	0.57	0.10	(2.469-in ID)				6
8					2.97	4.88	1.72	1.29	1.26	0.61	0.76	0.17			3-in pipe		8
10					3.71	7.12	2.14	1.95	1.57	0.90	0.96	0.26	0.67	0.11	(3.068-in ID)		10
15	$3\frac{1}{2}$-in pipe				5.56	15.0	3.21	4.06	2.36	1.87	1.43	0.54	1.00	0.23			15
20	(3.548-in ID)				7.41	25.6	4.28	6.80	3.15	3.12	1.91	0.92	1.34	0.38	0.87	0.13	20
			4-in pipe														
25	0.81	0.10	(4.026-in ID)				5.35	10.3	3.94	4.70	2.38	1.39	1.67	0.58	1.08	0.20	25
30	0.97	0.13					6.43	14.4	4.72	6.60	2.86	1.92	2.00	0.81	1.30	0.28	30
40	1.30	0.23	1.01	0.12					6.30	11.2	3.82	3.35	2.68	1.36	1.73	0.47	40
50	1.62	0.34	1.26	0.18	5-in pipe				7.87	16.6	4.77	5.00	3.34	2.06	2.16	0.72	50
60	1.94	0.48	1.51	0.25	(5.047-in ID)						5.72	7.00	4.02	2.85	2.60	0.99	60
70	2.27	0.63	1.76	0.34	1.12	0.11	6-in pipe				6.68	9.40	4.68	3.80	3.03	1.33	70
80	2.59	0.82	2.01	0.43	1.28	0.15	(6.065-in ID)				7.62	11.9	5.35	4.95	3.46	1.72	80
90	2.91	1.00	2.27	0.54	1.44	0.18					8.60	14.7	6.02	6.05	3.89	2.13	90
100	3.24	1.24	2.52	0.65	1.60	0.22	1.11	0.09			9.56	18.7	6.70	7.47	4.83	2.58	100
125	4.05	1.82	3.15	1.00	2.00	0.33	1.39	0.13					8.37	11.1	5.41	3.90	125
150	4.86	2.55	3.78	1.39	2.40	0.47	1.66	0.18	8-in pipe				10.0	15.4	6.50	5.44	150
175	5.66	3.40	4.40	1.90	2.80	0.62	1.94	0.24	(7.981-in ID)				11.7	20.8	7.58	7.30	175
200	6.48	4.35	5.04	2.40	3.20	0.80	2.22	0.31							8.66	9.18	200
225	7.30	5.44	5.66	2.98	3.60	0.97	2.50	0.38	1.44	0.10					9.75	11.6	225
250	8.10	6.59	6.29	3.68	4.00	1.19	2.77	0.47	1.60	0.12					10.8	14.0	250

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275	8.91	7.90	6.92	4.35	4.40	1.43	3.05	0.56	1.76	0.15	10-in pipe			11.9	16.9	275
300	9.72	9.30	7.55	5.04	4.80	1.65	3.32	0.66	1.92	0.17	(10.020-in ID)			13.0	19.6	300
350	11.3	12.2	8.80	6.85	5.60	2.21	3.88	0.88	2.24	0.23						350
400	13.0	15.9	10.1	8.67	6.40	2.89	4.44	1.12	2.56	0.29	1.62	0.10				400
450	14.6	20.0	11.3	10.9	7.20	3.56	4.99	1.40	2.88	0.37	1.82	0.12				450
500			12.6	13.3	8.00	4.36	5.54	1.72	3.20	0.45	2.03	0.15				500
550			13.9	16.0	8.80	5.17	6.10	2.06	3.52	0.55	2.23	0.18				550
600			15.1	19.1	9.60	6.16	6.65	2.42	3.84	0.63	2.44	0.21				600
650					10.4	7.22	7.20	2.78	4.16	0.73	2.64	0.24				650
700					11.2	8.29	7.75	3.25	4.47	0.85	2.84	0.28				700
750					12.0	9.40	8.31	3.63	4.80	0.97	3.04	0.31				750
800					12.8	10.3	8.87	4.11	5.11	1.11	3.25	0.35				800
900					14.4	13.0	9.96	5.12	5.75	1.33	3.65	0.44				900
1,000					16.0	15.8	11.1	6.17	6.40	1.64	4.06	0.55				1,000
1,100					17.6	19.0	12.2	7.45	7.04	1.98	4.46	0.64				1,100
1,200							13.3	8.73	7.67	2.36	4.87	0.75				1,200
1,300							14.4	10.2	8.31	2.71	5.27	0.88				1,300
1,400							15.5	11.9	8.95	3.10	5.68	1.02				1,400
1,500							16.7	13.2	9.60	3.49	6.09	1.18				1,500
1,600							17.8	15.0	10.2	3.92	6.49	1.31				1,600
1,800							20.0	18.5	11.5	4.99	7.30	1.60				1,800
2,000									12.8	5.96	8.11	1.97				2,000
2,500									16.0	9.00	10.2	2.95				2,500
3,000									19.2	12.5	12.2	4.15				3,000
3,500									22.4	16.6	14.2	5.60				3,500
4,000											16.2	6.90				4,000
4,500											18.3	8.80				4,500
5,000											20.3	10.8				5,000
5,500											22.3	13.0				5,500
6,000											24.4	15.3				6,000

[a]1 gpm = 6.31×10^{-5} m³/s = 0.227 m³/h = 0.0631 liters/s

[b]Velocity, in feet per second. 1 ft/s = 0.305 m/s

[c]Frictional head loss, in feet of water per 100 feet of pipe. 1 ft = 0.305 m. Friction values apply to pipe carrying water at 60°F (15.6°C).

[d]1 in = 25.4 mm

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TABLE 6 Viscosity of common liquids
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	Viscosity			
		SSU	Centistokes	At °F	At °C
Freon	1.37 to 1.49 @ 70°F (21°C)		0.27–0.32	70	21
Glycerine (100%)	1.26 @ 68°F (20°C)	2950	648	68.6	20.3
		813	176	100	38
Glycol					
Propylene	1.038 @ 68°F (20°C)	240.6	52	70	21
Triethylene	1.125 @ 68°F (20°C)	185.7	40	70	21
Diethylene	1.12	149.7	32	70	21
Ethylene	1.125	88.4	17.8	70	21
Hydrochloric acid (31.5%)	1.05 @ 68°F (20°C)		1.9	68	20
Mercury	13.6		0.118	70	21
			0.11	100	38
Phenol (carbolic acid)	0.95 to 1.08	65	11.7	65	18
Silicate of soda	40 Baumé	365	79	100	38
	42 Baumé	637.6	138	100	38
Sulfuric acid (100%)	1.83	75.7	14.6	68	20
		Fish and animal oils			
Bone oil	0.918	220	47.5	130	54
		65	11.6	212	100
Cod oil	0.928	**150**	32.1	100	38
		95	19.4	130	54
Lard	0.96	**287**	62.1	100	38
		160	34.3	130	54
Lard oil	0.912 to 0.925	**190** to **220**	41 to 47.5	100	38
		112 to 128	23.4 to 27.1	130	54

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Menhaden oil	0.933	**140**	29.8	100	38
		90	18.2	130	54
Neatsfoot oil	0.917	**230**	49.7	100	38
		130	27.5	130	54
Sperm oil	0.883	**110**	23.0	100	38
		78	15.2	130	54
Whale oil	0.925	**163 to 184**	35 to 39.6	100	38
		97 to 112	19.9 to 23.4	130	54
		Mineral oils			
Automobile crankcase oils (average mid-continent paraffin base):					
SAE 10	0.880 to 0.935[b]	165 to 240	35.4 to 51.9	100	38
		90 to 120	18.2 to 25.3	130	54
SAE 20	0.880 to 0.935[b]	240 to 400	51.9 to 86.6	100	38
		120 to 185	25.3 to 39.9	100	54
SAE 30	0.880 to 0.935[b]	400 to 580	86.6 to 125.5	100	38
		185 to 255	39.9 to 55.1	130	54
SAE 40	0.880 to 0.935[b]	580 to 950	125.5 to 205.6	100	38
		255 to 80	55.1 to 15.6	130 210	54 99
SAE 50	0.880 to 0.935[b]	950 to 1600	205.6 to 352	100	38
		80 to 105	15.6 to 21.6	210	99
SAE 60	0.880 to 0.935[b]	1600 to 2300	352 to 507	100	38
		105 to 125	21.6 to 26.2	210	99
SAE 70	0.880 to 0.935[b]	2300 to 3100	507 to 682	100	38
		125 to 150	26.2 to 31.8	210	99
SAE 10W	0.880 to 0.935[b]	**5000 to 10,000**	1100 to 2200	0	−18
SAE 20W	0.880 to 0.935[b]	**10,000 to 40,000**	2200 to 8800	0	−18

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Mineral oils					
Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Automobile transmission lubricants					
SAE 80	0.880 to 0.935[b]	**100,000** max	22,000 max	0	−18
SAE 90	0.880 to 0.935[b]	**800** to **1500**	173.2 to 324.7	100	38
		300 to 500	64.5 to 108.2	130	54
SAE 140	0.880 to 0.935[b]	950 to 2300	205.6 to 507	130	54
		120 to **200**	25.1 to 42.9	210	99
SAE 250	0.880 to 0.935[b]	Over 2300	Over 507	130	54
		Over 200	Over 42.9	210	99
Crude oils					
Texas, Oklahoma	0.81 to 0.916	40 to 783	4.28 to 169.5	60	16
		34.2 to 210	2.45 to 45.3	100	38
Wyoming, Montana	0.86 to 0.88	74 to 1215	14.1 to 263	60	16
		46 to 320	6.16 to 69.3	100	38
California	0.78 to 0.92	40 to 4840	4.28 to 1063	60	16
		34 to 700	2.4 to 151.5	100	38
Pennsylvania	0.8 to 0.85	46 to 216	6.16 to 46.7	60	16
		38 to 86	3.64 to 17.2	100	38
Diesel engine lubricating oils (based on average mid-continent paraffin base)					
Federal specification no. 9110	0.880 to 0.935[b]	165 to 240	35.4 to 51.9	100	38
		90 to **120**	18.2 to 25.3	130	54
Federal specification no. 9170	0.880 to 0.935[b]	300 to 410	64.5 to 88.8	100	38
		140 to **180**	29.8 to 38.8	130	54
Federal specification no. 9250	0.880 to 0.935[b]	470 to 590	101.8 to 127.8	100	38
		200 to **255**	43.2 to 55.1	130	54

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Federal specification no. 9370	0.880 to 0.935[b]	800 to 1100	173.2 to 238.1	100	38
		320 to **430**	69.3 to 93.1	130	54
Federal specification no. 9500	0.880 to 0.935[b]	490 to 600	106.1 to 129.9	130	54
		92 to **105**	18.54 to 21.6	210	99
Diesel fuel oils					
No. 2 D	0.82 to 0.95[b]	32.6 to 45.5	2 to 6	100	38
		39	1 to 3.97	130	54
No. 3 D	0.82 to 0.95[b]	**45.5** to **65**	6 to 11.75	100	38
		39 to 48	3.97 to 6.78	130	54
No. 4 D	0.82 to 0.95[b]	**140** max	29.8 max	100	38
		70 max	13.1 max	130	54
No. 5 D	0.82 to 0.95[b]	**400** max	86.6 max	122	50
		165 max	35.2 max	160	71
Fuel oils					
No. 1	0.82 to 0.95[b]	34 to 40	2.39 to 4.28	70	21
		32 to **35**	2.69	100	38
No. 2	0.82 to 0.95[b]	36 to 50	3.0 to 7.4	70	21
		33 to **40**	2.11 to 4.28	100	38
No. 3	0.82 to 0.95[b]	**35** to **45**	2.69 to 0.584	100	38
		32.8 to 39	2.06 to 3.97	130	54
No. 5A	0.82 to 0.95[b]	**50** to **125**	7.4 to 26.4	100	38
		42 to 72	4.91 to 13.73	130	54
No. 5B	0.82 to 0.95[b]	125 to	26.4 to	100	38
		400	86.6	122	50
		72 to 310	13.63 to 67.1	130	54
No. 6	0.82 to 0.95[b]	**450** to **3000**	97.4 to 660	122	50
		175 to 780	37.5 to 172	160	71

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Mineral oils					
Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Fuel oil, navy specification	0.989 max[b]	**110** to **225**	23 to 48.6	122	50
		63 to 115	11.08 to 23.9	160	71
Fuel oil, navy II	1.0 max	**1500** max	324.7 max	122	50
		480 max	104 max	60	71
Gasoline	0.68 to 0.74		0.46 to 0.88	60	16
			0.40 to 0.71	100	38
Gasoline (natural)	76.5 degrees API		0.41	68	20
Gas oil	28 degrees API	73	13.9	70	21
		50	7.4	100	38
Insulating oil					
Transformer, switches, and circuit breakers		115 max	24.1 max	70	21
		65 max	11.75 max	100	38
Kerosene	0.78 to 0.82	35	2.69	68	20
		32.6	2	100	38
Machine lubricating oil (average Pennsylvania paraffin base)					
Federal specification no. 8	0.880 to 0.935[b]	112 to 160	23.4 to 34.3	100	38
		70 to **90**	13.1 to 18.2	130	54
Federal specification no. 10	0.880 to 0.935[b]	160 to 235	34.3 to 50.8	100	38
		90 to **120**	18.2 to 25.3	130	54
Federal specification no. 20	0.880 to 0.935[b]	235 to 385	50.8 to 83.4	100	38
		120 to **185**	25.3 to 39.9	130	54
Federal specification no. 30	0.880 to 0.935[b]	385 to 550	83.4 to 119	100	38
		185 to 255	39.9 to 55.1	130	54

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Mineral lard cutting oil					
Federal specification grade 1		**140** to **190**	29.8 to 41	100	38
		86 to 110	17.22 to 23	130	54
Federal specification grade 2		**190** to **220**	41 to 47.5	100	38
		110 to 125	23 to 26.4	130	54
Petrolatum	0.825	100	20.6	130	54
		77	14.8	160	71
Turbine lubricating oil					
Federal specification (penn base)	0.91 average	400 to 440	86.6 to 95.2	100	38
		185 to **205**	39.9 to 44.3	130	54
		Vegetable oils			
Castor oil	0.96 @ 68°F (20°C)	1200 to 1500	259.8 to 324.7	100	38
		450 to 600	97.4 to 129.9	130	54
China wood oil	0.943	**1425**	308.5	69	21
		580	125.5	100	38
Coconut oil	0.925	**140** to **148**	29.8 to 31.6	100	38
		76 to 80	14.69 to 15.7	130	54
Corn oil	0.924	135	28.7	130	54
		54	8.59	212	100
Cotton seed oil	0.98 to 0.925	**176**	37.9	100	38
		100	20.6	130	54
Linseed oil, raw	0.925 to 0.939	**143**	30.5	100	38
		93	18.94	130	54
Olive oil	0.912 to 0.918	**200**	43.2	100	38
		115	24.1	130	54

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Vegetable oils					
Palm oil	0.924	**221**	47.8	100	38
		125	26.4	130	54
Peanut Oil	0.920	**221**	47.8	100	38
		125	26.4	130	54
Rape seed oil	0.919	**250**	54.1	100	38
		145	31	130	54
Rosin oil	0.980	**1500**	324.7	100	38
		600	129.9	130	54
Rosin (wood)	1.09 avg.	500 to **20,000**	108.2 to 4,400	200	93
		1,000 to 50,000	216.4 to 11,000	190	88
Sesame oil	0.923	**184**	39.6	100	38
		110	23	130	54
Soybean oil	0.927 to 0.98	**165**	35.4	100	38
		96	**19.64**	130	54
Turpentine	0.86 to 0.87	33	2.11	60	16
		32.6	2.0	100	38
Sugar, syrups, molasses, and so on					
Corn syrups	1.4 to 1.47	5,000 to 500,000	1,100 to 110,000	100	38
		1,500 to 60,000	324.7 to 13,200	130	54
Glucose	1.35 to 1.44	35,000 to 100,000	7,700 to 22,000	100	38
		4,000 to 11,000	880 to 2,420	150	66
Honey (raw)		340	73.6	100	38
Molasses "A" (first)	1.40 to 1.46	1,300 to 23,000	281.1 to 5,070	100	38
		700 to 8,000	151.5 to 1,760	130	54

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Molasses "B" (second)	1.43 to 1.48	6,400 to 60,000	1,410 to 13,200	100	38
		3,000 to 15,000	660 to 3,300	130	54
Molasses "C" (blackstrap or final)	1.46 to 1.49	17,000 to 250,000	2,630 to 5,500	100	38
		6,000 to 75,000	1,320 to 16,500	130	54
Sucrose solutions (sugar syrups)					
60 Brix	1.29	230	49.7	70	21
		92	18.7	100	38
62 Brix	1.30	310	67.1	70	21
		111	23.2	100	38
64 Brix	1.31	440	95.2	70	21
		148	31.6	100	38
66 Brix	1.326	650	140.7	70	21
		195	42.0	100	38
68 Brix	1.338	1,000	216.4	70	21
		275	59.5	100	38
70 Brix	1.35	1,650	364	70	21
		400	86.6	100	38
72 Brix	1.36	2,700	595	70	21
		640	138.6	100	38
74 Brix	1.376	5,500	1,210	70	21
		1,100	238	100	38
76 Brix	1.39	10,000	2,200	70	21
		2,000	440	100	38
		Tars			
Tar, coke oven	1.12+	3,000 to 8,000	600 to 1,760	71	21
		650 to 1,400	140.7 to 308	100	38
Tar, gas house	1.16 to 1.30	15,000 to 300,000	3,300 to 66,000	70	21
		2,000 to 20,000	440 to 4,400	100	38

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Tars					
Road tar					
Grade RT-2	1.07+	**200** to **300**	43.2 to 64.9	122	50
		55 to 60	8.77 to 10.22	212	100
Grade RT-4	1.08+	**400** to **700**	86.6 to 154	122	50
		65 to 75	11.63 to 14.28	212	100
Grade RT-6	1.09+	**1,000** to **2,000**	216.4 to 440	122	50
		85 to 125	16.83 to 26.2	212	100
Grade RT-8	1.13+	**3,000** to **8,000**	660 to 1,760	122	50
		150 to 225	31.8 to 48.3	212	100
Grade RT-10	1.14+	**20,000** to **60,000**	4,400 to 13,200	122	50
		250 to 400	53.7 to 86.6	212	100
Grade RT-12	1.15+	114,000 to 456,000	25,000 to 75,000	122	50
		500 to **800**	108.2 to 173.2	212	100
Pine tar	1.06	2,500	559	100	38
		500	108.2	132	56
Miscellaneous					
Corn starch solutions					
22 Baumé	1.18	150	32.1	70	21
		130	27.5	100	38
24 Baumé	1.20	600	129.8	70	21
		440	95.2	100	38
25 Baumé	1.21	1,400	303	70	21
		800	173.2	100	38

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Ink, printers	1.00 to 1.38	2,500 to 10,000 1,100 to 3,000	550 to 2,200 238.1 to 660	100 130	38 54
Tallow	0.918 avg.	56	9.07	212	100
Milk	1.02 to 1.05		1.13	68	20
Varnish, spar	0.9	1425 650	313 143	68 100	20 38
Water, fresh	1.0		1.123 0.516	60 130	16 54

[a]Unless otherwise noted

[b]Depends on origin or percent and type of solvent

Source: *Hydraulic Institute Engineering Data Book,* 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

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TABLE 7 Viscosity conversion table

This table gives an approximate comparison of ratings so that, if a viscosity is given in terms other than Saybolt seconds universal, it can be translated quickly by following horizontally to the SSU column.

Saybolt universal seconds SSU	Kinematic viscosity, cSt[a]	Approx. seconds, Mac Michael	Approx. Gardner Holt bubble	Seconds, Zahn cup 1	Seconds, Zahn cup 2	Seconds, Zahn cup 3	Seconds, Zahn cup 4	Seconds, Zahn cup 5	Seconds, Demmler cup 1	Seconds, Demmler cup 10	Approx. seconds, Stormer 100-g load	Seconds, Pratt and Lambert "F"
31	1.00	—	—	—	—	—	—	—	—	—	—	—
35	2.56	—	—	—	—	—	—	—	—	—	—	—
40	4.30	—	—	—	—	—	—	—	1.3	—	—	—
50	7.40	—	—	—	—	—	—	—	2.3	—	2.6	—
60	10.3	—	—	—	—	—	—	—	3.2	—	3.6	—
70	13.1	—	—	—	—	—	—	—	4.1	—	4.6	—
80	15.7	—	—	—	—	—	—	—	4.9	—	5.5	—
90	18.2	—	—	—	—	—	—	—	5.7	—	6.4	—
100	20.6	125	—	38	18	—	—	—	6.5	—	7.3	—
150	32.1	145	—	47	20	—	—	—	10.0	1.0	11.3	—
200	43.2	165	A	54	23	—	—	—	13.5	1.4	15.2	—
250	54.0	198	A	62	26	—	—	—	16.9	1.7	19	—
300	65.0	225	B	73	29	—	—	—	20.4	2.0	23	—
400	87.0	270	C	90	37	—	—	—	27.4	2.7	31	7
500	110.0	320	D	—	46	—	—	—	34.5	3.5	39	8
600	132	370	F	—	55	—	—	—	41	4.1	46	9
700	154	420	G	—	63	22.5	—	—	48	4.8	54	9.5
800	176	470	—	—	72	24.5	—	—	55	5.5	62	10.8
900	198	515	H	—	80	27	18	—	62	6.2	70	11.9
1000	220	570	I	—	88	29	20	138	69	6.9	77	12.4
1500	330	805	M	—	—	40	28	18	103	10.3	116	16.8
2000	440	1070	Q	—	—	51	34	24	137	13.7	154	22
2500	550	1325	T	—	—	63	41	29	172	17.2	193	27.6
3000	660	1690	U	—	—	75	48	33	206	20.6	232	33.7

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4000	880	2110	V	—	—	—	63	43	275	27.5	308	45
5000	1100	2635	W	—	—	—	77	50	344	34.4	385	55.8
6000	1320	3145	X	—	—	—	—	65	413	41.3	462	65.5
7000	1540	3670	—	—	—	—	—	75	481	48	540	77
8000	1760	4170	Y	—	—	—	—	86	550	55	618	89
9000	1980	4700	—	—	—	—	—	96	620	62	695	102
10,000	2200	5220	Z	—	—	—	—	—	690	69	770	113
15,000	3300	7720	Z2	—	—	—	—	—	1030	103	1160	172
20,000	4400	10500	Z3	—	—	—	—	—	1370	137	1540	234

[a]Kinematic viscosity (in centistokes) = $\frac{\text{absolute viscosity (in centipoises)}}{\text{density}}$

When the SI units centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following will be sufficiently accurate for most calculations:

Kinematic viscosity (in centistokes) = $\frac{\text{absolute viscosity (in centipoises)}}{\text{specific gravity}}$

When USCS units are used, the density must be used rather than the specific gravity.

Note: For values of 70 cSt and above, use the conversion

$$\text{SSU} = \text{centistokes} \times 4.635$$

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor
Mac Michael	1.92 (approx.)
Demmler 1	14.6
Demmler 10	146
Stormer	13 (approx.)

Source: *Hydraulic Institute Engineering Data Book*, 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

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TABLE 8 Properties of water at temperatures from 40 to 705.4°F (4.4 to 374.1°C)

°F	°C	Specific volume[a], ft³/lb	Specific gravity			Wt[b], lb/ft³	Vapor pressure[c], lb/in²abs
			39.2°F reference	60°F reference	70°F reference		
40	4.4	0.01602	1.000	1.001	1.002	62.42	0.1217
50	10.0	0.01603	0.999	1.001	1.002	62.38	0.1781
60	15.6	0.01604	0.999	1.000	1.001	62.34	0.2563
70	21.1	0.01606	0.998	0.999	1.000	62.27	0.3631
80	26.7	0.01608	0.996	0.998	0.999	62.19	0.5069
90	32.2	0.01610	0.995	0.996	0.997	62.11	0.6982
100	37.8	0.01613	0.993	0.994	0.995	62.00	0.9492
120	48.9	0.01620	0.989	0.990	0.991	61.73	1.692
140	60.0	0.01629	0.983	0.985	0.986	61.39	2.889
160	71.1	0.01639	0.977	0.979	0.979	61.01	4.741
180	82.2	0.01651	0.970	0.972	0.973	60.57	7.510
200	93.3	0.01663	0.963	0.964	0.966	60.13	11.526
212	100.0	0.01672	0.958	0.959	0.960	59.81	14.696
220	104.4	0.01677	0.955	0.956	0.957	59.63	17.186
240	115.6	0.01692	0.947	0.948	0.949	59.10	24.97
260	126.7	0.01709	0.938	0.939	0.940	58.51	35.43
280	137.8	0.01726	0.928	0.929	0.930	58.00	49.20
300	148.9	0.01745	0.918	0.919	0.920	57.31	67.01
320	160.0	0.01765	0.908	0.909	0.910	56.66	89.66
340	171.1	0.01787	0.896	0.898	0.899	55.96	118.01
360	182.2	0.01811	0.885	0.886	0.887	55.22	153.04
380	193.3	0.01836	0.873	0.874	0.875	54.47	195.77
400	204.4	0.01864	0.859	0.860	0.862	53.65	247.31
420	215.6	0.01894	0.846	0.847	0.848	52.80	308.83
440	226.7	0.01926	0.832	0.833	0.834	51.92	381.59
460	237.8	0.0196	0.817	0.818	0.819	51.02	466.0
480	248.9	0.0200	0.801	0.802	0.803	50.00	566.1
500	260.0	0.0204	0.785	0.786	0.787	49.02	680.8
520	271.1	0.0209	0.765	0.766	0.767	47.85	812.4
540	282.2	0.0215	0.746	0.747	0.748	46.51	962.5
560	293.3	0.0221	0.726	0.727	0.728	45.3	1133.1
580	304.4	0.0228	0.703	0.704	0.704	43.9	1325.8
600	315.6	0.0236	0.678	0.679	0.680	42.3	1542.9
620	326.7	0.0247	0.649	0.650	0.650	40.5	1786.6
640	337.8	0.0260	0.617	0.618	0.618	38.5	2059.7
660	348.9	0.0278	0.577	0.577	0.578	36.0	2365.4
680	360.0	0.0305	0.525	0.526	0.527	32.8	2708.1
700	371.1	0.0369	0.434	0.435	0.435	27.1	3093.7
705.4	374.1	0.0503	0.319	0.319	0.320	19.9	3206.2

[a] 1 m^3/kg = 16.02 ft^3/lb

[b] 1 kg/m^3 = 0.06243 lb/ft^3

[c] 1 kPa = 0.145 lb/in^2; 1 bar = 14.5 lb/in^2

Source: *Hydraulic Institute Engineering Data Book*, 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

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TABLE 9 Conversions of USCS to SI units

The first two digits of each numeral entry represent a power of 10. An asterisk follows each number that expresses an exact definition.

Acceleration

To convert from	to	multiply by
foot/second^2	meter/second^2	−01 3.048*
free fall, standard	meter/second^2	+00 9.806 65*
gal (galileo)	meter/second^2	−02 1.00*
inch/second^2	meter/second^2	−02 2.54*

Area

To convert from	to	multiply by
acre	meter^2	+03 4.046 856 4224*
circular mil	meter^2	−10 5.067 074 8
foot^2	meter^2	−02 9.290 304*
inch^2	meter^2	−04 6.4516*
mile^2 (U.S. statute)	meter^2	+06 2.589 988 110 336*
yard^2	meter^2	−01 8.361 273 6*

Density

To convert from	to	multiply by
gram/centimeter^3	kilogram/meter3	+03 1.00*
lbm/inch^3	kilogram/meter^3	+04 2.767 990 5
lbm/foot^3	kilogram/meter^3	+01 1.601 846 3
slug/foot^3	kilogram/meter^3	+02 5.153 79

Energy

To convert from	to	multiply by
British thermal unit (ISO/TC 12)	joule	+03 1.055 06
British thermal unit (International Steam Table)	joule	+03 1.055 04
British thermal unit (mean)	joule	+03 1.055 87
British thermal unit (thermochemical)	joule	+03 1.054 350 264 488
British thermal unit (39°F)	joule	+03 1.059 67
British thermal unit (60°F)	joule	+03 1.054 68
calorie (International Steam Table)	joule	+00 4.1868
calorie (mean)	joule	+00 4.190 02
calorie (thermochemical)	joule	+00 4.184*
calorie (15°C)	joule	+00 4.185 80

Length

To convert from	to	multiply by
fathom	meter	+00 1.8288*
foot	meter	−01 3.048*
foot (U.S. survey)	meter	+00 1200/3937*
foot (U.S. survey)	meter	−01 3.048 006 096
furlong	meter	+02 2.011 68*
inch	meter	−02 2.54*
league (U.K. nautical)	meter	03 5.559 552*
league (international nautical)	meter	+03 5.556*
league (statute)	meter	+03 4.828 032*
light-year	meter	+15 9.460 55
meter	wavelengths Kr 86	+06 1.650 763 73*
micron	meter	−06 1.00*
mil	meter	−05 2.54*
mile (U.S. statute)	meter	+03 1.609 344*
mile (U.K. nautical)	meter	+03 1.853 184*
mile (international nautical)	meter	+03 1.852*
mile (U.S. nautical)	meter	+03 1.852*
nautical mile (U.K.)	meter	+03 1.853 184*
nautical mile (international)	meter	+03 1.852*
nautical mile (U.S.)	meter	+03 1.852*
rod	meter	+00 5.0292*
statute mile (U.S.	meter	+03 1.609 344*
yard	meter	−01 9.144*

Mass

To convert from	to	multiply by
carat (metric)	kilogram	−04 2.00*
grain	kilogram	−05 6.479 891*
gram	kilogram	−03 1.00*
kgf second^2 meter (mass)	kilogram	+00 9.806 65*
kilogram mass	kilogram	+00 1.00*
lbm (pound mass, avoirdupois)	kilogram	−01 4.535 923 7*
ounce mass (avoirdupois)	kilogram	−02 2.834 952 312 5*
ounce mass (troy or apothecary)	kilogram	−02 3.110 347 68*
pennyweight	kilogram	−03 1.555 173 84*
pound mass, lbm (avoirdupois)	kilogram	−01 4.535 923 7*
pound mass (troy or apothecary)	kilogram	−01 3.732 417 216*
slug	kilogram	+01 1.459 390 29
ton (assay)	kilogram	−02 2.916 666 6
ton (long)	kilogram	+03 1.016 046 908 8*

(*Continued*)

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TABLE 9 Conversions of USCS to SI units (*Continued*)
The first two digits of each numeral entry represent a power of 10. An asterisk follows each number that expresses an exact definition.

Energy

To convert from	to	multiply by
calorie (20°C)	joule	+00 4.181 90
calorie (kilogram, International Steam Table)	joule	+03 4.1868
calorie (kilogram, mean)	joule	+03 4.190 02
calorie (kilogram, thermochemical)	joule	+03 4.184*
electronvolt	joule	−19 1.602 10
erg	joule	−07 1.00*
foot pound force (ft • lbf)	joule	+00 1.355 817 9
foot poundal	joule	−02 4.214 011 0
joule (international of 1948)	joule	+00 1.000 165
kilocalorie (International Steam Table)	joule	+03 4.1868
kilocalorie (mean)	joule	+03 4.190 02
kilocalorie (thermochemical)	joule	+03 4.184*
kilowatthour	joule	+06 3.60*
kilowatthour (international of 1948)	joule	+06 3.600 59
ton (nuclear equivalent of TNT)	joule	+09 4.20
watthour	joule	+03 3.60*

Energy/Area Time

To convert from	to	multiply by
Btu (thermochemical)/foot^2 second	watt/meter^2	+04 1.134 893 1
Btu (thermochemical)/foot^2 minute	watt/meter^2	+02 1.891 488 5
Btu (thermochemical)/foot^2 hour	watt/meter^2	+00 3.152 480 8
Btu (thermochemical)/inch^2 second	watt/meter^2	+06 1.634 246 2
calorie (thermochemical)/cm^2 minute	watt/meter^2	+02 6.973 333 3
erg/centimeter^2 second	watt/meter^2	−03 1.00*
watt/centimeter^2	watt/meter^2	+04 1.00*

Force

To convert from	to	multiply by
dyne	newton	−05 1.00*
kilogram force (kgf)	newton	+00 9.806 65*
kilopond force	newton	+00 9.806 65*
kip	newton	+03 4.448 221 615 260 5*
lbf (pound force, avoirdupois)	newton	+00 4.448 221 615 260 5*
ounce force (avoirdupois)	newton	−01 2.780 138 5
pound force lbf (avoirdupois)	newton	+00 4.448 221 615 260 5*
poundal	newton	−01 1.382 549 543 76*

Mass

To convert from	to	multiply by
ton (metric)	kilogram	+03 1.00*
ton (short, 2000 pound)	kilogram	+02 9.071 847 4*

Power

To convert from	to	multiply by
Btu (thermochemical)/second	watt	+03 1.054 350 264 488
Btu (thermochemical)/minute	watt	+01 1.757 250 4
calorie (thermochemical)/second	watt	+00 4.184*
calorie (thermochemical)/minute	watt	−02 6.973 333 3
foot lbf/hour	watt	−04 3.766 161 0
foot lbf/minute	watt	−02 2.259 696 6
foot lbf/second	watt	+00 1.355 817 9
horsepower (550 foot lbf/second)	watt	+02 7.456 998 7
horsepower (boiler)	watt	+03 9.809 50
horsepower (electric)	watt	+02 7.46*
horsepower (metric)	watt	+02 7.354 99
horsepower (U.K.)	watt	+02 7.457
horsepower (water)	watt	+02 7.460 43
kilocalorie (thermochemical)/minute	watt	+01 6.973 333 3
kilocalorie (thermochemical)/second	watt	+03 4.184*
watt (international of 1948)	watt	+00 1.000 165

Pressure

To convert from	to	multiply by
atmosphere	newton/meter^2	+05 1.013 25*
bar	newton/meter^2	+05 1.00*
centimeter of mercury (0°C)	newton/meter^2	+03 1.333 22
centimeter of water (4°C)	newton/meter^2	+01 9.806 38
dyne/centimeter^2	newton/meter^2	−01 1.00*
foot of water (39.2°F)	newton/meter^2	+03 2.988 98
inch of mercury (32°F)	newton/meter^2	+03 3.386 389
inch of mercury (60°F)	newton/meter^2	+03 3.376 85
inch of water (39.2°F)	newton/meter^2	+02 2.490 82
inch of water (60°F)	newton/meter^2	+02 2.4884
kgf/centimeter^2	newton/meter^2	+04 9.806 65*
kgf/meter^2	newton/meter^2	+00 9.806 65*
lbf/foot^2	newton/meter^2	+01 4.788 025 8

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Pressure

To convert from	to	multiply by
lbf/inch^2 (psi)	newton/meter^2	+03 6.894 757 2
millibar	newton/meter^2	+02 1.00*
millimeter of mercury (0°C)	newton/meter^2	+02 1.333 224
pascal	newton/meter^2	+00 1.00*
psi (lbf/inch^2)	newton/meter^2	+03 6.894 757 2

Speed

To convert from	to	multiply by
foot/hour	meter/second	−05 8.466 666 6
foot/minute	meter/second	−03 5.08*
foot/second	meter/second	−01 3.048*
inch/second	meter/second	−02 2.54*
kilometer/hour	meter/second	−01 2.777 777 8
knot (international)	meter/second	−01 5.144 444 444
mile/hour (U.S. statute)	meter/second	−01 4.4704*
mile/minute (U.S. statute)	meter/second	+01 2.682 24*
mile/second (U.S. statute)	meter/second	+03 1.609 344*

Temperature

To convert from	to	proceed as follows
Celsius	Kelvin	$t_K = t_C + 273.15$
Fahrenheit	Kelvin	$t_K = (\frac{5}{9})(t_F + 459.67)$
Fahrenheit	Celsius	$t_C = (\frac{5}{9})(t_F - 32)$
Rankine	Kelvin	$t_K = (\frac{5}{9})t_R$

Viscosity

To convert from	to	multiply by
centistoke	meter^2/second	−06 1.00*
stoke	meter^2/second	−04 1.00*
foot^2/second	meter^2/second	−02 9.290 304*
centipoise	newton second/meter^2	−03 1.00*
lbm/foot second	newton second/meter^2	+00 1.488 163 9
lbf second/foot^2	newton second/meter^2	+01 4.788 025 8

Viscosity

To convert from	to	multiply by
poise	newton second/meter^2	−01 1.00*
poundal second/foot^2	newton second/meter^2	+00 1.488 163 9
slug/foot second	newton second/meter^2	+01 4.788 025 8
rhe	meter^2/newton second	+01 1.00*

Volume

To convert from	to	multiply by
acre foot	meter^3	+03 1.233 481 9
barrel (petroleum, 42 gallons)	meter^3	−01 1.589 873
bushel (U.S.)	meter^3	−02 3.523 907 016 688*
cord	meter^3	+00 3.624 556 3
cup	meter^3	−04 2.365 882 365*
dram (U.S. fluid)	meter^3	−06 3.696 691 195 312 5*
fluid ounce (U.S.)	meter^3	−05 2.957 352 956 25*
foot^3	meter^3	−02 2.831 684 659 2*
gallon (U.K. liquid)	meter^3	−03 4.546 087
gallon (U.S. dry)	meter^3	−03 4.404 883 770 86*
gallon (U.S. liquid)	meter^3	−03 3.785 411 784*
gill (U.S.)	meter^3	−04 1.182 941 2
inch^3	meter^3	−05 1.638 706 4*
liter	meter^3	−03 1.00*
ounce (U.S. fluid)	meter^3	−05 2.957 352 956 25*
peck (U.S.)	meter^3	−03 8.809 767 541 72*
pint (U.S. dry)	meter^3	−04 5.506 104 713 575*
pint (U.S. liquid)	meter^3	−04 4.731 764 73*
quart (U.S. dry)	meter^3	−03 1.101 220 942 715*
quart (U.S. liquid)	meter^3	−04 9.463 529 5
tablespoon	meter^3	−05 1.478 676 478 125*
teaspoon	meter^3	−06 4.928 921 593 75*
ton (register)	meter^3	+00 2.831 684 659 2*
yard^3	meter^3	−01 7.645 548 579 84*

Volume/Unit Time

To convert from	to	multiply by
gallon/minute	meter^3/second	−05 6.309 020
gallon/minute	meter^3/hour	−01 2.271 247
gallon/minute	liter/second	−02 6.309 020

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