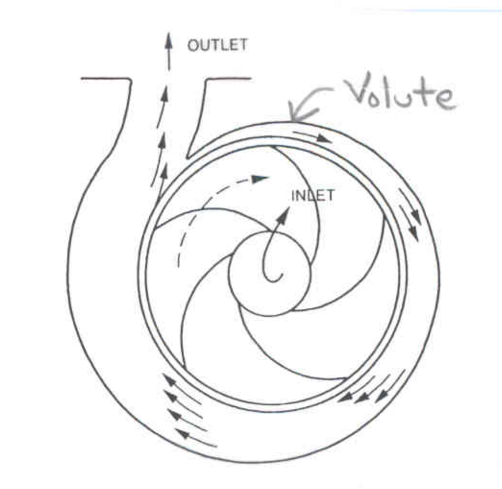
Fluid flow pump system

Pump Types

Fluid flow applications can be divided into two categories: "low flow at high pressure" and "high flow at low pressure". "Low flow at high pressure" applications include hydraulic power systems and typically employ positive-displacement pumps. The vast majorities of fluid-flow applications are categorized as "high flow at low pressure" and typically use centrifugal pumps. The following analysis is for "high flow at low pressure" applications only.

In centrifugal pumps, the fluid enters along the centerline of the pump, is pushed outward by the rotation of the impeller blades, and exits along the outside of the pump. A schematic of a centrifugal pump is shown below.

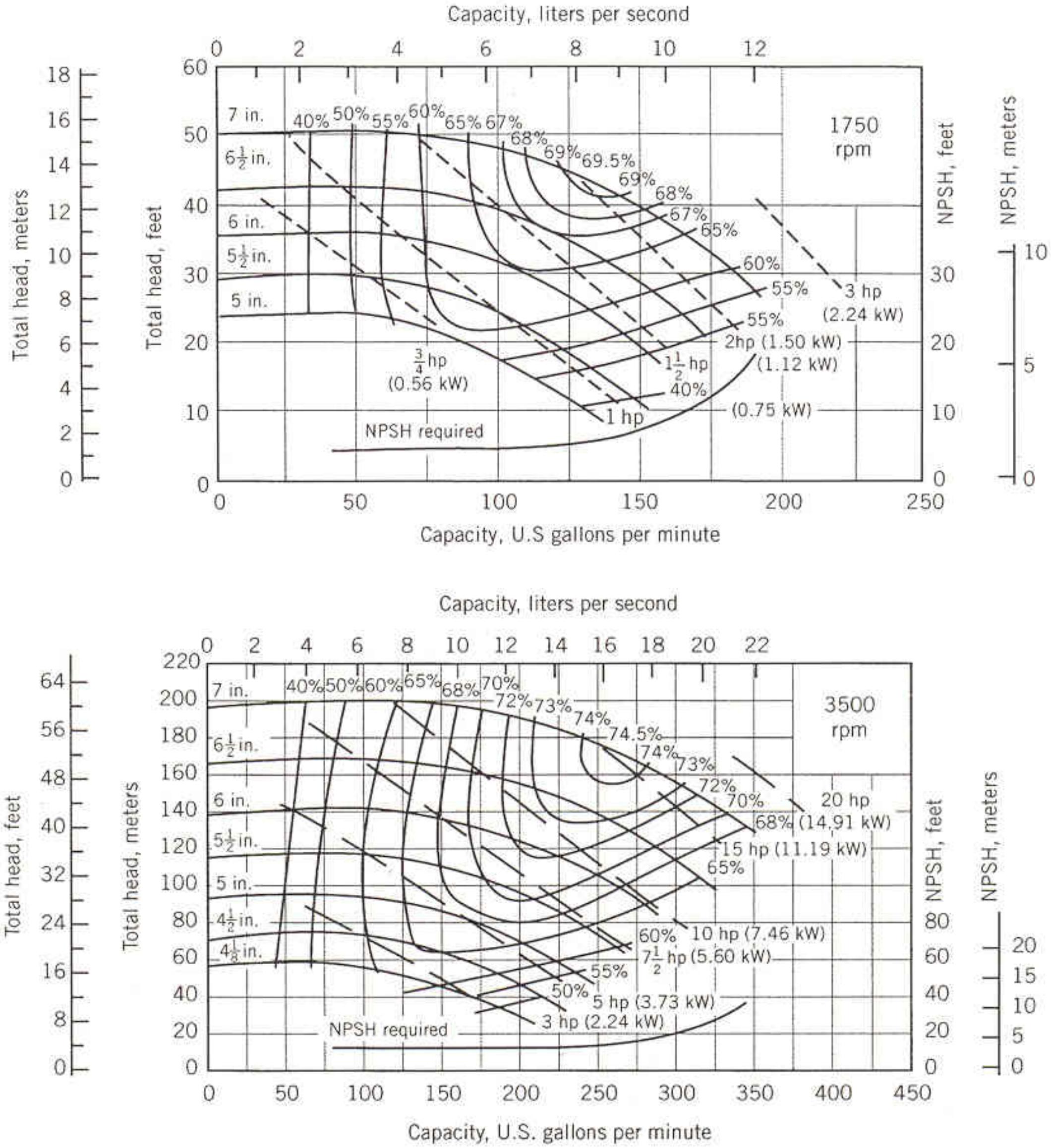


Centrifugal pump

Pump Curves

Pumps can generate high volume flow rates when pumping against low pressure or low volume flow rates when pumping against high pressure. The possible combinations of total pressure and volume flow rate for a specific pump can be plotted to create a pump curve. The curve defines the range of possible operating conditions for the pump.

If a pump is offered with multiple impellers with different diameters, manufacturers typically plot a separate pump curve for each size of impellor on the same pump performance chart. Smaller impellors produce less pressure at lower flow rates. Pump performance charts with multiple pump curves are shown below.



Typical pump performance charts. Source: McQuiston and Parker, 1994, Heating Ventilating and Air Conditioning, John Wiley and Sons, Inc.

The power required to push the fluid through the pipe, W_{fluid}, is the product of the volume flow rate and system pressure drop.

$$W_{fluid} = V \quad \Delta P_{total}$$

Graphically, fluid work is represented by the area under the rectangle defined by the operating point on a pump performance chart.
Typically, the efficiency of the pump at converting the power supplied to the pump into kinetic energy of the fluid is also plotted on the pump performance chart. Pump efficiencies typically range from about 50% to 80%. Power that is not converted into kinetic energy is lost as heat. The power required by the pump, which is sometimes called the “shaft work” or “brake horsepower”, can be calculated

from the flow rate, total pressure, and efficiency values from the pump curve, using the following equation.

$W_{pump} = W_{fluid} \;/\; Eff_{pump} = V \;\; \Delta P_{total} \,/\, Eff_{pump}$

A dimensional version of this equation, using U.S. units for pumping water at standard conditions, is:

W_{pump} (hp) = V (gal/min) ΔP_{total} (ft-H_20) / (3,960 (gal-ft/min-hp) x Eff_{pump})

Many pump performance graphs, including those shown above, also plot curves showing the work required by the pump to produce a specific flow and pressure. Note that these curves show work required by the pump including the efficiency of the pump. Calculating the work supplied to the pump using the preceding equation and comparing it to the value indicated on a pump performance graph is a useful exercise.

System Curve

The total pressure that the pump must produce to move the fluid is determined by the piping system. This total pressure of the piping system is the sum of the pressure due to inlet and outlet conditions and the pressure loss due to friction. In a piping system, pressure loss due to friction increases with increasing fluid flow; thus, system curves have positive slopes on pump performance charts. The operating point of a pump is determined by the intersection of the pump and system curves.

To determine the form of a system curve, consider the equation for total pressure in a piping system. The total pressure caused by a piping system is the sum of the pressure due to inlet and outlet conditions and the pressure required to overcome friction through the pipes and fittings.

$\Delta P_{total} = (\Delta P_{static} + \Delta P_{velocity} + \Delta P_{elevation})_{inlet\text{-}outlet} + \Delta P_{fricition}$

Inlet/Outlet Pressure

The inlet/outlet pressure that the pump must overcome is the sum of the static, velocity and elevation pressures between the inlet and outlet of the piping system. For closed loop piping systems, the inlet and outlet are at the same location; hence, the static, velocity and elevation pressure differences are all zero. For open systems, the differences in static, velocity and elevation pressures must be calculated (See Fluid_Flow Piping).

In many pumping applications, the velocity pressure difference between the inlet and outlet is zero or negligible, and inlet/outlet pressure is simply the sum of the static and elevation heads. In these cases, the inlet/outlet pressure is independent of flow and is represented on a pump performance chart as the pressure at zero flow.

Friction Pressure Drop
Friction between the fluid and pipe walls and fittings increases with the volume flow rate. The equations for pressure loss from friction through the fittings, hlf, and pressure loss from friction with the pipes, hlp, show that in each case friction pressure drop is proportional to the square of velocity:

$hl_f = k_f \underline{V}^2 / 2$

$hl_p = (f\ L\ \underline{V}^2) / (2\ D)$

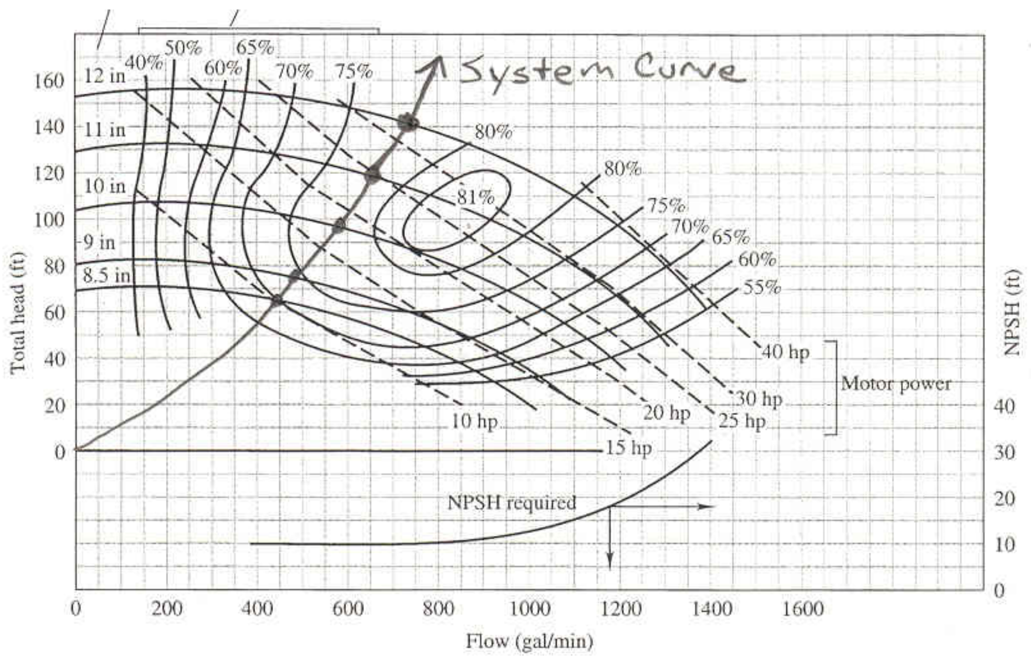
Velocity is proportional to volume flow rate; hence, friction pressure drop is also proportional to the square of volume flow rate.

$\Delta P_{headloss} = \rho(hl) = \rho(hlf + hlp) = \quad \rho\ [\ (k_f \underline{V}^2 / 2) + ((f\ L\ \underline{V}^2) / (2\ D))\ ] = C_1 \underline{V}^2 \quad = C_2 V^2$

This quadratic relationship can be plotted on the pump curve to show the friction component of the “system curve”.

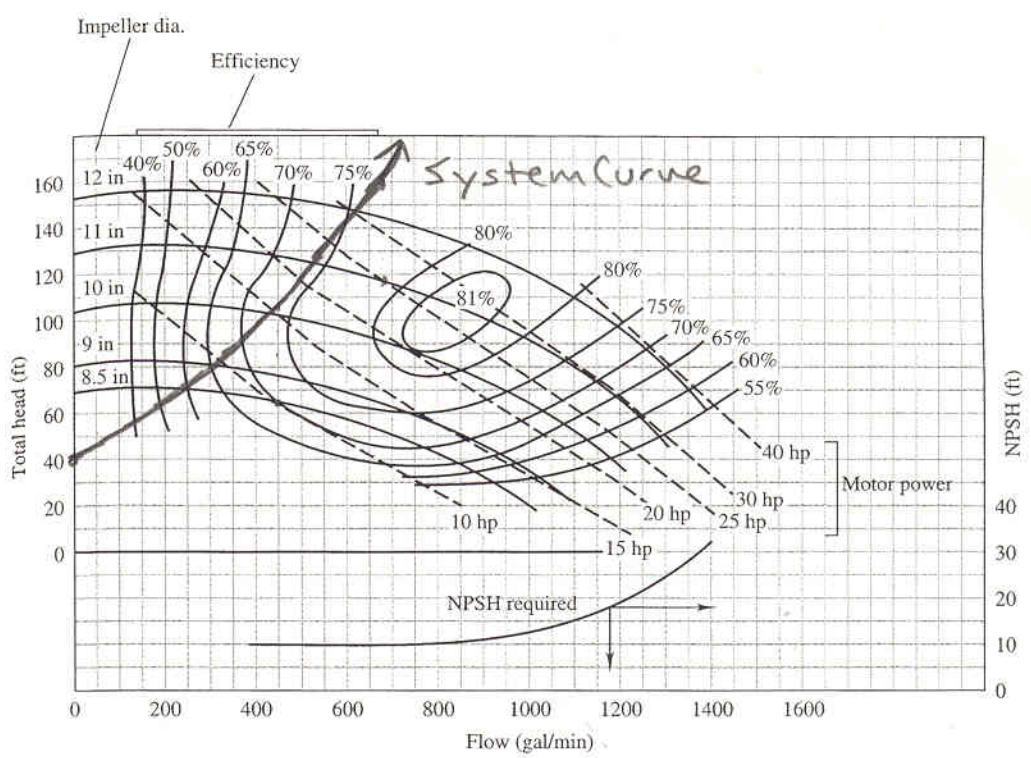
Plotting a System Curve
As the preceding discussion showed, system curves have a flow-independent component (of inlet/outlet pressure) and a flow-dependent component that varies with the square of flow rate (of friction pressure).

A system curve for a closed-loop piping system with no inlet/outlet pressure difference is shown below. The curve is a parabola of the form $\Delta P_{headloss} = C_2 V^2$. The curve passes through the origin because the inlet/outlet pressure difference, sometimes called the static head, is zero. The coefficient C_2 can be determined if the operating point is known by substituting the known pressure drop and flow rate into the equation and solving for C_2. The fluid work required to push the fluid through the pipe is the product of the volume flow rate and system pressure drop and is represented graphically by the area under the rectangle defined by the operating point.



Source of original pump curve: Kreider and Rabl, 1996.

A system curve for an open-loop piping system with a “static” or “inlet/outlet” pressure of 40 ft is shown below. This system curve is of the form $P_{headloss} = A + C_2 \; V^2$; where A is the “static” or “inlet/outlet” pressure drop. As before, the coefficient C_2 can be determined if the operating point and inlet/outlet pressure are known by substituting the known values into the equation and solving for C_2.

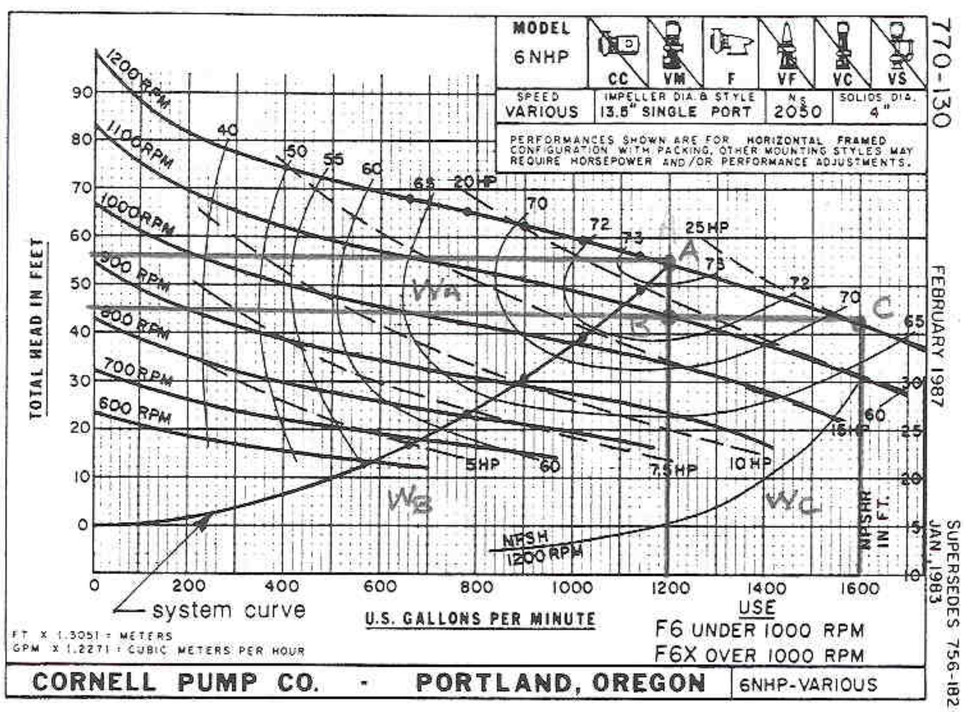


Source of original pump curve: Kreider and Rabl, 1996.

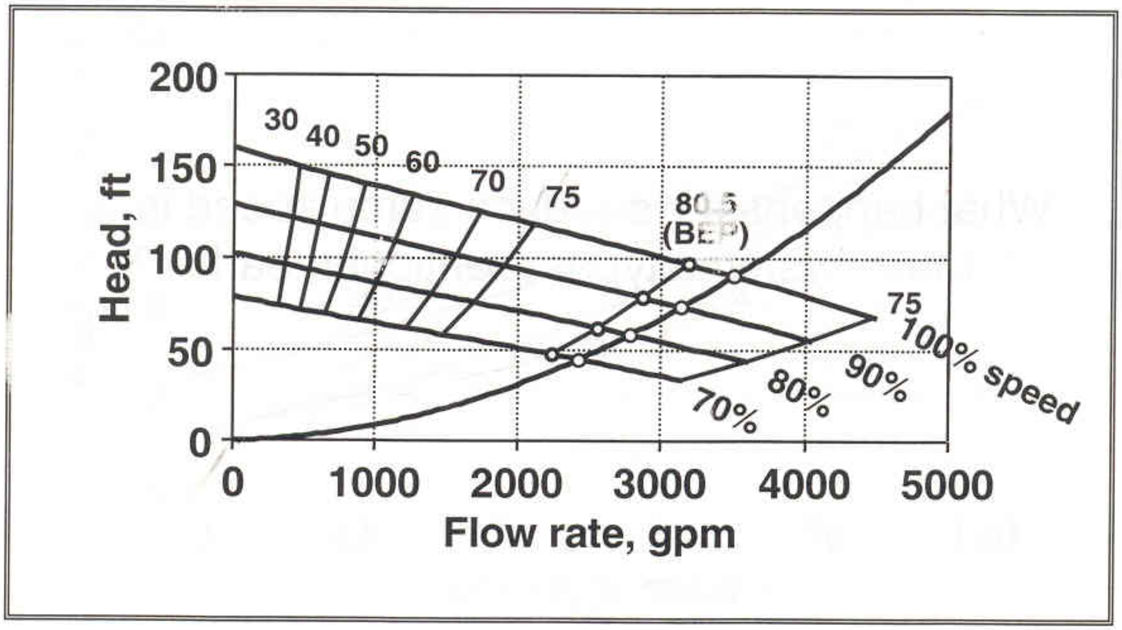
Variation of Pump Efficiency With Impeller Diameter and Rotational Speed

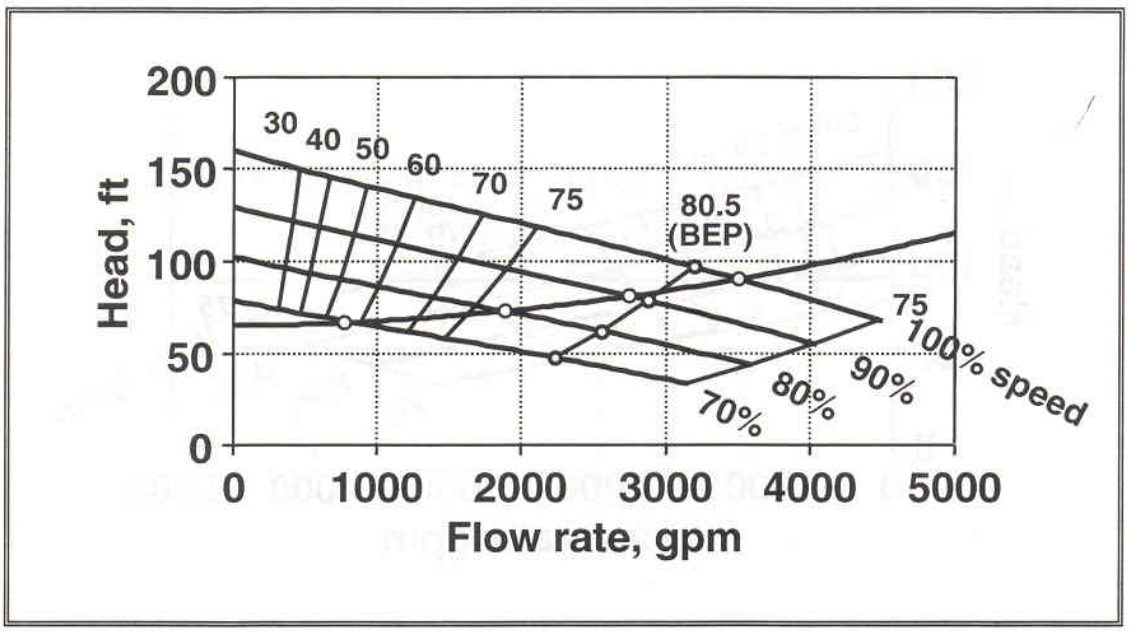
Pump efficiency is greatest when the largest possible impeller is installed in a pump casing. Pump efficiency decreases when smaller impellers are installed in a pump because of the increased amount of fluid that slips through the space between the tips of the impeller blades and the pump casing. The pump performance charts shown above clearly show the decrease in pump efficiency with smaller impellers.

Pump efficiency also decreases as the rotational speed of a pump is reduced. However, the magnitude of the decrease in pump efficiency depends on the individual pump. For example, in the pump performance chart shown below, pump efficiency declines from about 75% at full speed to about 55% at half speed while following the system curve with zero static head.



For other pumps, the magnitude of the decrease in pump efficiency may be negligible. For example, in the pump performance chart shown below, pump efficiency at variable speeds remains constant for pumping systems following a system curve with zero static head, but decreases for pumping systems with high static head.





Source: "Pumping Systems Field Monitoring and Application of the Pumping System Assessment Tool PSAT", U.S. Department of Energy, 2002.

In general, the "Pumping Systems Field Monitoring and Application of the Pumping System Assessment Tool PSAT" (U.S. Department of Energy, 2002) states that pump efficiency at variable speeds remains approximately constant for pumping systems following a system curve with zero static head.

Multiple Pumps Operating In Parallel

Many pumping systems employ multiple pumps in parallel rather a single large pump. One advantage of specifying multiple pumps in a parallel configuration is redundancy in case of failure. For example, it is common to design a pumping system with three pumps in parallel configuration, even though no more than two pumps would ever run simultaneously. The third pump provides redundancy in case of failure, and allows the system to function at full capacity even when one pump is being serviced. Another advantage of parallel pumping configuration is the ability to vary flow by turning one or more of the pumps on and off. Finally, in many applications, it is more energy efficient to operate multiple smaller pumps in parallel rather than operating a single large pump.

When two pumps are operated in parallel, they perform like a single pump with twice the flow rate at the same pressure drop. The figure below shows the pump curve of a single pump A, two pumps operating in parallel B, and the system curve C.

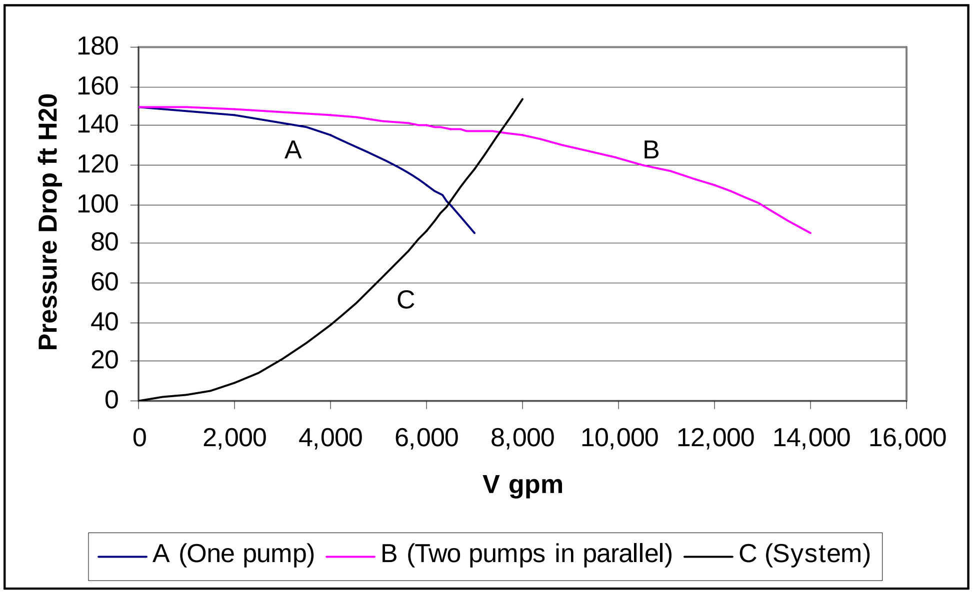


Figure 5.1.3 Pump and systems curves for secondary chilled water loop.

The system curve C describes the relationship of pressure drop and flow rate for the given piping system with no static head. Because pressure drop varies with the square of flow rate, the equation of the system curve can be estimated by fitting a quadratic equation through the origin and the design operating point:

$$\Delta P \text{ (ft H20)} = (2.4 \times 10^{-6}) \times V^2$$
(1)

where V is the flow rate in gpm. The equation for curve B can be estimated by the fitting a regression equation through the data points on the curve:

ΔP (ft H20) = $149 + 0.00106 \times V + (3.65 \times 10^{-7}) \times V^2$
(2)

The operating point of curve B for two pumps in parallel can be found from Equations 1 and 2 to be about:

ΔP = 135 ft H_20 V = 7,500 gpm

Note that the total volume flow rate of two pumps operating in parallel is less than twice the flow rate of a single pump operating alone (at the intersection of C and A).

Pump Affinity Laws

The fundamental fluid mechanic relationships developed thus far can be modified to generate other useful relations between pump/fan parameters. These relationships are known as pump/fan affinity laws. The two most important relationships are derived below.

The friction headloss is:

$$\Delta P_{headloss} = \rho \left[(k_f \underline{V}^2 / 2) + ((f\ L\ \underline{V}^2) / (2\ D)) \right]$$

For this relation, it is evident that for a given pipe/duct system, friction headloss is proportional to the square of velocity and (assuming incompressible flow) the square of the volume flow rate.

$$\Delta P_{headloss} = C_1 \underline{V}^2 = C_2 V^2$$

By substitution, fluid work is proportional to the cube of volume flow rate

$$W_f = V \Delta P_{headloss} = V C_2 V^2 = C_2 V^3$$

Since W_f / V^3 is constant, it follows that:

$$(W_f / V^3)_1 = C = (W_f / V^3)_2$$

$$W_{f2} = W_{f1} (V_2 / V_1)^3$$

This relation shows that a small reduction in the volume flow rate results in a large reduction in the fluid work. For example, reducing the volume flow rate by one half reduces fluid work by 88%!

$W_{f2} = W_{f1} (1/2)^3 == W_{f1} (1/8)$
$(W_{f1} – W_{f2}) / W_{f1} = [W_{f1} - W_{f1} (1/8)] / W_{f1} = 1 – (1/8) = 88\%$

Another useful relation can be derived from the relationship between volume flow rate V and the rotational speed of the pump fan. In centrifugal pumps and fans, the volume flow rate is proportional to the rotational speed of the pump fan.

$V = C\ RPM$

Since V/RPM is constant, it follows that:

$(V / RPM)_1 = C = (V / RPM)_2$
$V_2 = V_1\ (RPM_2 / RPM_1)$

Thus, volume flow rate varies in proportion to pump/fan speed.

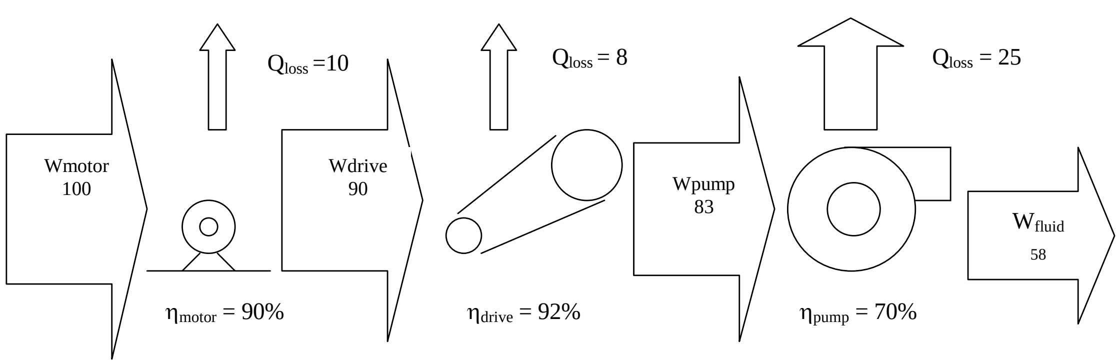
Pump Motor Work

Pumps and fans are typically driven by electrical motors. The power required by the motor is greater than the fluid work because the pump/fan, power transmission and motor all incur losses. Thus, fluid work must be divided by the product of the efficiencies of all components of the pump energy-delivery system to determine the electricity required by the motor.

$W_{elec} = W_f / (\ Efficiency_{pump.fan} \times Efficiency_{drive} \times Efficiency_{motor})$

A dimensional version of this equation, using U.S. units for pumping water at standard conditions, is:

$W_{elec}\ (hp) = V\ (gal/min)\ \Delta P_{total}\ (ft\text{-}H_20) / (3{,}960\ (gal\text{-}ft/min\text{-}hp) \times Eff_{pump} \times Eff_{drive} \times Eff_{motor})$



For example, if the efficiency of the motor at converting electrical energy to motor shaft work is 90%, the efficiency of belt drives at transferring motor shaft work to pump is 92%, and the efficiency of a pump at converting pump shaft work to fluid work is 75%, the electrical energy use required by the motor would be 73% greater than the required fluid work.

$W_{elec} = W_f / (\ 90\% \times 92\% \times 70\%) \quad = 1.73\ W_f$

Electronic Variable Speed Drives

Electronic variable speed drives (VSDs) control the speed of AC motors by converting the frequency and voltage of the AC line supply from fixed to variable values. VSDs are used in both constant and variable torque applications. In variable torque applications, such as pumping and fan systems, slowing the motor speed reduces the torque on the motor and can result in significant energy savings. These savings can be estimated using the Pump/Fan Affinity Laws.

VSDs subject motors to voltage spikes and fast voltage rise and fall times. These voltage spikes can “punch” through traditional winding insulation. Because of this, VSDs should only be coupled to motors that the manufacturer specifies as suitable for PWM VSDs. If a motor is to be rewound, be sure to specify rewinding characteristics for PWM VSD motors. Most energy-efficient motors are suitable PWM VSDs.

In some applications, it may be possible to simply reduce the flow to a fixed rate rather than vary it continuously. In these cases, slowing the pump by increasing the diameter of the pump pulley or decreasing the diameter or the motor pulley would generate the same savings.

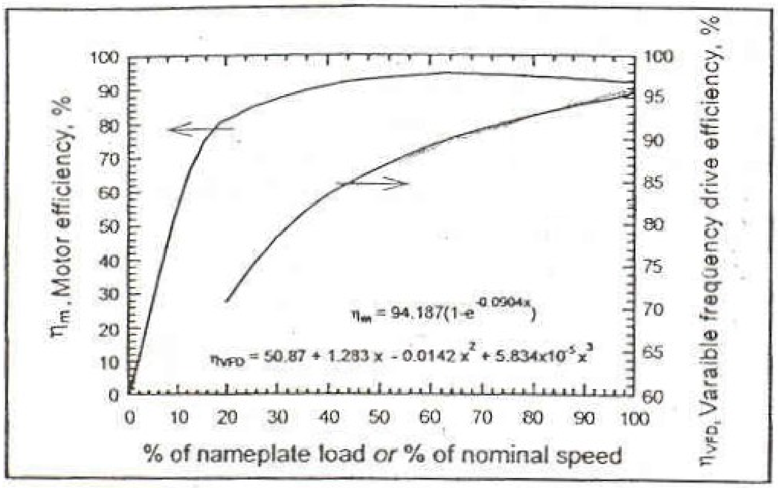
Because VSDs work best with premium efficiency motors, the motor may need to be upgraded if it is not a premium efficiency motor. Typical installed costs of VSDs are shown below.

Motor Size	Cost of VSD
5 hp	$ 1,150
10 hp	$ 1,775
30 hp	$ 3,500
50 hp	$ 5,600
75 hp	$ 9,200
250 hp	$16,900

Source: C&E Sales. Dayton, Ohio (10/1999)

VSD and Motor Efficiency at Part Load

Typical motor and VSD efficiency curves at part load are shown below (“Pumping Energy and Variable Frequency Drives”, Bernier and Bourret, ASHRAE Journal, December 1999).



Motor and VSD Efficiency as a Function of Percent of Motor Nameplate Load
(Source: "Pumping Energy and Variable Frequency Drives", Bernier and Bourret, ASHRAE Journal, December 1999)

Using these relationships, motor and VSD efficiency can be approximated as:

$\eta_{motor} = 94.187(1 – e^{-0.0904\ pl})$

$\eta_{VSD} = 50.87 + 1.283\ pl – 0.0142\ pl^2 + (5.834 \times 10^{-5})\ pl^3$

The electrical power to a motor with a VSD can be calculated using the following equation.

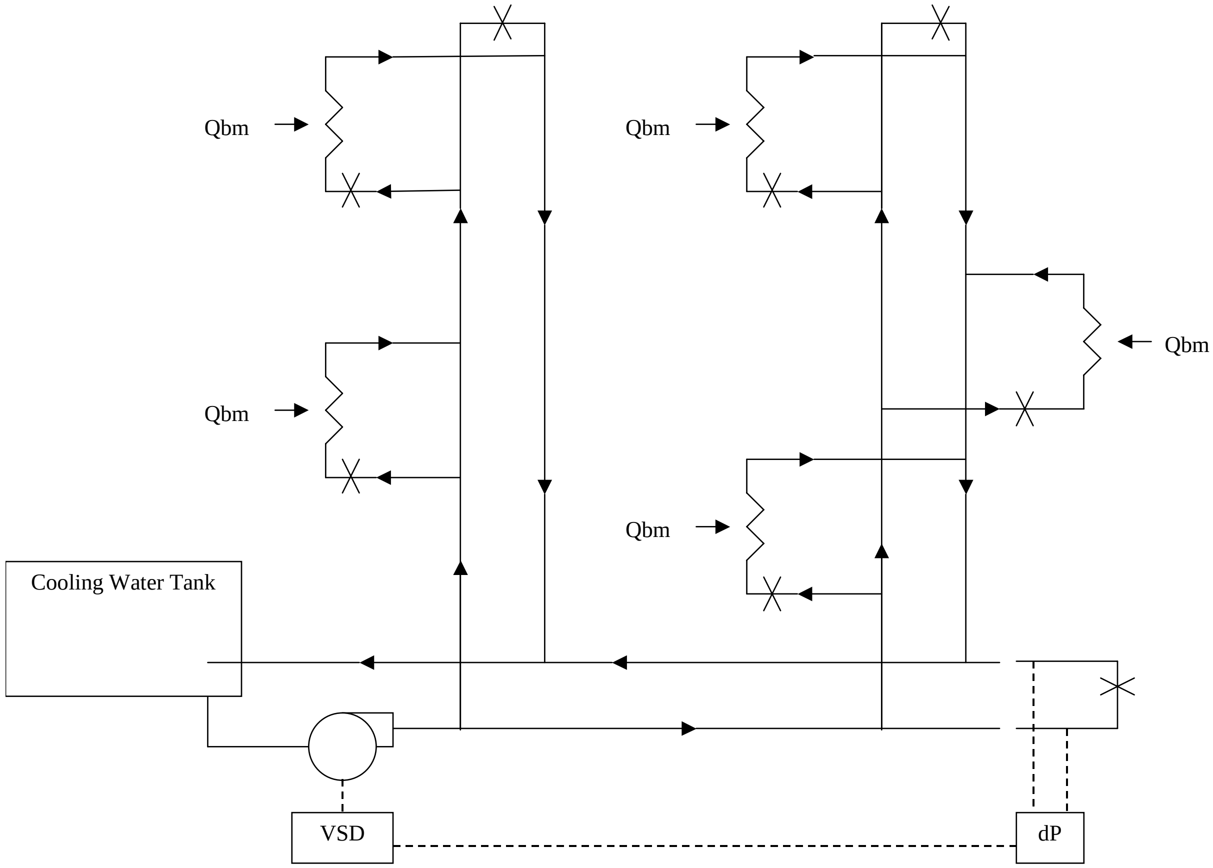
Power Input (kW) = (Power Output (hp) x 0.75 kW/hp) / ($\eta_{motor} \times \eta_{VSD}$)

<u>VSD Pumping Retrofits</u>

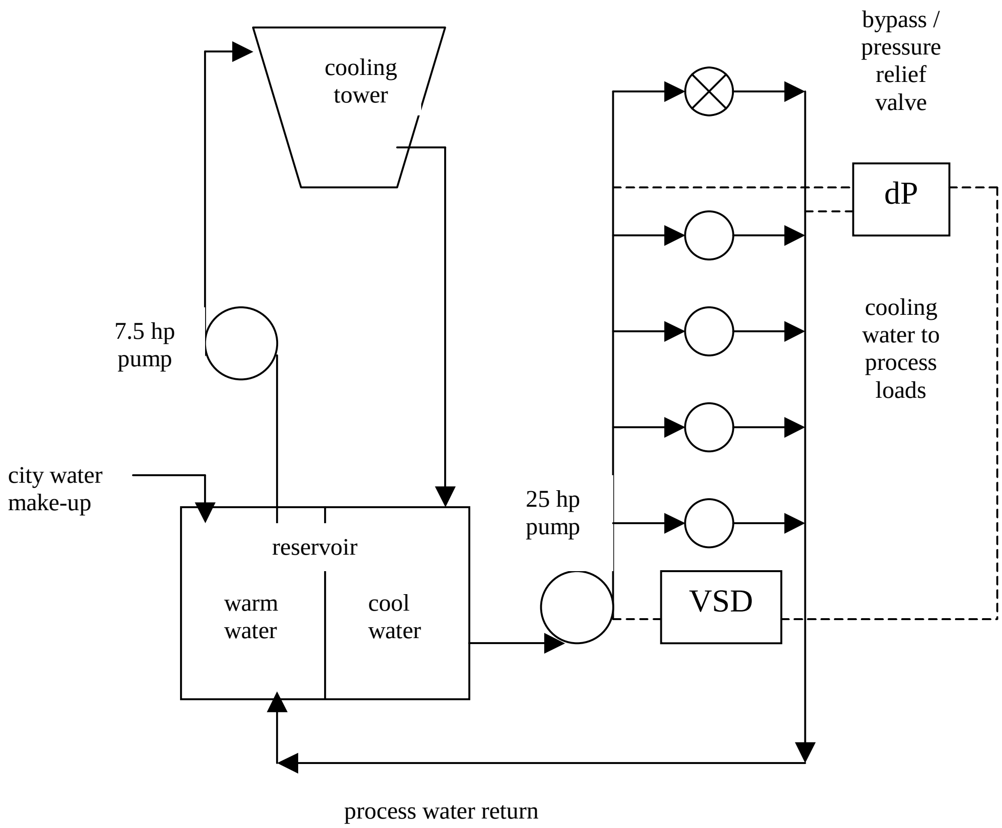
VSD pumping retrofits typically require making three changes to the existing pumping system:

1) Install a VSD on power supply to the pump motor. In parallel pumping configurations, one VSD is generally needed for each operational pump, but not for the backup pump.
2) Close valves all by-pass pipes.
3) Install a differential-pressure sensor between the supply and return headers at the process load located the farthest distance from the pump. Determine the pressure drop needed to guarantee sufficient flow through the farthest process load at this point. Control the speed of the VSD to maintain this differential pressure.

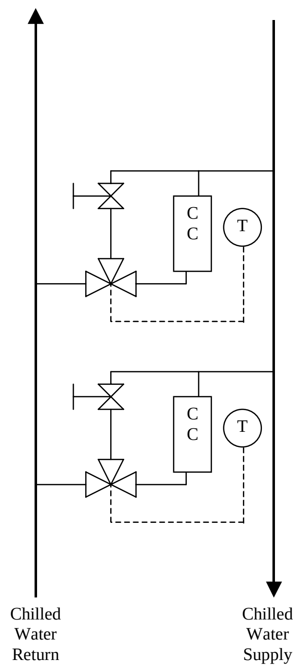
For example, the figure below shows a cooling loop with VSD retrofit. dP is a differential-pressure sensor which would control the speed of the VSD. The by-pass pressure relief valves at the end of each run would be closed. Flow through each process load would be controlled at the load.



The figure below shows another process cooling water loop with VSD retrofit. The by-pass pressure relief valve would be closed. Flow through each process load would be controlled at the load.



The following figure shows a schematic of the typical piping configuration at the air handler cooling coils in a constant-volume chilled-water supply system. The three-way valves direct chilled water either through the cooling coil or around the cooling coil via the bypass loop. The flow of chilled water through the cooling coils is varied to maintain the temperature of the air leaving the cooling coils at a constant temperature. In a VSD retrofit, the bypass valves would be closed, and a differential-pressure sensor would be installed between the supply and return headers at the air handler located farthest from the pump. In some cases, it may be necessary to replace the three-way valves with two-way valves if the three-way valves were not designed to handle larger pressure drops in a VSD situation.



Whole-System Inside-Out Approach to Low Energy Pump Systems

We have found that the most effective approach for designing low-energy pump/fan systems and for identifying energy savings opportunities in existing fluid flow systems is the “whole-system, inside-out approach”. The “whole-system” part of this approach emphasizes the importance of considering the entire conversion, delivery and end-use system. The “inside-out” part of the approach describes the preferred sequence of analysis, which begins at the point of the energy’s final use “inside” of the process, followed by sequential investigations the energy distribution and primary energy conversion systems. This approach can tends to multiply savings and result in smaller, more efficient and less costly systems.

<u>Reduce Elevation Head</u>

Many pumping applications involve lifting fluids from lower to higher elevations. The total pressure drop across the pump, and hence pumping power, includes this elevation pressure drop. In some applications, it may be possible to reduce the elevation pressure drop by increasing the height of the fluid in the supply tank or reducing the height of the fluid in the outlet tank.

<u>Reduce Friction Pressure Drop: Increase Pipe Diameter</u>

Friction headloss in internal flow is strongly related to the diameter of the pipe/duct. Small pipes and ducts dramatically increase the velocity of the fluid and friction headloss.

The friction headloss through pipes and ducts is:

$$\Delta P_{headloss} = \rho \; f \; L \; \underline{V}^2 \, / \, (2 \; D)$$

The velocity V̲ is the quotient of volume flow rate V and area A, thus

$\Delta P_{headloss} = \rho f L (V / A)^2 / (2 D) = \rho f L (V / 3.14 D^2)^2 / (2 D) = \rho f L V^2 / (2\ 3.14^2 D^5)$

Thus, friction loss through pipes is inversely proportional to the fifth power of the diameter

$\Delta P_{headloss} \sim C / D^5$

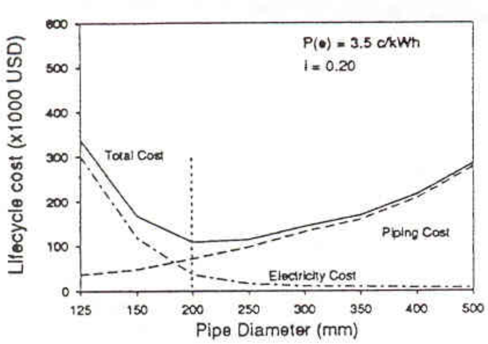
This means that doubling the pipe/duct diameter reduces friction headloss by about 97%! To check this, consider pumping 4 gpm of water through 0.5-inch and 1-inch diameter schedule 40 steel pipes. From the monogram:

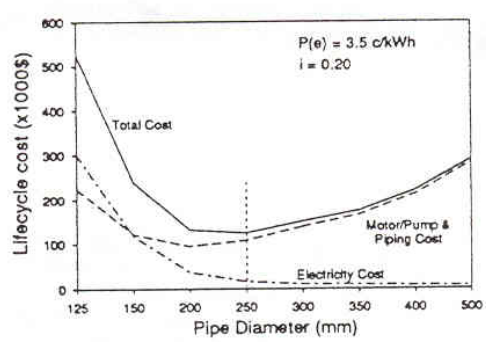
$\Delta P_{headloss, 0.5\text{-inch}}$ = 1.3 ft-$H_2$0/100 ft
$\Delta P_{headloss, 1\text{-inch}}$ = 17 ft-$H_2$0/100 ft

The percent reduction in friction headloss from doubling the diameter of the pipe would be about:

(17 – 1.3) / 17 = 92% !

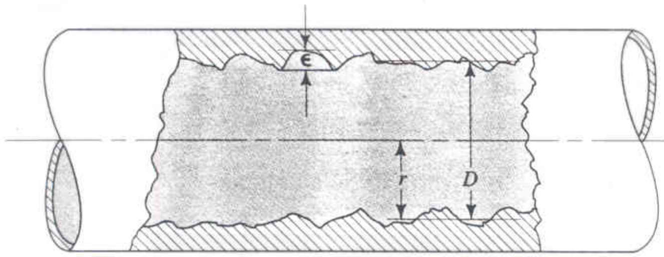
Optimum pipe diameter is often calculated based on the net present value of the cost of the pipe plus pumping energy costs. Using this method in the figure below (Larson, E.D. and Nilsson, L.J., 1991, "Electricity Use and Efficiency in Pumping and Air Handling Systems, ASHRAE Transactions, pgs. 363-377.) optimum pipe diameter was found to be 200 mm. When the cost of the pump was also included in the analysis, the optimum diameter was found to be 250 mm and energy use was reduced by 50%. This illustrates the importance of considering the whole system.





Reduce Friction Pressure Drop: Use Smooth Pipes and Ducts

Smoother pipes/ducts reduce friction headloss. The roughness e of common types of piping is shown below.



Material	Roughness, ε (m)	Roughness, ε (ft)
Glass, plastic	Smooth	Smooth
Copper, brass, lead (tubing)	1.5×10^{-6}	5×10^{-6}
Cast iron—uncoated	2.4×10^{-4}	8×10^{-4}
Cast iron—asphalt coated	1.2×10^{-4}	4×10^{-4}
Commercial steel or welded steel	4.6×10^{-5}	1.5×10^{-4}
Wrought iron	4.6×10^{-5}	1.5×10^{-4}
Riveted steel	1.8×10^{-3}	6×10^{-3}
Concrete	1.2×10^{-3}	4×10^{-3}

Source: Applied Fluid Mechanics, Mott, 2000

As an example, consider pumping water with Re = 100,000 through 4-inch plastic (smooth) and schedule 40 steel pipe.

ε/D_{steel} = 0.00015 ft / 0.3333 ft = 0.00045

From the Moody Diagram, f_{steel} = 0.31 and $f_{plastic}$ = 0.018. The percent reduction in friction headloss from the use of plastic pipe would be about:

(0.031 – 0.018) / 0.018 = 42%

Reduce Friction Pressure Drop: Use Low Pressure-Drop Fittings

Minimizing fittings, including turns, and the use of low-pressure drop fittings can significantly reduce friction headloss. Consider for example, the table below. The use of fully-open gate valves instead of globe valves reduces the friction headloss through the valve by 98%. Similarly, the use of swing type check valves instead of butterfly valves reduces the friction headloss through the valve by 33%, and long radius elbows reduce the friction headloss by 50% compared to standard radius elbows.

Type	Equivalent Length in Pipe Diameters, L_e/D
Globe valve—fully open	340
Angle valve—fully open	150
Gate valve—fully open	8
—¾ open	35
—½ open	160
—¼ open	900
Check valve—swing type	100
Check valve—ball type	150
Butterfly valve—fully open	45
90° standard elbow	30
90° long radius elbow	20
90° street elbow	50
45° standard elbow	16
45° street elbow	26
Close return bend	50
Standard tee—with flow through run	20
—with flow through branch	60

Note Huge Difference!

Source: Crane Valves, Joliet, IL.

where $k_f = (Le/D)\ f_t$

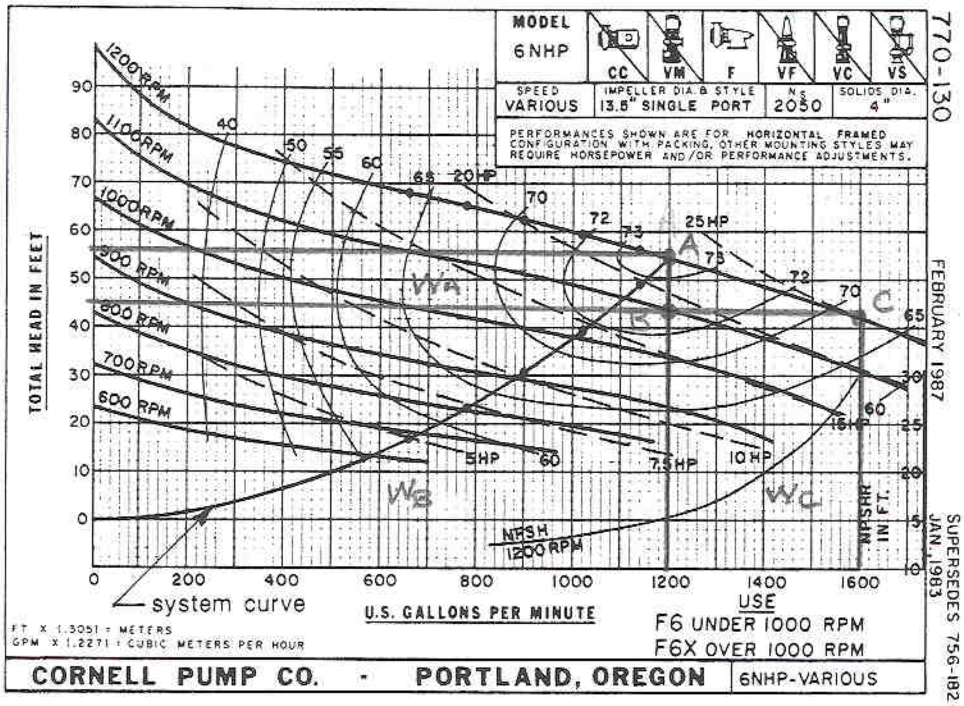
Steel Pipe Diameter (inches)	f_t
½	0.027
1	0.023
2	0.019
4	0.017
6	0.015
8-10	0.014
12-16	0.013
18-24	0.012

Source: Applied Fluid Mechanics, Mott, 2000

<u>Modify Pump To Realize Savings From Reducing Friction Pressure Drop</u>

Reducing pressure drop without modifying the pump generally increases the volume flow rate and pump energy consumption. Thus, it is important to reduce the diameter of the pump impellor or slow the pump to take advantage of the reduced pressure drop, as the following example demonstrates.

The figure below shows pump performance at two system pressure drops. Reducing the system pressure drop from 55 ft-H_20 at point A to 45 ft-H_20 without altering the pump impellor or speed would cause the pump to operate at point C. The power required to pump a fluid is the product of the volume flow rate and pressure drop; hence, the areas enclosed by the rectangles defined by each operating point represent the fluid power requirements, W_A and $W_{C,}$ at the different system pressure drops.



Source: Nadel et al., 1991

W_A = 1,200 gpm x 55 ft-H_20 / 3,960 gpm-ft-H_20/hp = 16.7 hp
Wc = 1,600 gpm x 45 ft-H_20 / 3,960 gpm-ft-H_20/hp = 18.2 hp

The power, P, required by the pump is the fluid power requirements divided by the pump efficiency.

P_A = 16.7 hp / .74 = 22.6 hp
Pc = 18.2 hp / .68 = 26.8 hp

Thus, decreasing system pressure drop without altering the pump impellor or speed would cause the pump to consume more energy.

Savings = $P_A - P_C$ = 22.6 hp – 26.8 hp = -4.2 hp

To realize energy savings from reducing pressure drop, it is necessary to slow the pump speed or decrease the size of the impellor. To determine the pump speed required to deliver the initial flow of 1,200 gpm with the new low-pressure drop pipe system, it is necessary to develop a system curve for the new pipe system. Pressure drop through piping systems varies with the square of flow rate. Thus, the equation for a system curve can be written as:

$dP = C V^2$

The coefficient, C, for the new system curve can be found by substituting the values of pressure drop and volume flow rate for point C.

$C = dP / V^2 = 45 / 16^2 = 0.1758$

Thus the pressure drop through the new duct system at 1,200 gpm would be about:

$dP = C\ V^2 = 0.1758\ 12^2 = 25$ ft-H_20

According to the pump curves, the pump would deliver 1,200 at 25 ft-H_20 if the pump speed were slowed to about 900 rpm. At this operating point, the power delivered to the pump would be about 11 hp. Thus, the savings from reducing the pressure drop in the pipe system, if the pump speed were modified, would be about:

Savings = $P_A – P_C$ = 22.6 hp – 11 hp = 11.6 hp

Energy-Efficient Flow Control

Most pump systems are designed to handle peak conditions. Since peak conditions typically occur infrequently, substantial energy savings are possible by controlling fluid flow rate to match actual demand. The inside-out approach to low-energy pump systems recommends reviewing all end-use applications to determine the minimum required flow, before proceeding "upstream" with the analyses of the piping and pumping systems.

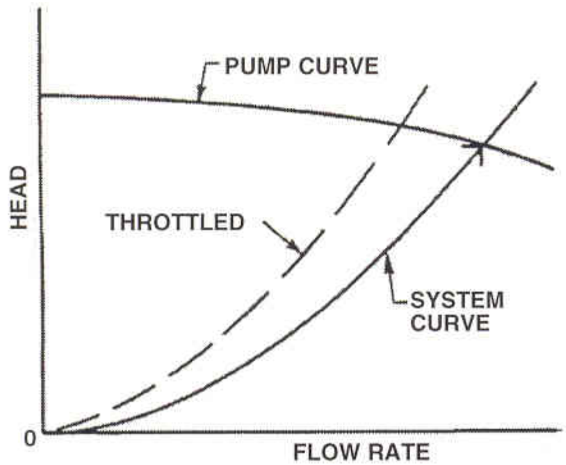
Once the minimum required flow is determined, it is necessary to determine how the flow is currently controlled and consider more energy-efficient methods of flow control. Common methods of flow control are:

- Bypass
- Throttling
- Intermittent and multiple pump operation
- Trimming impellors
- Slowing pump rotational speed by increasing pulley diameter
- Continual variation of pump rotational speed by a variable speed drive.

Bypass and throttling and are energy inefficient compared to the other methods and immediately signal the potential for cost-effectively reducing energy costs. Intermittent and multiple pump operation are relatively energy efficient; however, pumping energy use can sometimes be reduced by pumping at low flow rates for longer periods rather than pumping at high flow rates for shorter periods of time. When the required flow rate is constant and less than the current capacity of the pump, trimming pump impellers and slowing pump rotational speed by changing pulley sizes are typically highly cost-effective due to their relatively low implementation costs. Energy use in variable flow applications can frequently be substantially and cost-effectively reduced through use of variable speed drives.

Bypass: Many processes use constant speed pumps with variable process loads. Valves are opened or closed to direct water through the process or through a bypass loop. Thus, the flow of water through the pumps remains nearly constant even as the flow of water through the process varies. Bypass is the least efficient method of flow control, since pump power remains nearly constant even as the load varies.

Throttling: Controlling flow by closing a flow-control valve downstream of the pump increases pressure drop and causes the operating point to move up and left on the pump curve.



Source: Gould Pumps, GPM 7-CD, Technical Information.

This results in relatively small energy savings, since

$Wf_2 = V_2\ \Delta P_2$ where $V_2 < V_1$ but $\Delta P_2 > \Delta P_1$

Thus, throttling is an energy inefficient method of flow control.

To quantify specific energy savings potential from replacing throttling with other methods of flow control, it is frequently useful to quantify the pressure drop or flow through a throttling valve. Cv is a coefficient that relates volume flow rate to pressure drop across a valve according to the following valve equation.

$$\dot{V}(gpm) = Cv\sqrt{\Delta P(psi)}$$

The two most common types of throttling valves are high performance butterfly valves (HPBV) and V-ball valves. The tables below show Cv values for typical HPBV and V-ball valves at different rotation angles.

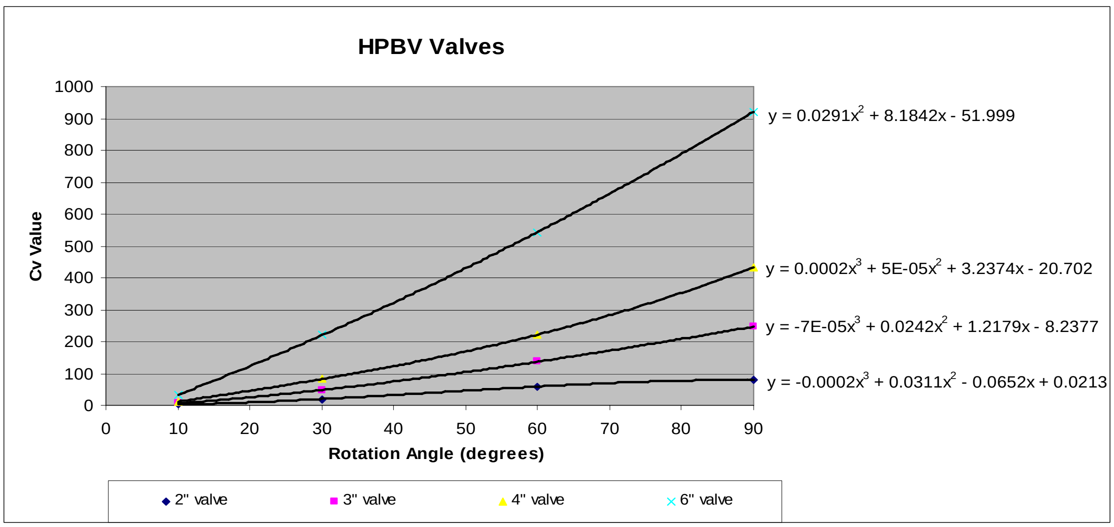
Table P2: HPBV Cv Values Table P3: V-ball Cv Values

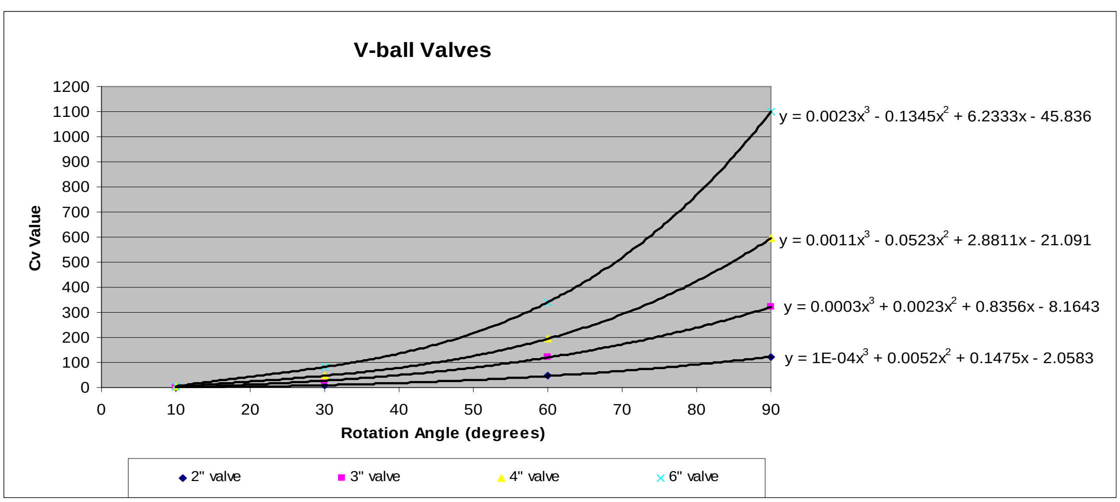
Valve Size	Rotation Angle (degrees)			
	10	30	60	90
2"	2.25	19.9	58.9	80.2
3"	6.29	48.2	137	247
4"	11.9	82.5	222	434
6"	32.2	221	543	921

Valve Size	Rotation Angle (degrees)			
	10	30	60	90
2"	0.028	9.6	46.1	123
3"	0.746	27.7	120	321
4"	3.56	47.2	195	596
6"	5.34	82.1	340	1100

(Source: www.emersonprocess.com)

Using the data from these tables, regression equations to relate Cv values to rotation angle were created and are shown below.





Cv Versus Rotation Angle with Regression Equations for HPBV and V-ball Valves. 90^o rotation corresponds to 100% output.

Using these relations, Cv for a valve can be determined from the rotational angle of the valve. Next, pressure drop across the valve can be determined if the flow is known, or the flow can be determined if the pressure drop is known._

Intermittent Pump Operation ("Pump Long, Pump Slow"): In some applications, pumps may operate at a relative high flow rate for part of the time and then be turned off until needed again. Because friction losses are proportional to the square of flow, it is more energy-efficient to pump a lower volume flow rate for a longer period of time. We call this the "Pump Long, Pump Slow" principle. "Pump long, pump slow" opportunities may exist whenever pumps run intermittently. If a single pump operates intermittently, then application of the "pump long, pump slow" principal would require installing a smaller pump, trimming the impeller or slowing the pump rotational speed. If multiple pumps operate in parallel, it may be possible to simply run fewer pumps more continuously.

Slowing pump rotational speed by increasing pulley diameter: Reducing flow by installing a smaller diameter impeller or slowing pump speed results in relatively large energy savings, since:

$$Wf_2 = Wf_2 \quad (V_2 / V_1)^3$$

The following example compares energy savings from reducing flow with a flow-control valve to energy savings from reducing flow by slowing pump speed. If the transmission between the pump and motor uses belts and pulleys, pump speed can be slowed by changing the increasing the diameter of the pump pulley. Alternately, pump speed can be slowed and varied with a variable speed drive (VSD), which alters the frequency of current to the motor. The savings from reducing impellor diameter are similar to the savings from reducing pump speed.

The figure below shows pump performance at various speeds and a system curve. Assume the original operating point, A, is 1,200 gpm at 55 ft-H_20. According to the chart, the required power to the pump at this operating point is about 23 hp. Alternately, pump power could be calculated as:

$$W_A = 1{,}200 \text{ gpm x } 55 \text{ ft-}H_20 / (3{,}960 \text{ gpm-ft-}H_20\text{/hp x } 0.74) = 22.6 \text{ hp}$$

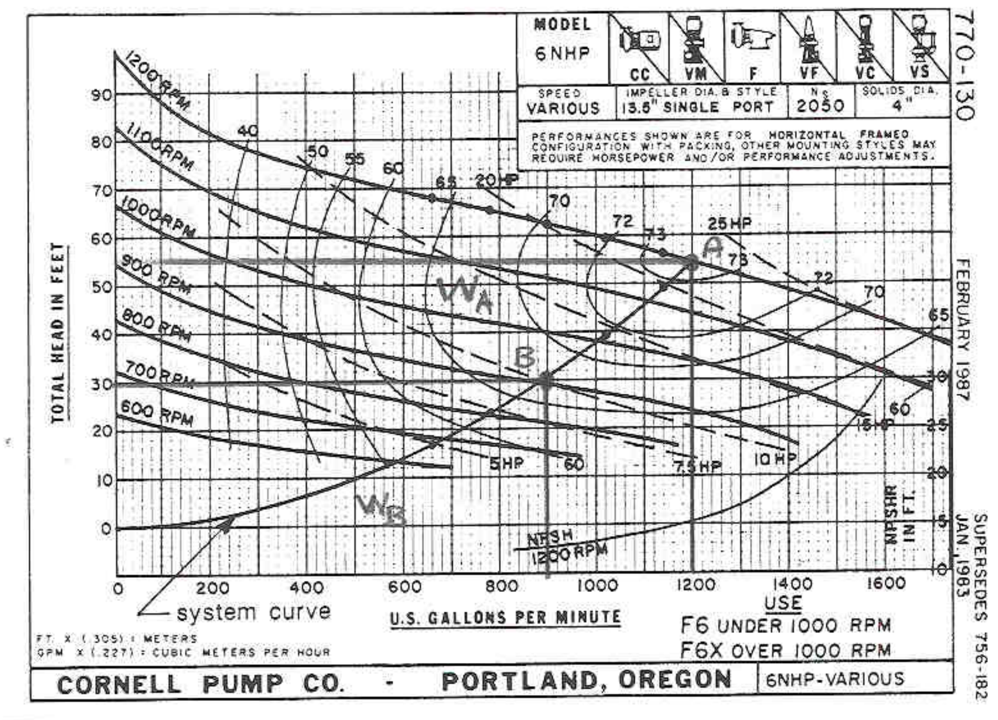
If the flow were reduced to 900 gpm with a flow control valve, the operating point would move along the pump curve to 900 gpm at 62 ft-H_20. According to the chart, the required power to the pump at this operating point is about 20 hp. Alternately, pump power could be calculated as:

$$W = 900 \text{ gpm x } 62 \text{ ft-}H_20 / (3{,}960 \text{ gpm-ft-}H_20\text{/hp x } 0.70) = 20.1 \text{ hp}$$

Thus, the pump power savings from reducing flow from 1,200 gpm to 900 gpm with a flow-control valve would be about:

22.6 hp – 20.1 hp = 2.5 hp

Alternately, the flow could be reduced from 1,200 gpm to 900 gpm by slowing the pump speed with a VSD. Reducing the pump speed from 1,200 rpm at point A to 900 rpm at point B would reduce the volume flow rate from 1,200 gpm to 900 gpm. The reduced volume flow rate would also generate less friction, and the system pressure drop would be reduced from 55 ft-H_20 to 30 ft-H_20. The power required to pump a fluid is the product of the volume flow rate and pressure drop; hence, the areas enclosed by the rectangles defined by each operating point represent the fluid power requirements, W_A and $W_{B,}$ at the different flow rates.



Source: Nadel et al., 1991

W_A = 1,200 gpm x 55 ft-H_20 / 3,960 gpm-ft-H_20/hp = 16.7 hp
W_B = 900 gpm x 30 ft-H_20 / 3,960 gpm-ft-H_20/hp = 6.8 hp

The power, W_B, required by the pump at point B can be read from the chart to be about 10 hp. Alternately, the power could be calculated as:

W_B = 900 gpm x 30 ft-H_20 / (3,960 gpm-ft-H_20/hp x 0.67) = 10.1 hp

Pump power savings would be the difference between P_A and P_B.

Savings = $P_A - P_B$ = 22.6 hp – 10.1 hp = 11.5 hp

When estimating power savings from reducing the volume flow rate, engineers frequently rely on pump affinity laws. Theoretically, pump work varies with the cube of volume flow rate. Use of the cubic relationship would predict:

$P_B = P_A (V_B/V_A)^3$ = 22.6 hp x (900 gpm / 1200 gpm) ^{3} = 9.5 hp

The 9.5 hp predicted by the pump-affinity law is less than the 10.1 hp predicted by the pump curve. This example demonstrates how use of the cubic relationship typically exaggerates savings. In practice, the efficiencies of the VSD, pump and motor typically decline as flow rate decreases, resulting in slightly less savings than would be predicted using this 'cubic' relationship. Thus, we conservatively estimate that pump/fan work varies with the square of flow rather than the cube of flow. Using this relationship, if we measured P_A to be 22.6 hp at 1,200 gpm, we would estimate P_B for 900 gpm to be about:

$P_B = P_A (V_B/V_A)^2$ = 22.6 hp x (900 gpm / 1200 gpm) ^{2} = 12.7 hp