System Curves

Tutorial

A **system curve** represents the relationship between flow through a system and the hydraulic losses at that flow. The system curve consists of two parts: friction and static head. A system curve is generated by varying the flow rate through the system from zero flow to some maximum value.

Static Head

Static head consists of both the elevation and pressure difference between the supply and destination of the system. This, typically, does not depend on velocity and is therefore constant for the system curve. This can be calculated using the following equation:

=+=

$$\Delta h_{stat} = \Delta(z_{destination} - z_{supply}) + \frac{(P_{destination} - P_{supply})}{\rho g}$$

=+=

where:

- z is elevation
- P is pressure
- is fluid density
- g is gravitational acceleration

Note that if the supply and destination are at the same pressure, as is often the case when they are open tanks, then the static head is simply the difference in the liquid elevation.

Frictional Head (Major Losses)

The **head loss due to friction** will vary based on flow rate (velocity) and can be calculated for the system components, such as piping, valves, elbows and bends, and end-use equipment, etc. These losses typically vary proportional to the square of the velocity.

Frictional head losses in pipes can be calculated using the Darcy-Weisbach equation. The Darcy-Weisbach friction factor, *f*, can be determined using the Colebrook-White equation (defined in the general fluid flow section).

These equations will approximate the Moody diagram. The friction factor is based on the Reynolds Number (Re), the pipe diameter (Dh), and the pipe roughness ( ). The pipe roughness is dependent on the type of pipe being used. Other aspects, such as age, fouling, and coatings will also affect the pipe roughness.

The Hazen-Williams equation is another method to determine pipe losses. These values are only valid for water and do not account for temperature or viscosity. These values are a function of pipe material only and are not dependent on Reynolds Number. A table of typical values for various pipe materials can be found here.

Minor Losses

Minor losses in a piping system can consist of valves, bends, elbows, area changes, entrances from and exits to equipment, tees and branches, etc. Anything that will obstruct or change the flow and pressure can be considered a minor loss. These are categorized differently than the pipe frictional loss (or major loss).

The loss created by the component is often characterized by a constant, K. The head loss is usually proportional to the square of the fluid velocity and can be determined by the equation defined in the general fluid flow section (K values for some types of components are also tabulated here).

System Curve

The system curve represents the sum of the static, piping and minor head losses over a range of flow rates and can be represented using the following equation:

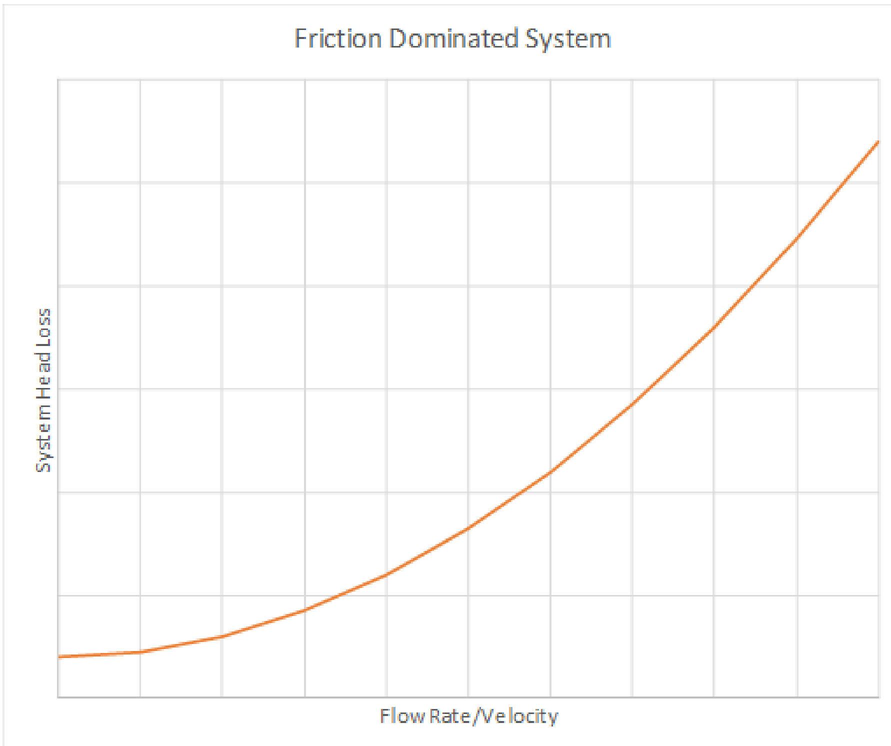
=+=

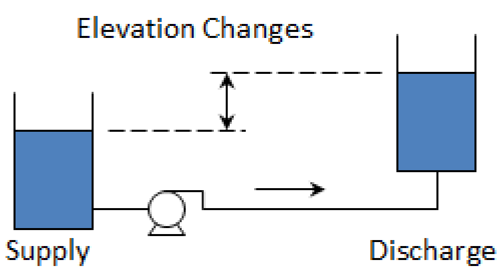
$$\Delta h_{system} = (z_{destination} - z_{supply}) + \frac{(P_{destination} - P_{supply})}{\rho g} + (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

=+=

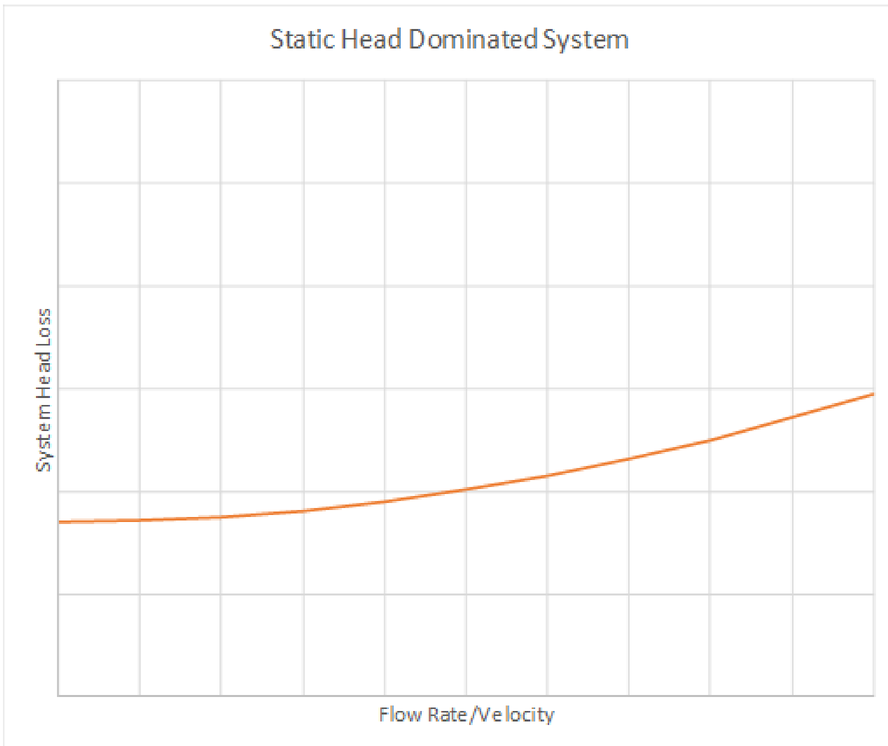
Shape of the System Curve

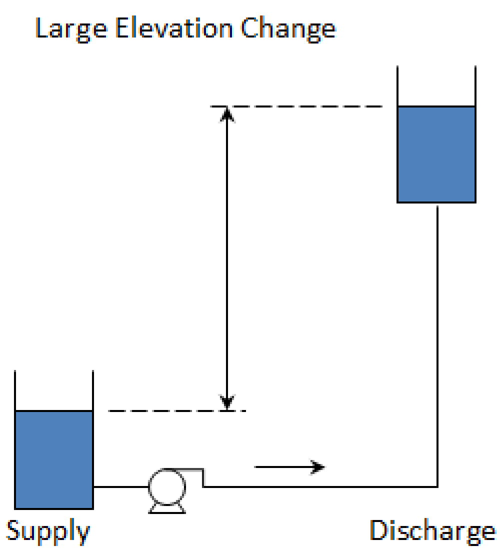
In some systems the frictional losses will be dominate part of the overall head loss. These systems will have a steeper system curve.





In other systems the elevation change, or static head, will be dominate part of the overall head loss. The system curve in this case will start at a higher value at zero flow and will tend to be flatter.





It is important to accurately characterize the system curve to select the correct pump for various operating conditions as the operating point of your system will be dependent on the intersection between the system curve and the pump curve.

System Curve Application

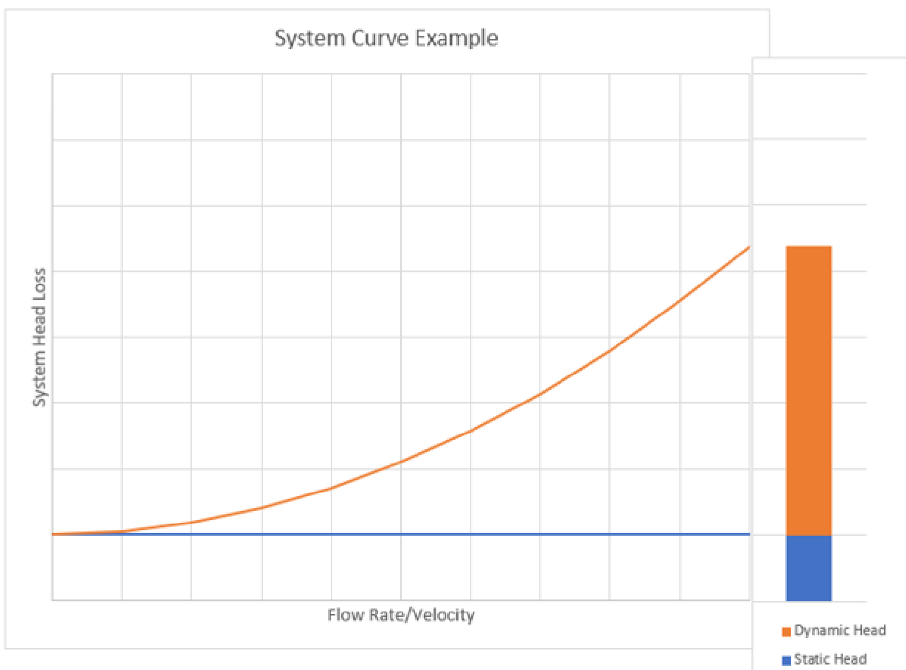
Real-world applications tend to consider a range or family of system curves. This would bracket the range of liquid levels, operating pressures, valve arrangements, etc.

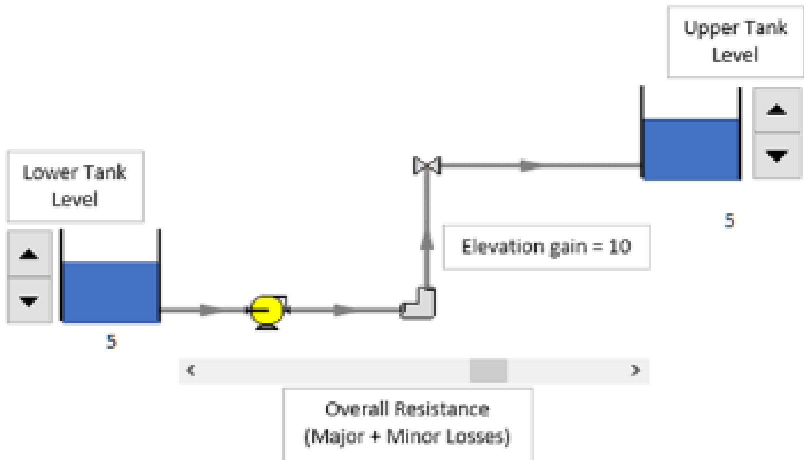
Educational Demonstration

(Demonstrator will be placed here)

This education demonstrator will show how changing the static and frictional losses changes the system curve.

The static head can be varied by changing the supply and destination levels. The overall frictional loss (piping and minor) can be varied by moving the slider. This is a representative demonstration and does not have actual values.





Worked Example (U.S. & Metric Units)

Determine the Static Head

Since both tanks have the same surface pressure, the static head is only dependent on the difference in surface elevation.

=+=

$$\Delta h_{stat} = (z_{destination} - z_{supply})$$

=+=

=+= [units = us]

$$\Delta h_{stat} = (289\,ft - 24\,ft) = 265\,ft$$

=+=

=+= [units = metric]

$$\Delta h_{stat} = (88.09\,m - 7.315\,m) = 80.77\,m$$

=+=

Determine the Pipe Friction and Properties

To simplify this example, we will consider the friction factor to be constant at 0.02. In general, the friction factor would vary as the flow rate (velocity) varies. Additionally, the flow would be laminar for low velocities. These considerations should be taken into account when calculating the pipe losses.

Determine the Minor or Component Loss

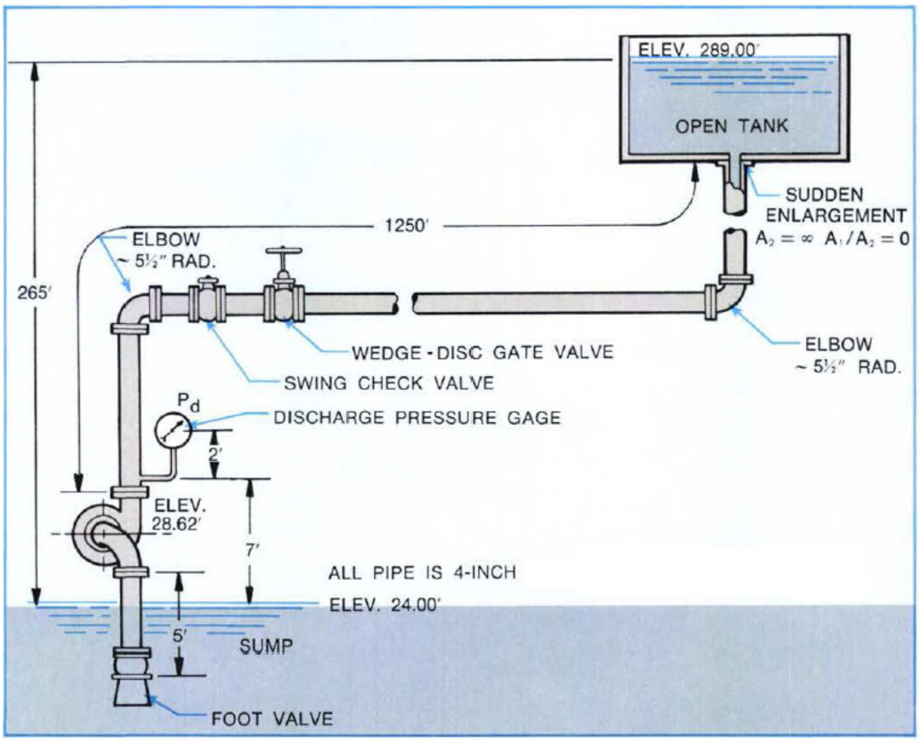

ELEV. 289.00′
OPEN TANK
SUDDEN
ENLARGEMENT
$A_2 = \infty$ $A_1/A_2 = 0$
1250′
ELBOW
~ 5½″ RAD.
265′
WEDGE - DISC GATE VALVE
SWING CHECK VALVE
ELBOW
~ 5½″ RAD.
P_d
DISCHARGE PRESSURE GAGE
2′
ELEV.
28.62′
7′
ALL PIPE IS 4-INCH
ELEV. 24.00′
5′
SUMP
FOOT VALVE

Figure 1:

The losses for the components can be found in tables. In this example we have the following:

- Regular flanged elbow (2), k = 0.31 each
- Swing check valve, k = 2.0
- Wedge-disc gate valve, k = 0.17
- Sudden enlargement, k = 1.0

This gives a total K factor equal to 3.79

Using the combined frictional loss equation above, we can determine the head loss (in feet) as a function of velocity

=+=

$$\Delta h_f = (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

=+=

=+= [units = us]

$$\Delta h_f = (\frac{0.02 * 1255ft}{0.3355ft} + 3.79) * \frac{v^2}{2 * 32.17ft/sec^2}$$

=+=

=+= [units = metric]

$$\Delta h_f = (\frac{0.02 * 382.52m}{0.10226m} + 3.79) * \frac{v^2}{2 * 9.81ft/sec^2}$$

=+=

=+= [units = us]

$$\Delta h_f = 1.22v^2$$

=+=

=+= [units = metric]

$$\Delta h_f = 4.01v^2$$

=+=

Determine the System Curve

The system curve can be calculated by varying the flow rate (velocity) using the above values. Combining the static and dynamic (pipe friction and minor losses) we have the following as a function of velocity.

=+=

$$\Delta h_{system} = \Delta h_{stat} + \Delta h_f$$

=+=

=+= [units = us]

$$\Delta h_{system} = 265ft + 1.22v^2$$

=+=

=+= [units = metric]

$$\Delta h_{system} = 80.77m + 4.01v^2$$

=+=

=+= [units us]

$$v = 0.002228 * Q * (\frac{4}{\pi D^2})$$

=+=

=+= [units metric]

$$v = 0.000278 * Q * (\frac{4}{\pi D^2})$$

=+=

Substituting this in for velocity and using the 4-inch pipe we can get the following as the system curve equation as a function of flow rate in gpm.

=+= [units us]

$$\Delta h_{system} = 265ft + (7.75E{-}04)Q^2$$

=+=

=+= [units metric]

$$\Delta h_{system} = 80.77m + (4.59E{-}03)Q^2$$

=+=

This, then, gives the following system curve data. This is a system that is dominated by the static head (there is a lift of

=/= title: Data data-us: datapoints_us.csv data-metric: datapoints_metric.csv x: 1 series: 3, 2 series_title_index: 0 =/=

=|= title: Data data-us: datapoints_us.csv data-metric: datapoints_metric.csv =|=

Pump Curves

Tutorial

A **pump performance curve** is a graphical representation of the head generated by a specific pump model at rates of flow from zero to maximum at a given operating speed.

Head and Flow Curve

The **head and flow curve**

=+= [units = us]

$$H = \frac{2.31 * p}{s}$$

=+=

=+= [units = metric]

$$H = \frac{0.102 * p}{s}$$

=+=

Specific gravity can be found by using the following equation, where   is density:

=+=

$$s = \frac{_{pumpedfluid}}{_{water}}$$

=+=

Using head, the performance of the pump can be shown independent of the density of the fluid pumped.

=/= title: Head vs. Flow data-us: pc-data.csv data-metric: pc-data-metric.csv x: 1 series: 2 series_title_index: 0 =/=

Efficiency Curve

Pump efficiency is shown as a percentage on most pump curves. Pump efficiency is defined by the equation below, where Pw is pump output power (power imparted to the liquid) and Pp is pump input power:

=+=

$$_p = \frac{P_w}{P_p}$$

=+=

The **efficiency curve** shows pump efficiency at various flow rates. The flow rate where efficiency is at a maximum is called the pump's best efficiency point (BEP). BEP is an important operating point that is further described later in this section.

=/= title: Efficiency Curve data-us: pc-data.csv data-metric: pc-data-metric.csv x: 1 series: 2, 3 series_title_index: 0 =/=

Pump Input Power Curve

The **pump input power curve** shows the amount of input power required for different flow rates. Pp can be determined by the following equation where Q is flow in

=+= [units = us]

$$P_p = \frac{Q * H * s}{3960 * {}_p}$$

=+=

=+= [units = metric]

$$P_p = \frac{Q * H * s}{366.6 * {}_p}$$

=+=

where:

- Pp = pump input power, in
- Q = rate of flow, in
- H = total head, in
- s = specific gravity
- np = pump efficiency

Pump input power can also be determined if the amount of power absorbed by the fluid and efficiency are known by rearranging the equation shown for the efficiency curve:

=+=

$$P_p = \frac{P_w}{{}_p}$$

=+=

The pump input power curve is important, as it allows proper selection of a driver for the pump.

=/= title: Pump Input Power Curve data-us: pc-data.csv data-metric: pc-data-metric.csv x: 1 series: 2, 3, 4 series_title_index: 0 =/=

Net Positive Suction Head Required (NPSHr) Curve

The final curve typically shown on a pump performance chart is the NPSHr for different flow rates. NPSHr is the minimum NPSH needed to achieve the specified performance at the specified flow rate, speed, and pumped liquid. NPSHr is further defined in the pump principles section.

=/= title: NPSHr Curve data-us: pc-data.csv data-metric: pc-data-metric.csv x: 1 series: 2, 3, 4, 5 series_title_index: 0 =/=

Operating Regions and Points

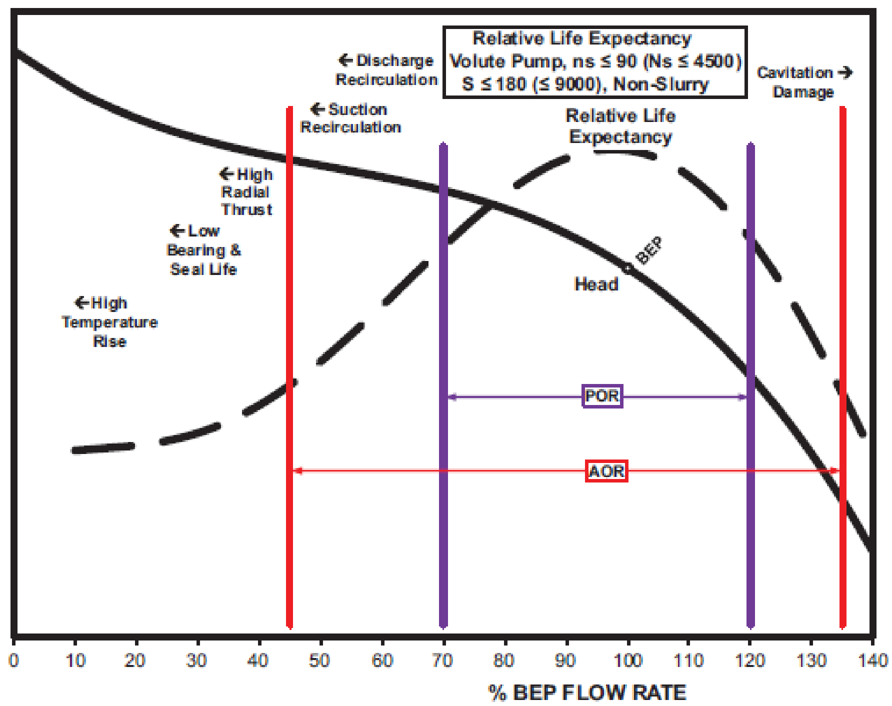


Figure 2:

Best efficiency point (BEP):

A pump's best efficiency pointis defined as the flow rate and head at which the pump efficiency is the maximum at a given speed and impeller diameter. Typically, a pump is specified to have its duty point, or designed operating point, at BEP. At BEP, a pump will have low vibration and noise when compared to other operating points. Also, there is minimum recirculation within the impeller and shockless entry into the impeller. Shockless entry is when the flow entering the impeller matches the angle of the impeller vanes at entry.

Preferred Operating Region (POR):

The preferred operating region (POR) is a range of rates of flow to either side of the BEP within which the hydraulic efficiency and the operational reliability of the pump are not substantially degraded. Flow induced vibrations and internal hydraulic loading is low in this region. Depending on the specific speed of the pump, which is further defined in the pump principles section, the POR can be anywhere from 90-110% of BEP flow to 70-120% of BEP flow.

Allowable Operating Region (AOR):

The AOR is the flow range at the rated speed with the impeller supplied in which the pump may be allowed to operate, as limited by cavitation, heating, vibration, noise, shaft deflection, fatigue, and other similar criteria. It is the flow range at which the pump can be run with acceptable service life. The pump manufacturer should be consulted to define this region. Typically, operating intermittently within this region does not cause issues over the life of the pump. The graph above shows the various operating regions, and the graph below shows the types of issues that can occur when operating outside of the POR and AOR.

Shut-off Head and Pump Runout:

These points are important during manufacturer testing to fully define the shape of the pump curve. They are the furthest points to the left and right on the curve. Shut-off is the condition of zero flow rate where no liquid is flowing through the pump, but the pump is primed and running. Operating at this point for more than a few seconds can cause serious mechanical issues. Pump Runout is the point at which flow is at a maximum. Operating at this flow can cause cavitation, vibration and, in some pumps, overloading of the driver. These points are to be avoided when operating pumps.

To learn more about pump operating regions, refer to ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Affinity Rules

Under the assumption that both pumps maintain the same efficiencies, the **Affinity Rules** show the relationships between pump parameters (flow, pressure/head, power) and pump characteristics (speed and impeller size). There are two parts to the Affinity Rules which vary by application; a change in speed while maintaining a constant impeller size or a change in impeller size while maintaining a constant speed.

1. Changing Speed / Constant Impeller Size

As seen below flow (Q), head (H), and power (P) are all proportional to the rotational speed (n):

=+= (1.1)

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

=+= =+= (1.2)

$$\frac{H_2}{H_1} = (\frac{n_2}{n_1})^2 = (\frac{Q_2}{Q_1})^2$$

=+= =+= (1.3)

$$\frac{P_2}{P_1} = (\frac{n_2}{n_1})^3 = (\frac{Q_2}{Q_1})^3$$

=+=

2. Changing Impeller Size / Constant Speed

As seen below flow (Q), head (H), and power (P) are all proportional to the impeller Size (D):

=+= (2.1)

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$

=+= =+= (2.2)

$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2 = (\frac{Q_2}{Q_1})^2$$

=+= =+= (2.3)

$$\frac{P_2}{P_1} = (\frac{D_2}{D_1})^3 = (\frac{Q_2}{Q_1})^3$$

=+=

Speed Reduction and Impeller Trimming

Part 1 of the affinity rules is ideal for instances where you have a Variable Frequency Drive attached to a pump motor. The VFD will reduce or increase the pump speed therefore allowing it to operate at a multitude of operating conditions. Part 2 is essential in calculating the new pumpcharacteristics after impeller trimming which is the reduction of the impeller diameter.

(Purchase HI's Application Guideline for Variable Speed Pumping at the Hydraulic Institute eStore.)

Pump Fundamentals: Parallel and Series Pump Implications

Two or more pumps in a system can be placed either in parallel or series. In **parallel**, a system consists of two or more pumps that are configured such that each draw from the same suction reservoir, wet well, or header, and each discharge to the same discharge reservoir or header. In **series**, a system consists of two or more pumps that are configured such that the discharge of one pump feeds the suction of a subsequent pump.

Pumps in Parallel

Pumps operating in parallel allow the pumping system to deliver greater flows than is possible with just one such pump. To determine the composite pump curve of two or more pumps operating in parallel, at each head value, the flowrate of each pump must be added together to obtain the composite flowrate.

The amount of increased flow that occurs within the system depends on both the shape of the system curve and shape of the pump curves. The **composite pump curve** intersects the system curve at different operating points yielding

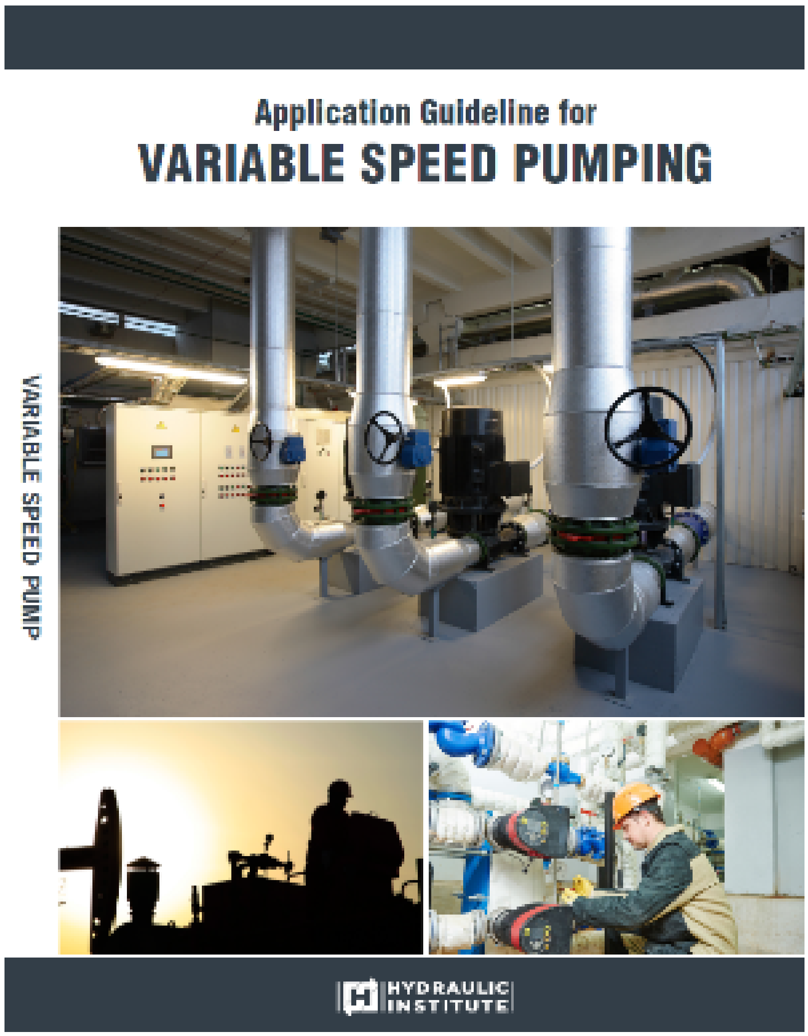

Application Guideline for
VARIABLE SPEED PUMPING
VARIABLE SPEED PUMP
HYDRAULIC
INSTITUTE

Figure 3:

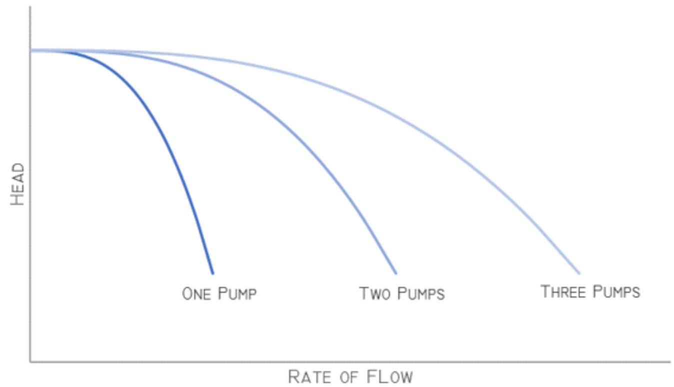


Figure 1. Composite pump curve – two and three identically sized pumps operating in parallel.

Figure 4:

different flowrates. As more pumps are called to operate, the flow will increase accordingly:

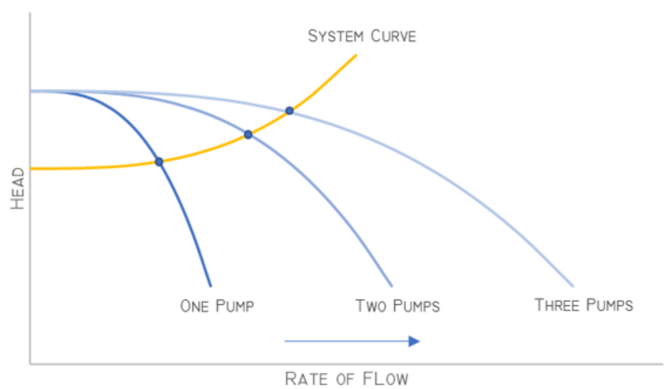


Figure 2. A shallow system curve superimposed on a composite pump curve (parallel). As each additional pump is brought online, the flowrate increases significantly.

Figure 5:

It should be noted, however, that unless the system curve is completely flat (which means friction and other dynamic losses are negligible), bringing a second pump on-line does not double the flow rate. The increased flow will be something less than double. How much less depends on the steepness of the system curve.

Pumps in Series

While pumps placed in parallel provide greater flow capabilities at the same head as one pump operating individually, pumps placed in series provide greater head capabilities at the same flowrate.

A composite pump curve representing pumps in series can be generated by adding the individual head values of the pumps for a given flow. Plotting this sum at various flow values will yield a composite pump curve for the group of pumps. Figure 3 shows a composite pump curve for two and three identically sized pumps operating in series:

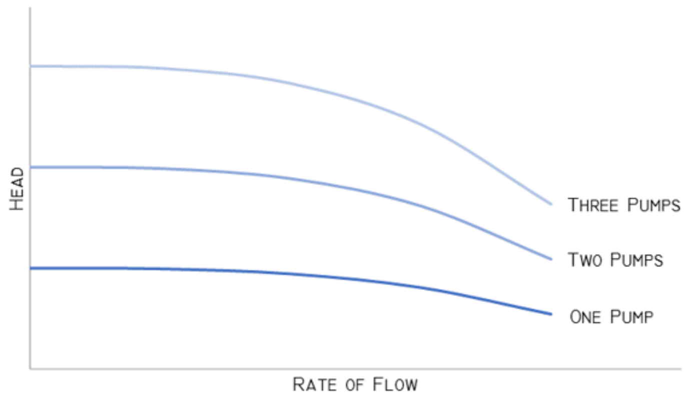


Figure 3. Composite pump curve – two and three identically sized pumps operating in series.

Figure 6:

Pumps operating in series allow the pumping system to deliver greater heads than is possible with just one such pump. This allows a pump station to be designed to satisfy systems that require large discharge pressures that may not be practical with one pump. Where certain applications require, it may also allow a pump station to address a wide variation in system pressures by staging the number of operating pumps. Figure 4 shows how applying a configuration with pumps in series to a system with a steep system curve may allow the pumps to address different head requirements so long as inter-stage discharge piping is configured to permit so.

Educational Demonstration

(Demonstrator will be placed here)

Pumps in Series

This demo explores how impeller diameter and speed affect three different pumps (A, B, and C). This demo also explores different scenarios of the three pumps operating in series.

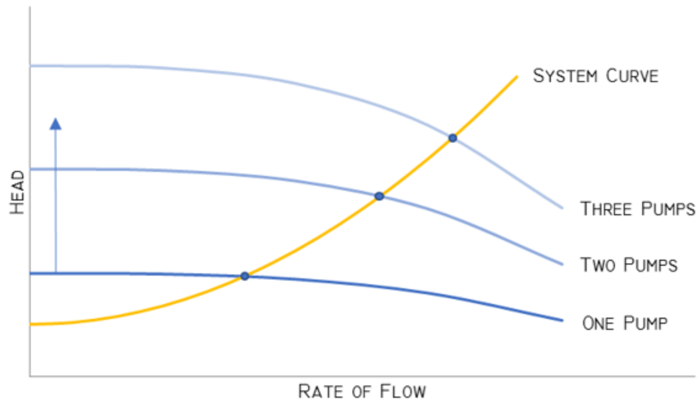


Figure 4. A steep system curve superimposed on a composite pump curve (series). As each additional pump is brought online, the head increases significantly.

Figure 7:

Change the Static Head

Use the toggle buttons by the reservoirs to increase or decrease elevation levels, thereby changing the static head.

Explore How Speed Affects a Pump Curve

Use the toggle buttons in the blue area to increase or decrease speed for a particular pump.

Explore How Impeller Diameter Affects a Pump Curve

Use the toggle buttons in the orange area to increase or decrease diameter for a particular pump.

Turn Different Curves On/Off

1) Click the chart
2) Select the "Chart Filters" Icon

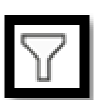


Figure 8:

3) Select/deselect desired curves (always make sure "Ghost" is checked - this keeps the axes values constant), then click "Apply".

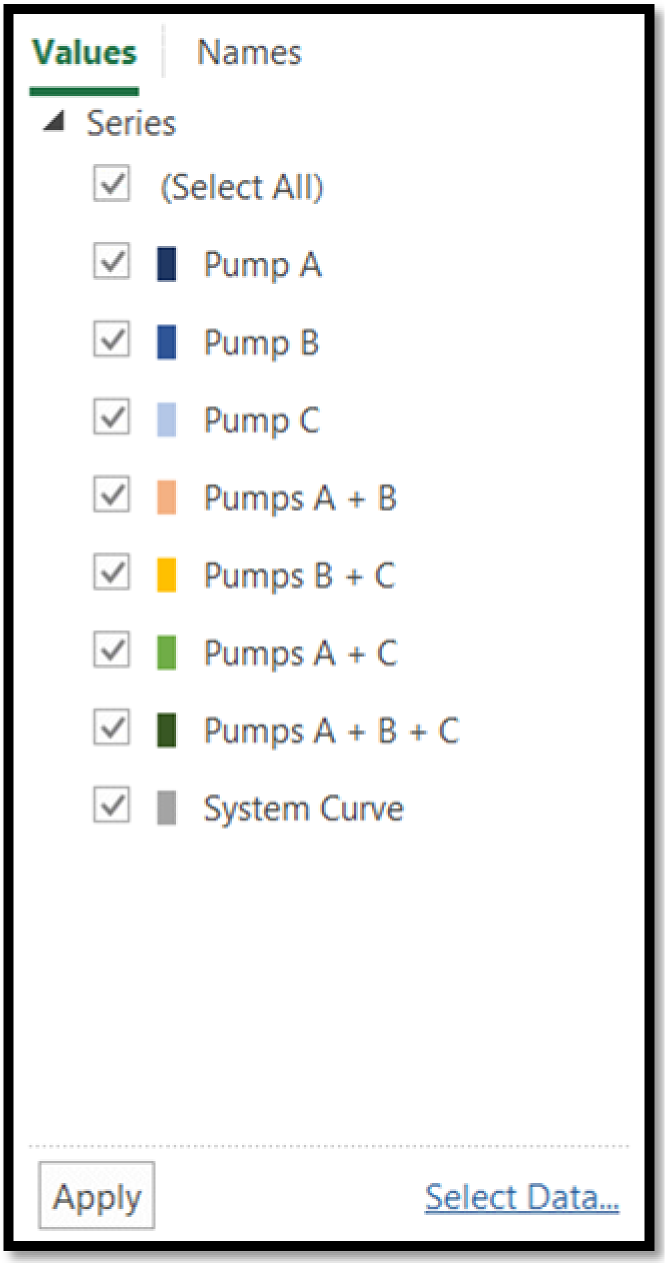

Values
Names
Series
(Select All)
Pump A
Pump B
Pump C
Pumps A + B
Pumps B + C
Pumps A + C
Pumps A + B + C
System Curve
Apply
Select Data...

Figure 9:

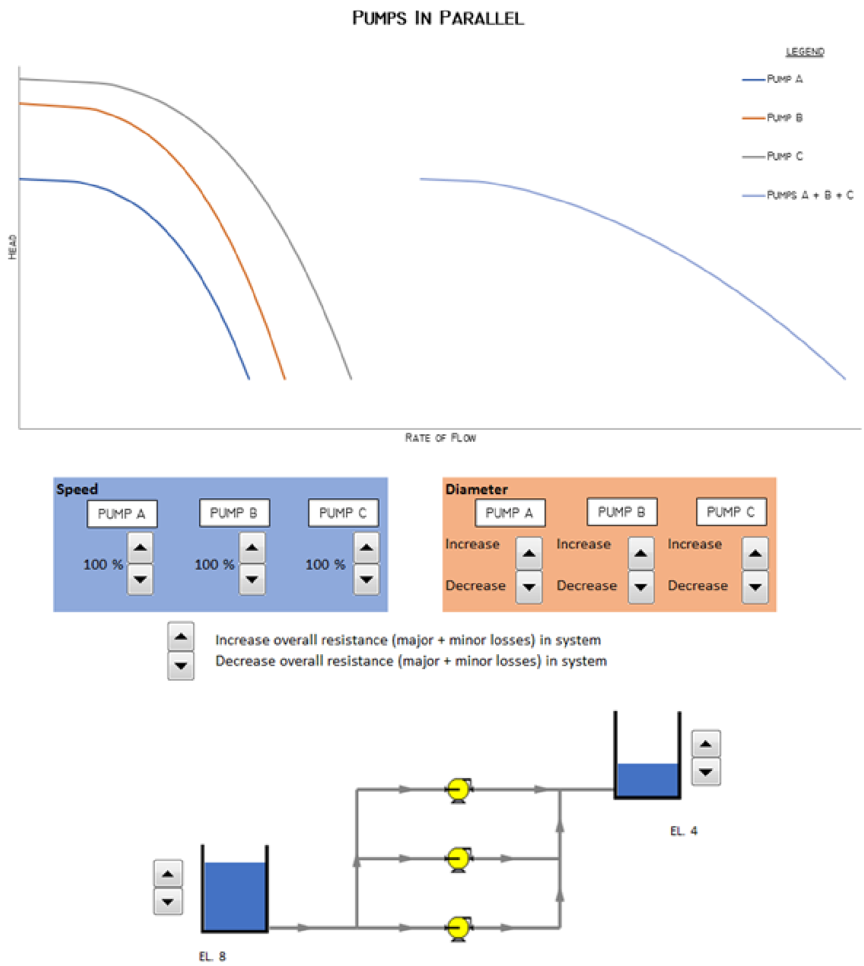

PUMPS IN PARALLEL
LEGEND
PUMP A
PUMP B
PUMP C
PUMPS A + B + C
HEAD
RATE OF FLOW
Speed
PUMP A
PUMP B
PUMP C
100 %
100 %
100 %
Diameter
PUMP A
PUMP B
PUMP C
Increase
Decrease
Increase
Decrease
Increase
Decrease
Increase overall resistance (major + minor losses) in system
Decrease overall resistance (major + minor losses) in system
EL. 4
EL. 8

Figure 10:

Worked Examples

Example 1 (U.S. Customary Units):

A booster pump is designed to operate at 1800 GPM and 135 ft., with a speed of 1740 RPM. Due to fluctuating flows the booster pump is equipped with a Variable Frequency Drive which reduces the pump speed by 10% during low flow conditions. Using the pump curve below and the affinity rules, generate the pump curve for low flow conditions and the new pumping conditions.

=/= title: data: total-head-feet.csv x: 1 series: 2 series_title_index: 0 =/=

=|= title: Data data: total-head-feet.csv =|=

Determine the Reduced Speed

During low flow conditions the speed of the pump is reduced by 10%.

=+=

$$n_2 = n_1(1 - .10)$$

$$n_2 = 1740(1 - .10)$$

$$n_2 = 1566\,RPM$$

=+=

Calculate New Flow Values

Using equation 1.1, calculate the new values (repeat until you convert all points under the flow column):

=+=

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}$$

$$\frac{Q_2}{0} = \frac{1566}{1740}$$

$$Q_2 = 0\,GPM$$

=+=

=+=

$$\frac{Q_2}{200} = \frac{1566}{1740}$$

$$Q_2 = 180\,GPM$$

=+=

=+=

$$\frac{Q_2}{400} = \frac{1566}{1740}$$

$$Q_2 = 360\,GPM$$

=+=

Calculate New Total Head Values

Using equation 1.2, calculate the new values (repeat until you convert all points under the Total Head column):

=+= [units = us]

$$\frac{H_2}{H_1} = (\frac{n_2}{n_1})^2$$
$$\frac{H_2}{173} = (\frac{1566}{1740})^2$$
$$H_2 = 173\, ft.$$

=+=

Plot Pump Curve

(need new plots and charts)

Example 2 (U.S. & Metric Units):

A pump designed with a

Calculate the New Impeller Diameter

During low flow conditions the speed of the pump is reduced by 10%.

=+= [units = us]

$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2$$
$$\frac{67}{80} = (\frac{D_2}{10.625})^2$$
$$\sqrt{\frac{67}{80}} = \sqrt((\frac{D_2}{10.625})^2)$$
$$0.915 = \frac{D_2}{10.625}$$
$$D_2 = 9.72\, in. \approx 9.75\, in.$$

=+=

=+= [units = metric]

$$\frac{H_2}{H_1} = (\frac{D_2}{D_1})^2$$
$$\frac{20.42}{24.38} = (\frac{D_2}{270})^2$$
$$\sqrt{\frac{20.42}{24.38}} = \sqrt((\frac{D_2}{270})^2)$$

$$0.915 = \frac{D_2}{270}$$

$$D_2 = 247\,mm$$

=+=

Calculate the New Flow

During low flow conditions the speed of the pump is reduced by 10%.

=+= [units = us]

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$

$$\frac{Q_2}{2000} = \frac{9.75}{10.625}$$

$$Q_2 = 1835\,GPM$$

=+=

=+= [units = metric]

$$\frac{Q_2}{Q_1} = \frac{D_2}{D_1}$$

$$\frac{Q_2}{454.2} = \frac{247}{270}$$

$$Q_2 = 416.8\,m^3/h$$

=+=

Miscellaneous Content

Combined Pump & System Curves

Tutorial

It is important to understand how the pump will interact with the system it is used in over a range of operating conditions. Combining the pump performance curve with the system curve will help show where the system will operate. In general, the system flow rate will be where the pump curve intersects the system curve. Pump and system curve interaction is covered in ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Knowing the shape of the system curve will help to understand how the pump operating conditions will change if the system changes due to things like valve position, parts of the system coming on and off line, and upset conditions. Additionally, by overlaying the pump curves, it will help in determining if the pump is sized correctly to overcome the static and dynamic head of the system.

Using a pump and system curve will also help evaluate pump speed and impeller trimming. Both of which will change the pump curve and, therefore, where the system will operate. This will also help ensure the pump operation will be as close to the Best Efficiency Point (BEP) as possible to reduce energy consumption and increase pump reliability.

Reading a Pump and System Curve Plot

As the name implies, a **pump and system curve plot** consists of at least two curves. The system curve will show the static head of the system (the head required to overcome gravity at zero flow) and the dynamic head, which is the frictional losses at varying flow rates. The operating point is where the two curves intersect.

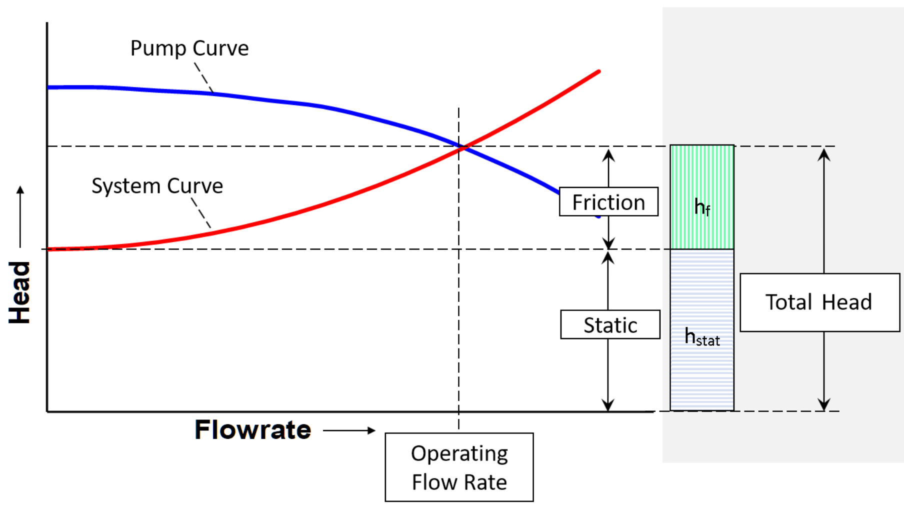


Figure 11:

Flow Rate Change Using Manual Throttling Valve

As a manual (or passive) valve is changed (opening or closing) it will change the system curve by affecting the K value. Closing a valve will add resistance to the system over the entire range of flows (an opening will reduce resistance). This can be shown on the pump system plot with the system curve bending upward. Note the static head at zero flow will still be the same. Using the revised pump system plot, a new operating point can be determined.

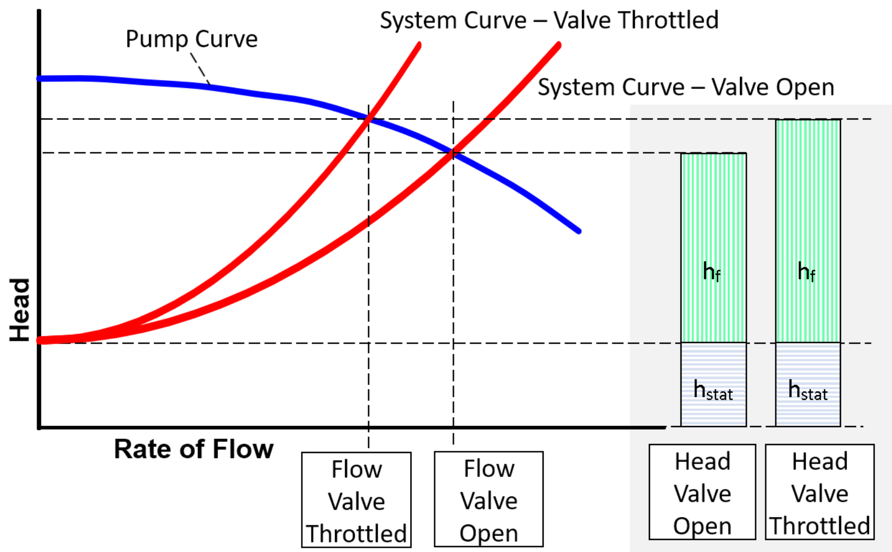


Figure 12:

Changes in Pump Speed

Changing the pump speed will change the pump curve. This can be represented using the affinity or similarity rule. As the pump is slowed the pump curve will be shifted down and to the left, getting closer to the plot origin.

With a system that has a manual throttling valve (active control valves are discussed later), changing the pump speed will change two things as shown in the pump system plot – the system flow and the pump head generated. Slowing the pump down, as depicted in the chart, will reduce pump head produced and reduce the system flow. Note the static head at zero flow will still be the same.

Changes in Impeller Size

In selecting the appropriate curve for a pump application to fit the desired system conditions, many centrifugal pumps can use different sized impellers to shift the pump curve. Trimming the impeller down in size will move the pump curve down much in the same way as reducing the speed of rotation. The same can be said for selecting a larger impeller; the curve will shift up. When sizing a pump for an application in which the pump is not hooked up to a variable speed controller, it is more appropriate to size the impeller to your desired duty conditions.

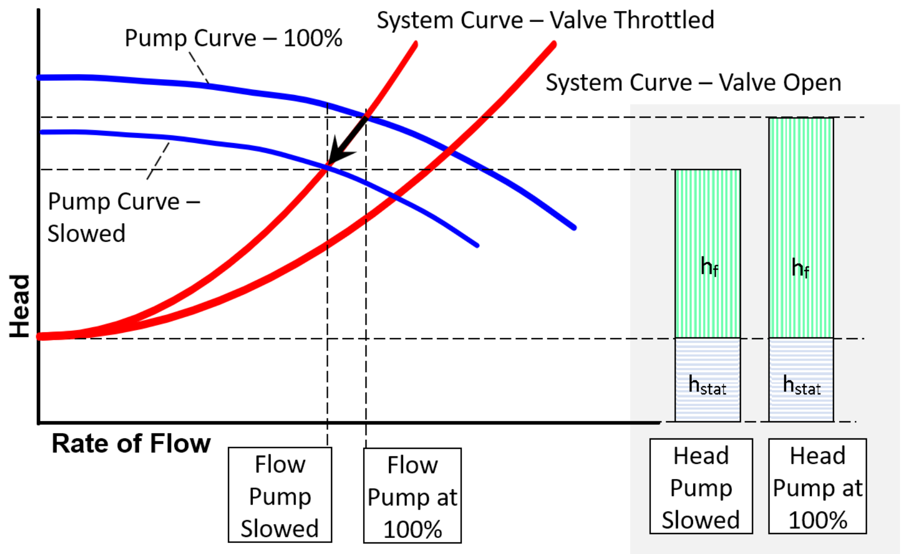


Figure 13:

Active Control Valves

An **active control valve** is one that continually changes position (loss) to maintain a set flow or pressure. It is important to note that there is no human intervention involved. Since they continuously vary their loss to maintain a flow or pressure, there is no unique valve over a range of system flows. Because of this, they are not normally included in the system curve. But they are shown on the pump system plot as the difference between the two curves at the operating point. In other words, a system with an active control valve will not operate at the intersection of the pump and system curves since the control valve will fix the system at a certain point.

This pump system plot shows two things. First, with the control valve in the system, the pump head required is the sum of the static head, frictional losses and the loss across the control valve. It also shows that the flow in the system is less than the flow in the system without the control valve.

The plot also is useful in determining the margin available on the control valve. Having sufficient, but not excessive, pressure drop across most control valves is required so the valve can properly control to the setpoint.

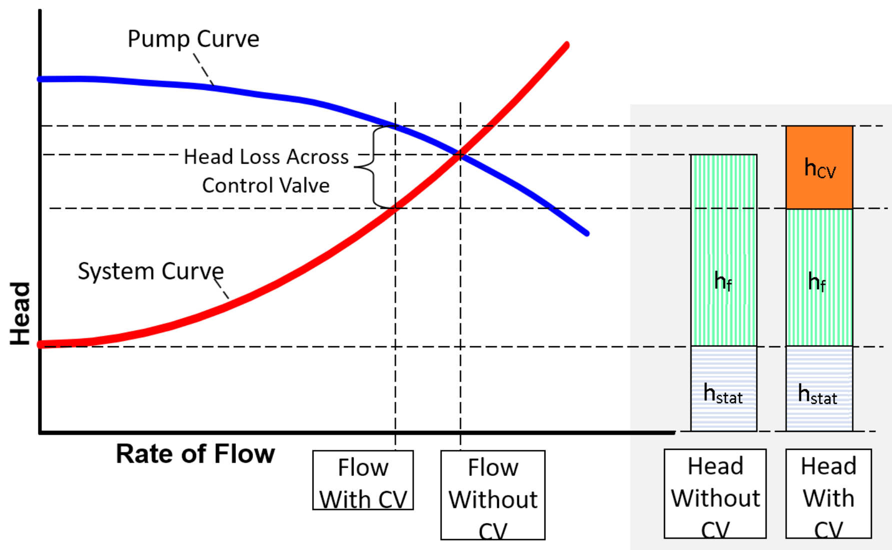


Figure 14:

Changing the Setpoint on an Active Control Valve

Changing the **setpoint (*or control point*)** on an active control valve will change the operating point in the system. This will result in changing the frictional loss, the loss across the control valve, and the head required on the pump. In the example here, the setpoint is changed so that the flow is decreased. Notice that with this decreased flow, the head required for the pump will increase.

Changing the Pump Speed with an Active Control Valve

When the pump speed is changed and there is an active control valve in the system, the difference in head produced by the pump will be reflected in the difference in loss across the control valve, since the system flow rate has not changed.

Pump Sizing in the Real World

Many real-world applications are designed for system curves that are provided as an envelope. This is due to varying head conditions caused by stormwater expectations, varying reservoir levels, piping conditions over time, or the use of pressurized tanks. Further discussion on this topic can be found in the FAQ section.

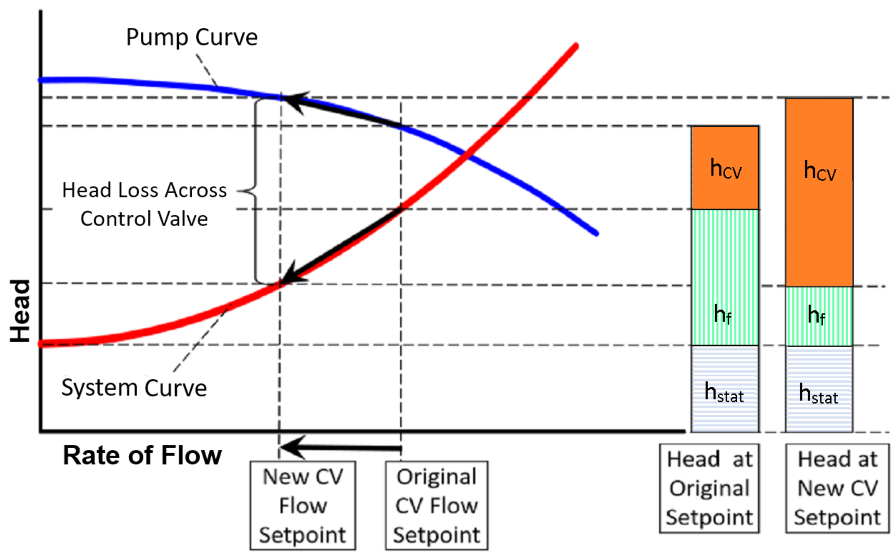


Figure 15:

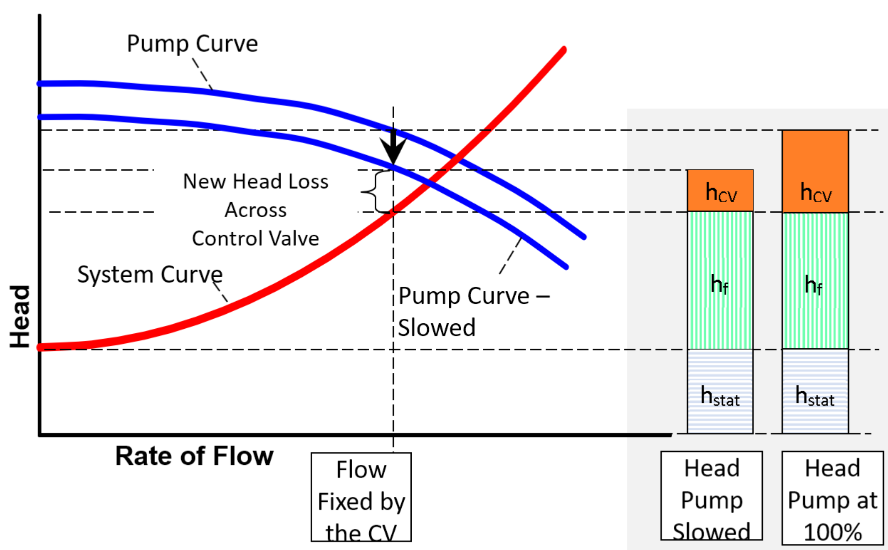


Figure 16:

Parallel and Series System Implications

The overall effect on the system behavior when adding pumps in parallel depends on the type of system, i.e. the shape of the system curve. For friction dominated systems, (steep system curve) bringing additional parallel pumps online may not change the operating point (more flow or head) much. Conversely, adding more parallel pumps to a system that is dominated by static head (flatter system curve) will have a greater effect on the operating point.

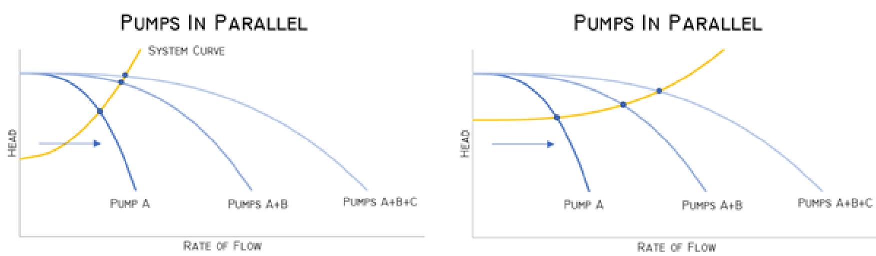


Figure 17:

Since the head is additive for series pumps, the effect would also be different. Using more pumps on a friction dominated system will have a significant increase in head with a lesser increase in flow. A static dominated system will be the opposite, there will be a significant increase in flow with a lesser increase in head.

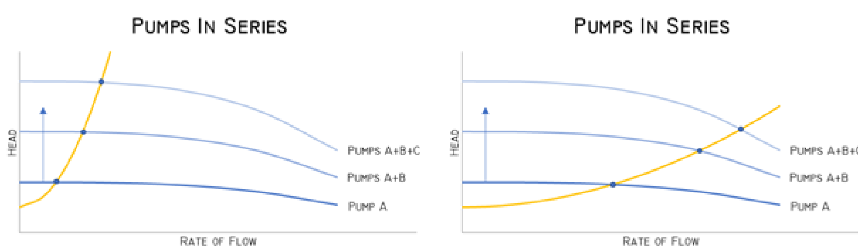


Figure 18:

Educational Demonstration

This educational demonstrator will show how changing the pump speed and the system curve will change the operating point. This is a representative demonstration and does not have actual values.

Change the pump speed slider to change the shape of the pump curve. The static head can be varied by changing the supply and destination levels. The overall frictional loss (piping and minor) can be varied by moving the slider.

Parallel Pumps

This demo explores how three different pumps (A, B, and C) operate in parallel, how impeller diameter and speed affect each of the pumps, and how the system curve interacts with the pump curves.

Explore how speed affects a pump curve by using the toggle buttons in the blue area to increase or decrease speed for a pump. Explore how impeller diameter affects a pump curve by using the toggle buttons in the orange area to increase or decrease diameter for a pump. Change the static head by increasing and/or decreasing the reservoir elevation levels.

Turn Different Curves On/Off

1) Click the chart
2) Select the "Chart Filters" Icon
3) Select/deselect desired curves, then click "Apply".

Series Pumps

This demo explores how three different pumps (A, B, and C) operate in series, how impeller diameter and speed affect each of the pumps, and how the system curve interacts with the pump curves.

Explore how speed affects a pump curve by using the toggle buttons in the blue area to increase or decrease speed for a pump. Explore how impeller diameter affects a pump curve by using the toggle buttons in the orange area to increase or decrease diameter for a pump. Change the static head by increasing and/or decreasing the reservoir elevation levels.

Turn Different Curves On/Off

1) Click the chart
2) Select the "Chart Filters" Icon
3) Select/deselect desired curves, then click "Apply".

Worked Example (U.S. Customary Units)

Previously we developed a system curve for the system shown below for flows from 0 to 300 gpm. Using 4-inch pipe, the function in terms of gpm is the

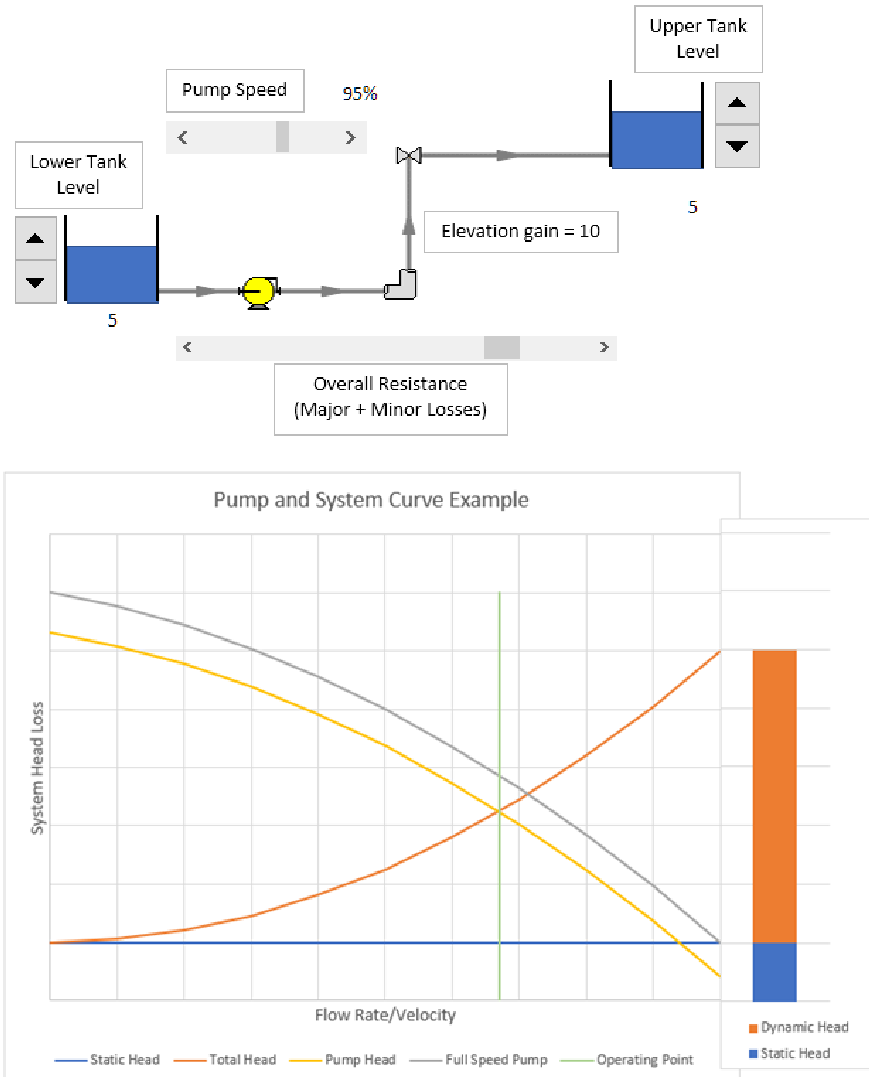


Figure 19:

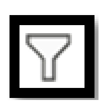


Figure 20:

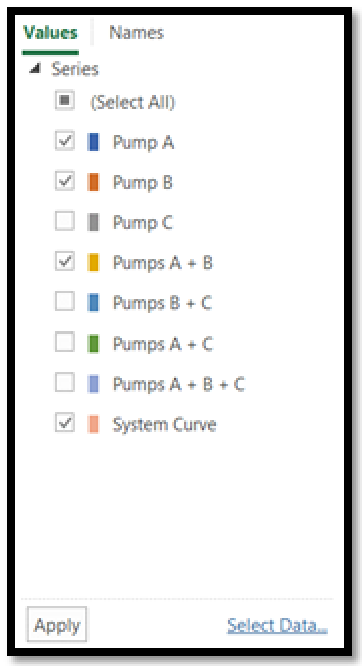

Values
Names
Series
(Select All)
Pump A
Pump B
Pump C
Pumps A + B
Pumps B + C
Pumps A + C
Pumps A + B + C
System Curve
Apply
Select Data...

Figure 21:

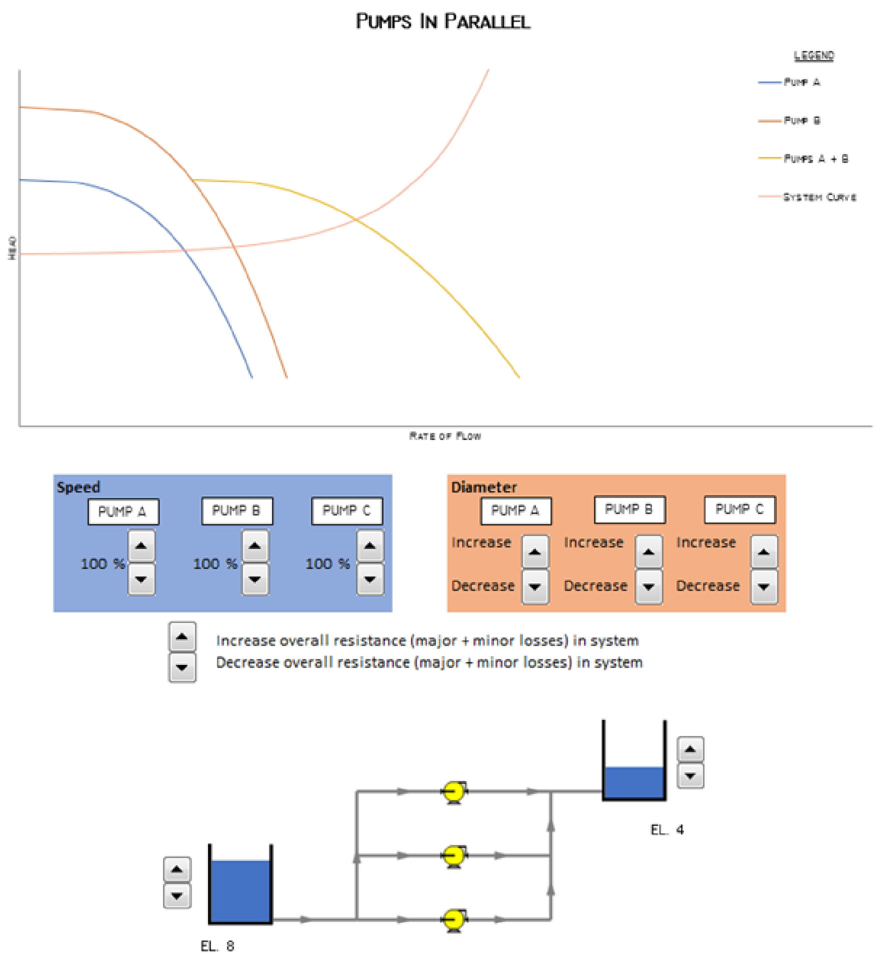


Figure 22:

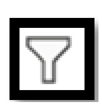


Figure 23:

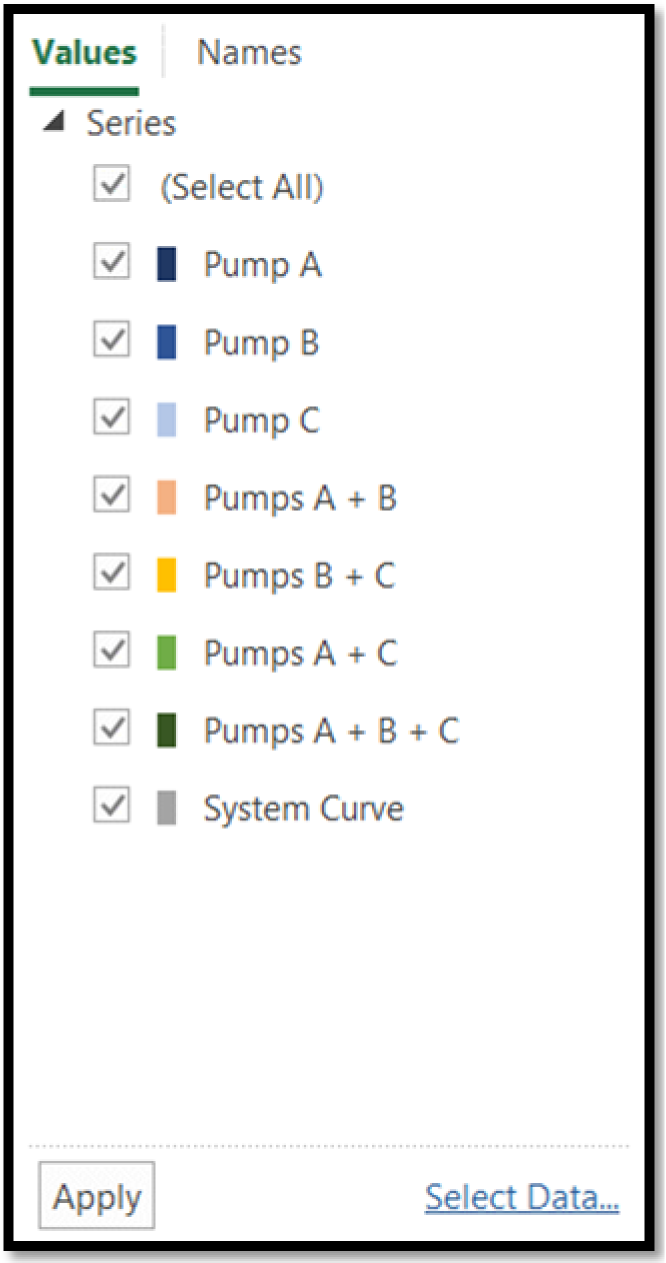

Values
Names
Series
(Select All)
Pump A
Pump B
Pump C
Pumps A + B
Pumps B + C
Pumps A + C
Pumps A + B + C
System Curve
Apply
Select Data...

Figure 24:

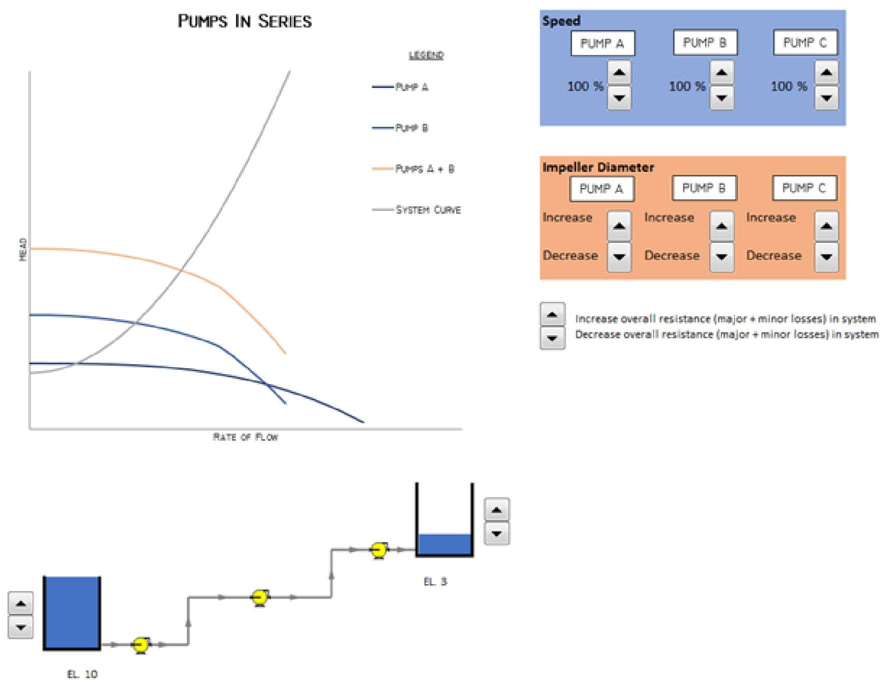

PUMPS IN SERIES
LEGEND
PUMP A
PUMP B
PUMPS A + B
SYSTEM CURVE
HEAD
RATE OF FLOW
Speed
PUMP A
PUMP B
PUMP C
100 %
100 %
100 %
Impeller Diameter
PUMP A
PUMP B
PUMP C
Increase
Increase
Increase
Decrease
Decrease
Decrease
Increase overall resistance (major + minor losses) in system
Decrease overall resistance (major + minor losses) in system
EL. 3
EL. 10

Figure 25:

following.

=+=

$$\Delta h_{system} = 265 feet + (7.75E{-}04)Q^2$$

=+=

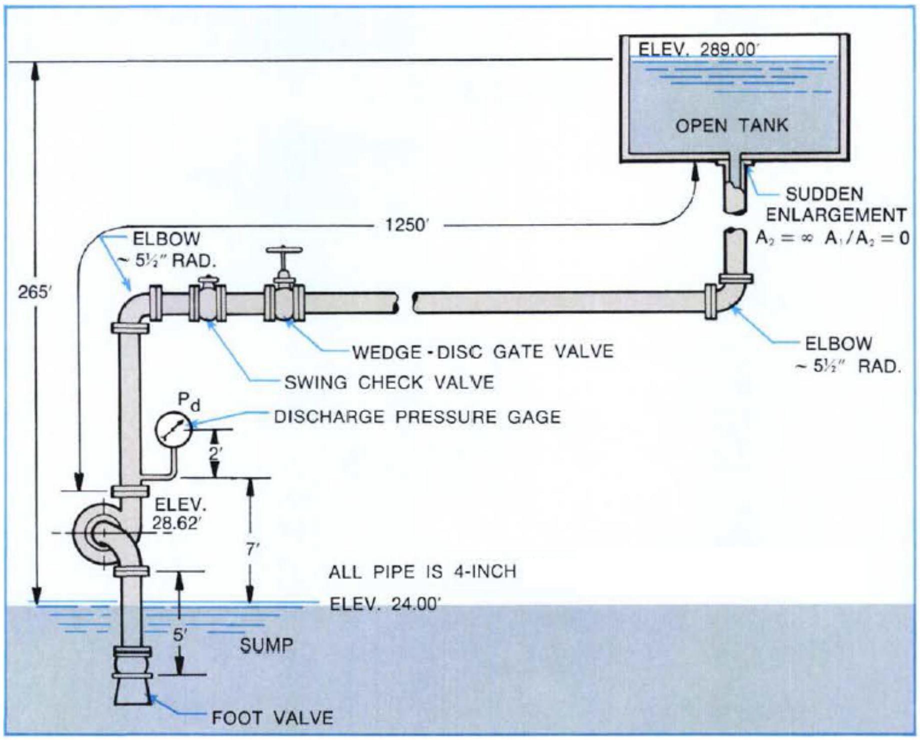


Figure 26:

Verifying the Pump Curve with the System

We need this system to operate at 200 GPM. Based on the system curve previously determined, this would require 296 feet of head. Finding the perfect pump from a vendor, we select some data points from the pump curve which are shown in the following table.

=|= title: Data data: qdH-us.csv =|=

Using a second-order polynomial curve fit, we get the following pump curve equation:

=+=

$$\Delta h_{pump} = 380 - 0.06Q - 0.0018Q^2$$

=+=

We can combine the system curve with the pump curve to get an overall understanding of how the system will operate.

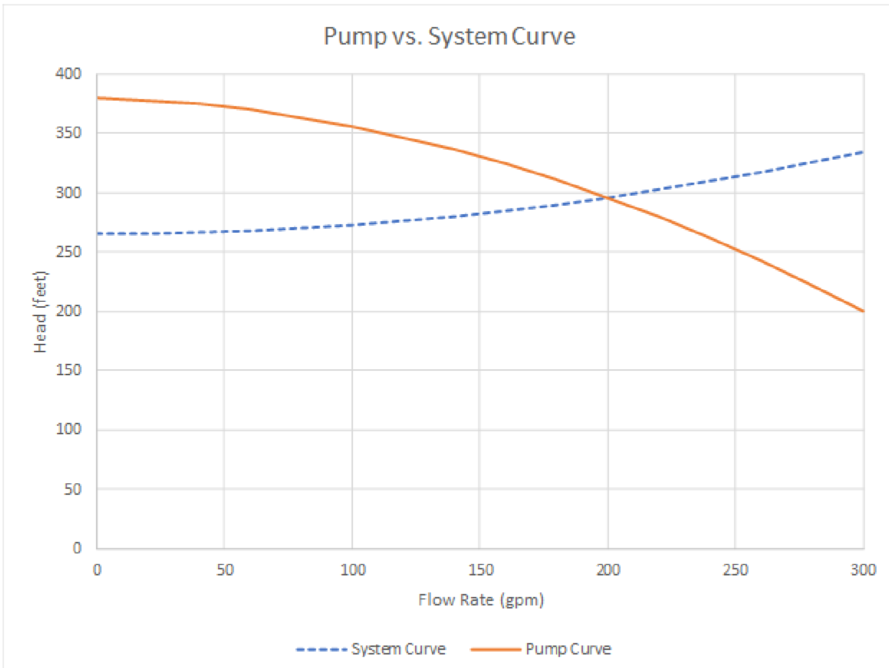


Figure 27:

Since this system does not have active control devices, the system will operate where the pump and system curves intersect, which is at

System Deviations

Both the pump and the system can deviate from this ideal design case. For example, the pump performance can degrade, or the system losses can increase with fouling over time. If we combine the pump and system curves we can evaluate what will happen in various cases.

For example, let's examine what happens with the tank level changes. With all other factors being held constant, this would change the static head of the system. The pump would also change its operating point in response. Since the operating point will be where the pump and system curves intersect, we can set the two equations equal and solve for flow rate.

=+=

$$\Delta h_{system} = \Delta h_{pump}$$

=+=

=+=

$$\Delta h_{static} + 7.75e^{-4}Q^2 = 380 - 0.06Q - 0.0018Q^2$$

=+=

=+=

$$(\Delta h_{static} - 380) + 0.06Q + (7.75e^{-4} + 0.0018)Q^2 = 0$$

=+=

We can solve this equation using the quadratic formula:

=+=

$$Q = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

=+=

where:

- a = 7.75e-4 + 0.0018
- b = 0.06

- c = Δhstatic - 380

If, for example, the tank level rises 10 additional feet, the static head would increase to 275 feet. Solving the above equation, we determine that the new flow rate into the tank would be 190.6 GPM.

Other cases (e.g. pipe or fitting resistances, pump speed, etc.) would require some corresponding factors to be left as variables in the equations so they can be changed. But the methodology would remain the same.

Frequently Asked Questions

Pump Sizing in the Real World

Many real-world applications are designed for system curves that are provided as an envelope. This is due to varying head conditions caused by stormwater expectations, varying reservoir levels, piping conditions over time, or the use of pressurized tanks. The below is an example of a system curve envelope which dictates all the conditions seen. It is up to the system designer to specify the important points the selected pumps need to hit.

NOTE: This image needs to be modified to remove customer references (also change the specific C values to low friction and high friction), maybe take off the specific values.

For many situations, the use of pumps in parallel or pumps in series can be used to define minimum and maximum conditions considering the use of a Variable

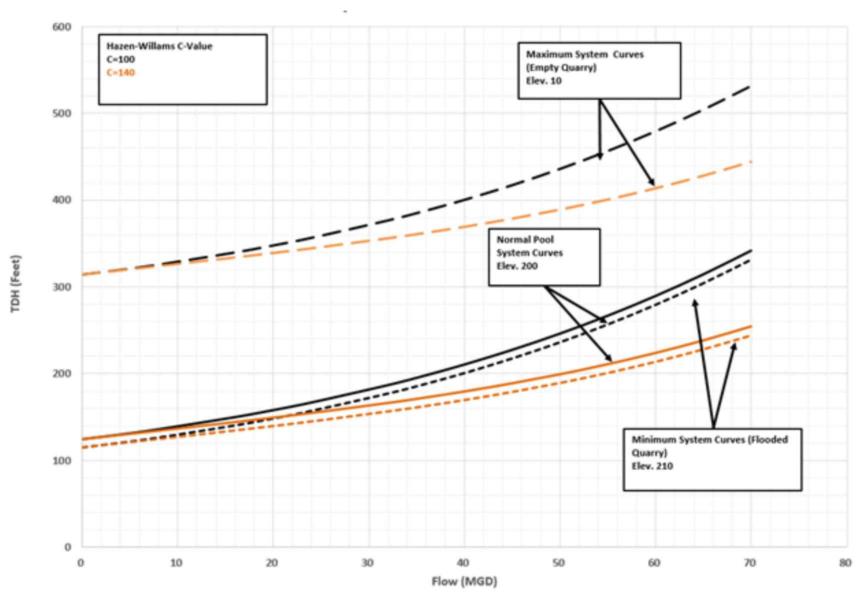


Figure 28:

Speed Controller in order to cover the points below the pump curves. Below is an example of 3 identical pumps in parallel being sized to cover a system curve with varying head conditions.

NOTE: This image needs to be modified to remove customer references (also change the specific C values to low friction and high friction), maybe take off the specific values.

When selecting the appropriate pump in a situation like this, the use of a Variable Speed Controller is expected which means it is possible to reach duty conditions that exist within the curve envelope. Using affinity rules will allow the engineer to calibrate the Variable Speed Controller to cause the pumps to hit any desired duty condition within the curve envelope. It is important to remember that at any given time, the only point to dictate pump performance is where the present system curve will intersect the pump curve.

Undersized or Oversized Pumps

It is important to note that as the flow increases, the pump head produced will decrease as the system flow losses increases. Oversized or undersized pumps will have curves that do not intersect at the required flow rate or will not cross at all. This will show that a different pump should be selected.

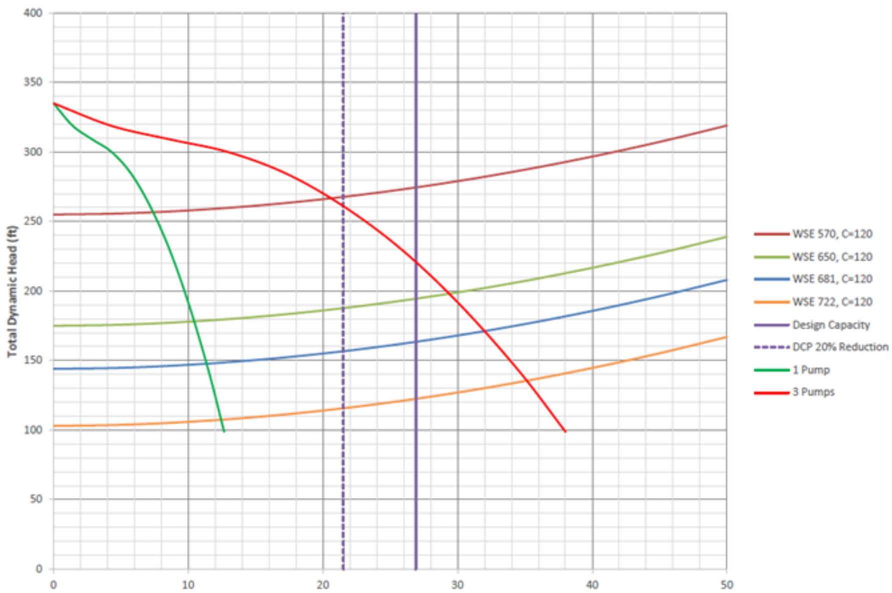


Figure 29:

Pump Principles

Impeller Specific Speed

Impeller specific speed is an index of pump performance at the pump's best efficiency point (BEP) rate of flow, with the maximum diameter impeller, and at a given rotational speed. Specific speed is expressed by the following equation:

=+= [units = us]

$$Ns = \frac{n(Q)^{0.5}}{(H_t)^{0.75}}$$

=+=

=+= [units = metric]

$$n_s = \frac{n(Q)^{0.5}}{(H_t)^{0.75}}$$

=+=

where:

-
- n = Rotational speed, in revolutions per minute

- Q = Discharge at best efficiency point (BEP)
- Ht = Total head,

The user is cautioned to check carefully the basis of calculation of specific speed and suction specific speed before making comparisons because there are subtle but significant differences in methods used throughout industry and in related textbooks and literature.

When calculating specific speed using units of cubic meters per second for flow rate and meters for head per stage, 51.6 is the conversion factor for specific speed in US gallons per minute and feet (i.e., metric × 51.6 = US customary units.)

The usual symbol for specific speed is:

=+= [units = us]

$$Ns$$

=+=

=+= [units = metric]

$$n_s$$

=+=

Below is the approximate shape of performance curves based on the impeller specific speed with metric and (US) units:

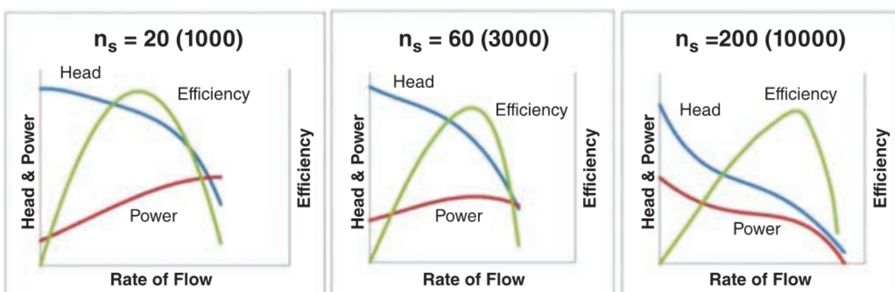


Figure 30:

Below is a graph that shows the maximum practically attainable efficiency for different flow rates. This proves useful in selection of hydraulics for particular applications, based on desired flow and head.

Type number is a variation of impeller specific speed. It is a dimensionless quantity calculated at the point of best efficiency, which is defined by the following formula:

=+=

$$K = \frac{2nQ'^{0.5}}{(gH')^{0.75}} = \frac{Q'^{0.5}}{(y')^{0.75}}$$

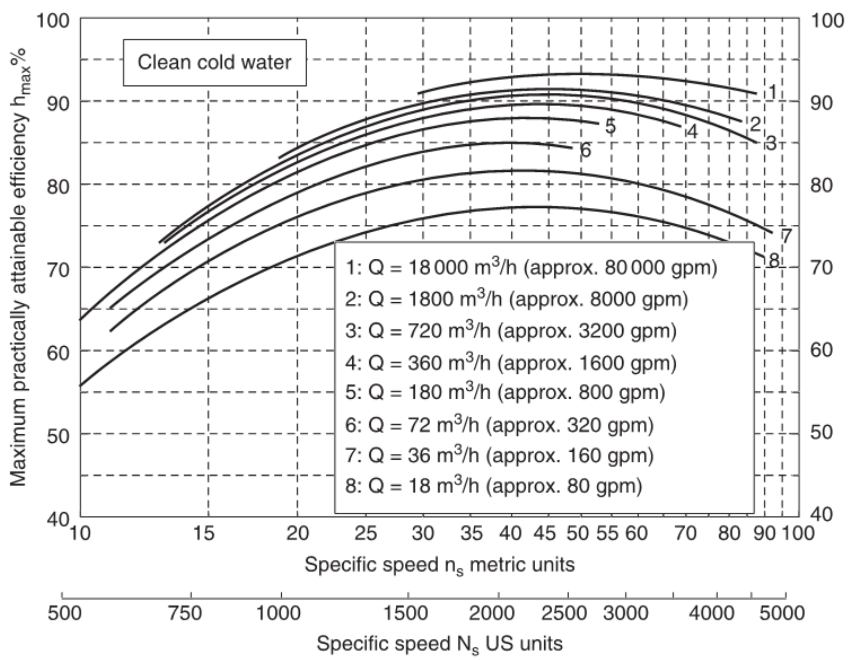

Clean cold water
Maximum practically attainable efficiency h_max %
Specific speed n_s metric units
Specific speed N_s US units
1: Q = 18 000 m³/h (approx. 80 000 gpm)
2: Q = 1800 m³/h (approx. 8000 gpm)
3: Q = 720 m³/h (approx. 3200 gpm)
4: Q = 360 m³/h (approx. 1600 gpm)
5: Q = 180 m³/h (approx. 800 gpm)
6: Q = 72 m³/h (approx. 320 gpm)
7: Q = 36 m³/h (approx. 160 gpm)
8: Q = 18 m³/h (approx. 80 gpm)

Figure 31:

=+=

where:

- Q' = discharge per eye,
- H' = head of first stage in
- n = rotational speed, in revolutions per minute
- g = gravitational acceleration,
- = angular velocity, in radians per second
- y' = specific energy,

Type number derived using the stated metric units, multiplied by a factor of 11.19, is equal to the type number derived using the stated (US) units.

Suction Specific Speed

Suction Specific Speed is an index of pump suction operating characteristics. It is determined at the BEP flow rate with the maximum diameter impeller. Suction specific speed is an indicator of the NPSHR for a 3% drop in head (NPSH3) at a given rate of flow and rotational speed. Suction specific speed is expressed by the following equation:

=+= [units = us]

$$Nss = \frac{n(Q')^{0.5}}{(NPSH3)^{0.75}}$$

=+=

=+= [units = metric]

$$S = \frac{n(Q')^{0.5}}{(NPSH3)^{0.75}}$$

=+=

where:

-
- n = Rotational speed, in revolutions per minute
- Q' = flow rate per impeller eye *OR* total flow rate for single suction impellers *OR* one half total flow rate for double suction impellers,
- NPSH3 = Net positive suctions head required in feet that will cause the total head (or first stage head of multistage pumps) to be reduced by 3%

Suction specific speed derived using cubic meters per second and meters, multiplied by a factor of 51.6, is equal to suction specific speed derived using US gallons per minute and feet. The US customary symbol NSS is sometimes used to designate suction specific speed.

The user is cautioned to check carefully the basis of calculation of specific speed and suction specific speed before making any comparisons because there are subtle but significant differences in methods used throughout industry and in related textbooks and literature.

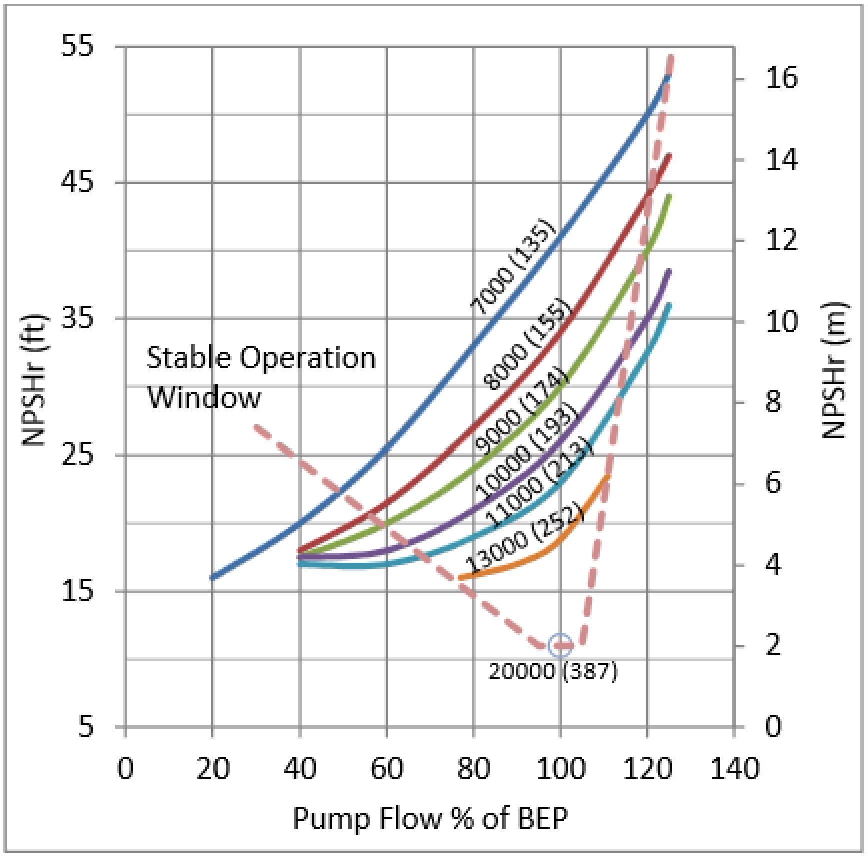


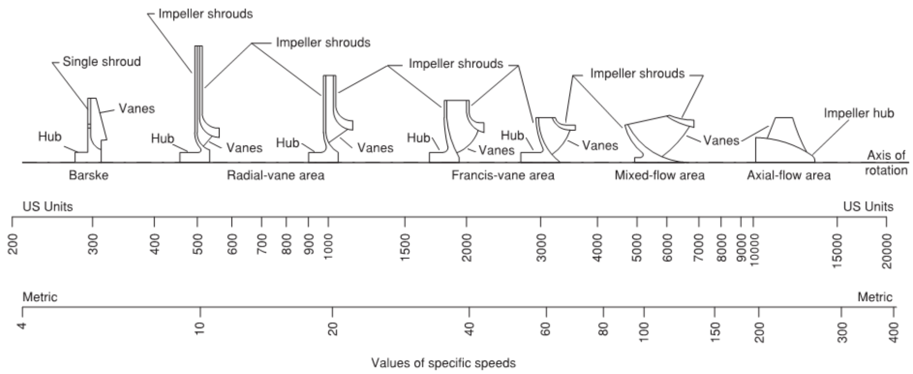
Figure 3: Stable window according to Lobanoff & Ross.

Figure 32:

Impeller Types

There are many different impeller types (pictured below) based on desired performance characteristics and type of fluid pumped. The main types of impellers are shown below. As the flow increases with respect to the developed head, the larger the waterways become and the smaller the diameter becomes.

The three configurations (pictured below) for an impeller are open, semi-open, and closed. Open impellers do not have a front or rear shroud. Semi-open impellers only have a rear shroud. Closed impellers have a front and rear shroud.



Side channel and Barske	Radial	Mixed flow	Axial propeller
$n_s = 4 - 12$	$n_s = 8 - 45$	$n_s = 40 - 160$	$n_s = 100 - 300$
(200 – 600)	(400 – 2300)	(2100 – 8300)	(5200 – 15,500)

Figure 33:

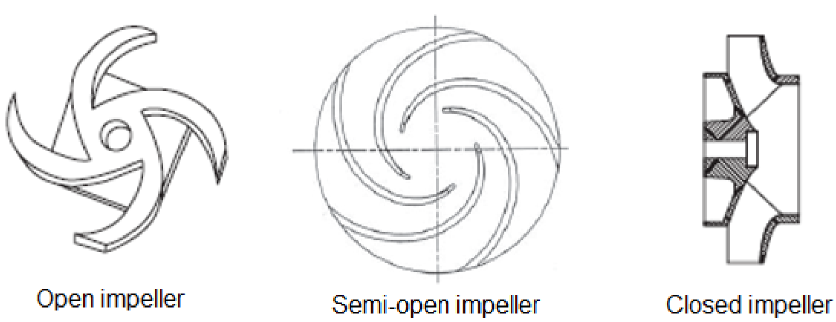


Figure 34:

Open impellers are typically used on smaller pumps and are weaker than closed impellers because of the lack of reinforcement of shrouds. They are typically cheaper to manufacture and easier to clean, but become inefficient as the pump wears. **Closed impellers** are more expensive to manufacture, more difficult to clean, and cannot pump as many types of fluids as open impellers, but they are stronger and experience a much lower decrease in efficiency over the life of the pump. **Semi-open impellers** share some of the advantages and disadvantages of each. The reason for loss of efficiency in an open or semi-open impeller is that the distance between the vanes and the pump case surface increases over time due to wear. This allows for leakage back to suction, reducing efficiency. Another benefit of closed impellers is that setting axial distance doesn't need to be as precise for this same reason.

Net Positive Suction Head (NPSH)

NPSH is the net positive suction head in

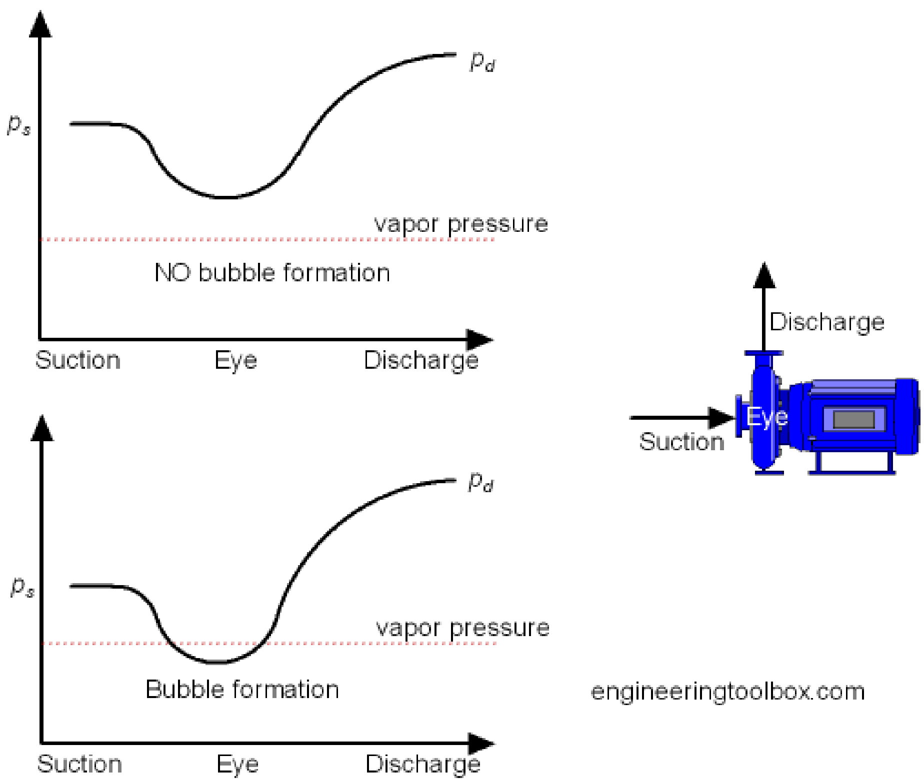


Figure 35:

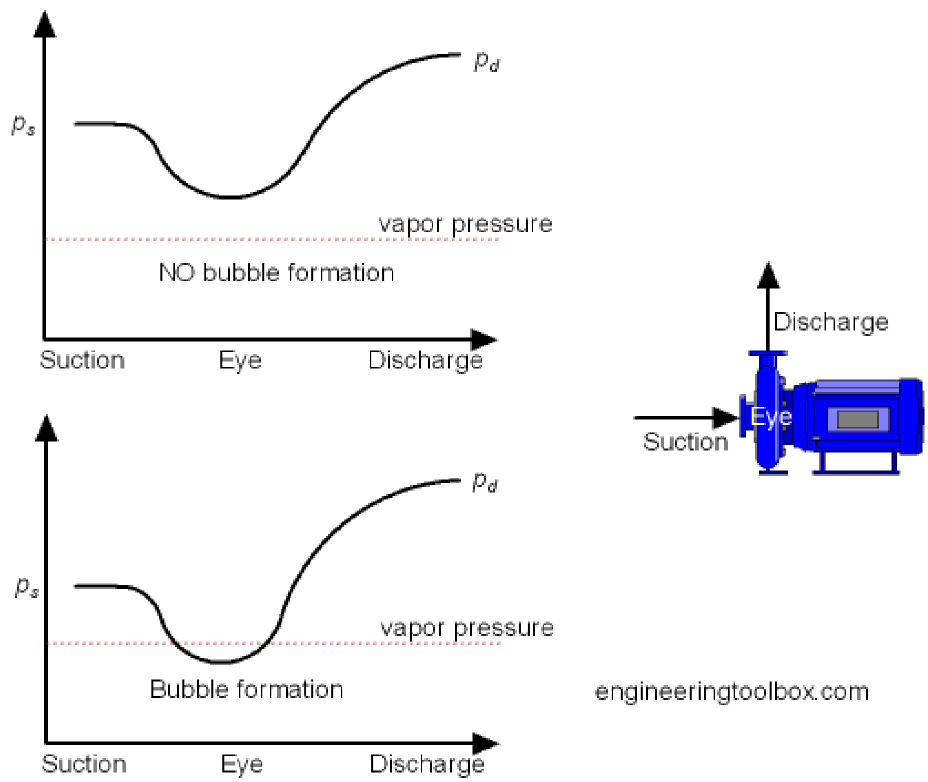

p_d
p_s
vapor pressure
NO bubble formation
Suction
Eye
Discharge
p_d
p_s
vapor pressure
Bubble formation
Suction
Eye
Discharge
Discharge
Eye
Suction
engineeringtoolbox.com

Figure 36:

The **net positive suction head available (NPSHa)**, which is the NPSH available at the pump site, is defined as:

=+=

$$NPSH_a = h_{sa} - h_{vp}$$

=+=

where:

- hsa = Total suction head absolute in
- hatm = Atmospheric pressure in head in
- hs = Suction head
- hvp = Vapor pressure of fluid in

A pump's **net positive suction head required (NPSHr)** is important, as it allows a pump user to determine the amount of NPSHa needed at their pump site to ensure pump performance is met. The occurrence of visible cavitation, increase of noise and vibration due to cavitation, beginning of head or efficiency drop, and cavitation erosion can occur when margin above NPSHr is present. NPSH3 is the value of NPSHr when the first-stage total head drops by 3% due to cavitation. ANSI/HI 9.6.1 – Rotodynamic Pumps – Guideline for NPSH Margin establishes recommended net positive suction head available (NPSHA) above the published NPSH required (NPSHR) that will lead to acceptable pump performance and service life.

Below is a common graph seen when testing a pump for NPSHr. This shows that as NPSHa is reduced, there is a point at which the head starts to drop off. On this test, the NPSH3 is approximately 11 feet, as this is when the head drops 3% - in this case from 200 feet to 194 feet.

Affinity Rules

This section will include examples, animations, and a calculator

Affinity rules are used in turbomachinery to approximately calculate a centrifugal pump's influence on capacity, head, and power consumption with changing speed or impeller diameter.
Flanges

Pipe Dimensions: Wrought Steel & Stainless Steel (According to ASME B36.10M-2015)

=|= data: steel1.csv =|=

=|= data: steel2.csv =|=

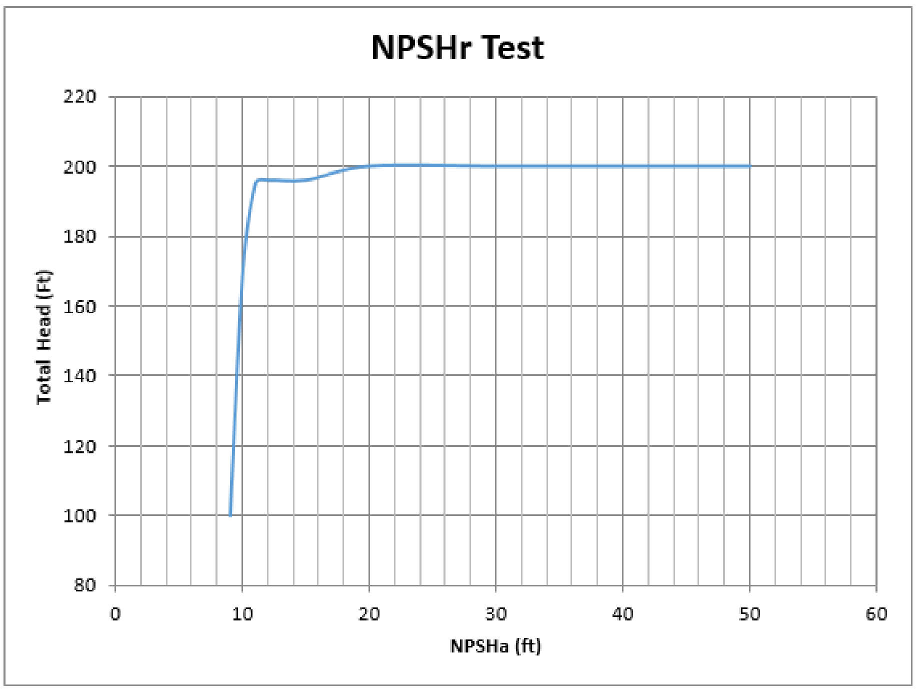

NPSHr Test
Total Head (Ft)
220
200
180
160
140
120
100
80
0
10
20
30
40
50
60
NPSHa (ft)

Figure 37:

=|= data: steel3.csv =|=

=|= data: steel4.csv =|=

=|= data: steel5.csv =|=

=|= data: steel6.csv =|=

Non-Ferrous Pipe

Iron Pipe

LaTex Quick start

All LaTex equations must be open and closed with `$$` characters.

LaTex is a *typesetting* language. Plain text *is* LaTex. The following:

`$$ x = y + z $$`

… renders like this:

=+=

$$x = y + z$$

=+=

The equations are automatically centered on the HTML page - however this is something we can change later if there is a need.

Fractions and grouping

Fractions are written using the `\over` command:

`$$ x = y \over z $$`

=+=

$$x = y \over z$$

=+=

Notice that the "over" part won't necessarily group things the way you want. The `{` and `}` braces allow you to define groups of text.

`$$ x = { y \over z}$$`

=+=

$$x = \frac{y}{z}$$

=+=

With the `{` and `}` in place, the `\over` command now works as expected - forming a fraction with only `y` and `z`.

Combinations of the grouping and `\over` command lets you define much more complex expressions:

```
$$ x = {{y \over z} + 6 \over {y +9}}$$
```

=+=

$$x = \frac{\frac{y}{z} + 6}{y + 9}$$

=+=

Notice how the `\over` between 6 and `{y+9}` puts a bar between everything to the left, within the group, and everyhing to the right.

Subscripts and Exponents

Subscripts are supported using `_` and supercripts / exponents use `^`.

```
$$ x_c = y^7 $$
```

=+=

$$x_c = y^7$$

=+=

Again, the grouping operator can allow you to create complex subscripts and exponents.

```
$$ x_{c+9} = y^{{x+6} \over 7} $$
```

=+=

```
$$ x_{c+9} = y^{{x+6} \over 7} $$
```

=+=

Mathematical Operators

Addition and subtraction use `+` and `-` as would be expected. For multiplication, you may omit the operator entirely.

```
$$ x_c = y^7z + 8x $$
```

=+=

$$x_c = y^7 z + 8x$$

=+=

Division will typically be noted using the `\over` command as shown above, but if you want to use a division sign, you may do so with the `\div` command.

```
$$ x = 15 \div 2 $$
```

=+=

$$x = 15 \div 2$$

=+=

Note the pattern, like `\over`, the `\` is used to designate a special operation/symbol. There are many supported - here are a few that are quite common.

```
\alpha
\beta
\gamma
\delta
\eta
\theta
\lambda
\omega
\parallel
\times
\bigtriangleup
\cdot
\measuredangle
\Longrightarrow
\infty
\sin
\cos
\log
```

=+=

$$\alpha\beta\gamma\delta\eta\theta\lambda\omega \parallel \times \bigtriangleup \cdot \measuredangle \Longrightarrow \infty \sin \cos \log$$

=+=

There is a complete list of mathematical notation symbols found here

Math functions

For things like square roots and logorithms, you must remember to use the grouping operator `{ }` to ensure the right parts of the equation are included in the layout.

For example:

```
$$ \sqrt 9 + x $$
```

=+=

$$\sqrt 9 + x$$

=+=

If you wanted to compute the square root of 9+x instead, you'd write it as such

```
$$ \sqrt {x + 9} $$
```

=+=

$$\sqrt {x + 9}$$

=+=

Complex example

Below is the friction factor equation, which is fairly representative of what the above commands can accomplish.

```
$$ {1 \over \sqrt f} = -2 \log({\epsilon \over 2.7D_h} + {2.51 \over Re \sqrt f}) $$
```

=+=

$${1 \over \sqrt f} = -2 \log({\epsilon \over 2.7D_h} + {2.51 \over Re \sqrt f})$$

=+=

Unit Switching

Equations specific for US units of measure should have a `[units = us]` line directly above the `$$` LaTex equation. See the underlying source code…

Equations specific for Metric units of measure should have a `[units = metric]` line directly above the `$$` LaTex equation. This is case sensitive, and should be written exactly as specified.

Switch units to see this in action.

=+= [units = us]

$$velo = {y_{ft} \over z_{sec}}$$

=+=

=+= [units = metric]

$$velo = {y_{meters} \over z_{sec}}$$

=+=

Units are optional, leave out the `[units=something]` entirely to allow the equation to be shown for all units.

=+=

$$velo = \frac{y_{distance}}{z_{time}}$$

=+=

Unit Element with Subscripts and Superscripts

When using the `<units>` element, you **cannot** embed `<sup>` and `<sub>`. Surround the text with `^` or `_` symbols instead, and the app will render using the appropriate element.

Units in lists

Bullet and numbered lists support units of measure as well.

- The unit for distance is .
- The unit for weight is

Numbered lists work well too:

1. Small distances are in
2. Long distances are in

X Axis Labels

=/= title: Data Points data-us: datapoints_us.csv data-metric: datapoints_metric.csv x: 1 series: 3, 2 series_title_index: 0 =/=

Friction Loss of Nozzles

Friction Loss – Water

(Tables 1-31 will be replaced by calculator)

IIIB-4 – Friction Loss for Water in

=|= title: Table 1 – 1/8 Inch Nominal, Steel Schedule 40 (ID = 0.269 in, /D = 0.00669) data-us: flw1-us.csv data-metric: flw1-metric.csv =|=

(Transition to turbulent flow occurs between the

=|= title: Table 2 – 1/4 Inch Nominal, Steel Schedule 40 (ID = 0.364 in, /D = 0.00495) data-us: flw2-us.csv data-metric: flw2-metric.csv =|=

(Transition to turbulent flow occurs between the

IIIB-5 – Typical Resistance Coefficients for Valves and Fittings

Table 32(a)

Table 32(b)

Table 32(c)

Table 33 – Resistance Coefficients for Miter Bends at Reynolds Number 2.25 x 105

IIIB-5A – Resistance Coefficients for 90 Degree Bends of Uniform Diameter

IIIB-5B – Resistance Coefficients for 90 Degree Bends of Uniform Diameter and Smooth Surface at Reynolds Number 2.25 x 105

General

Fluid Flow – General

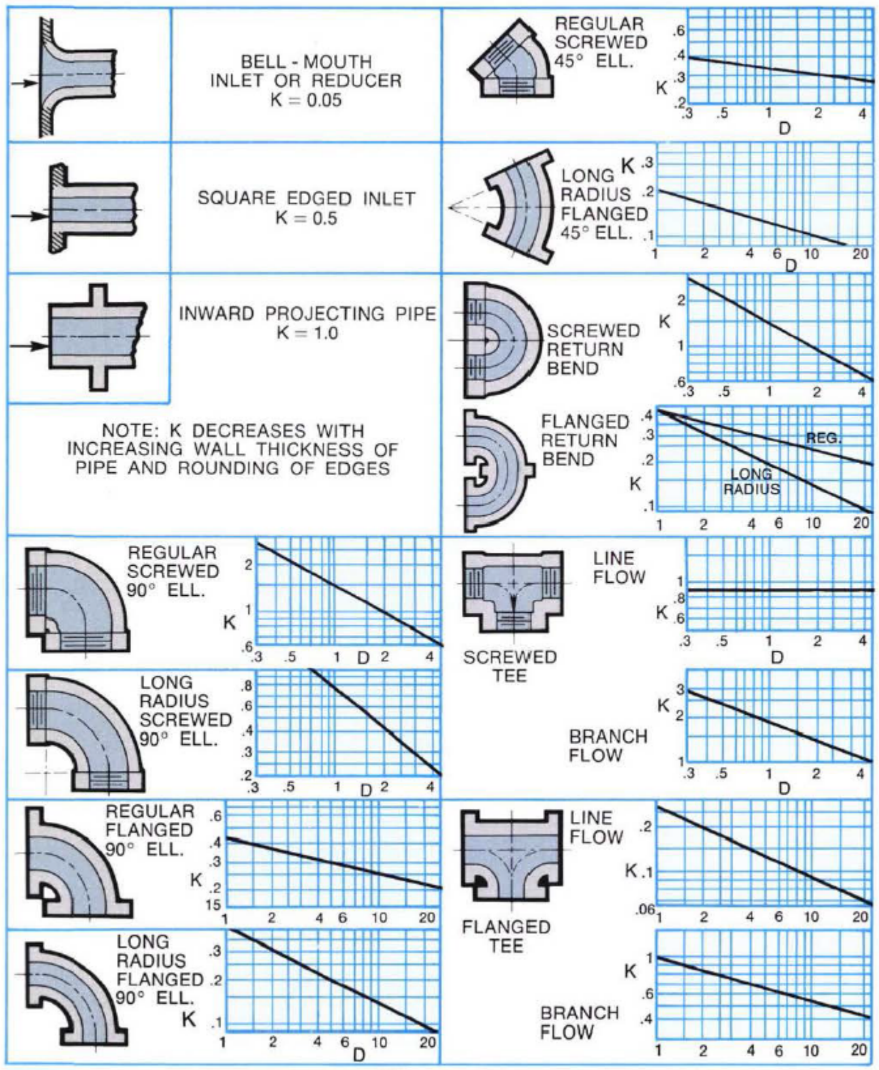
Pipe Friction (Major Losses)

The resistance to the incompressible flow of any fluid (head loss) in any pipe may be computed from the equation:

=+= (1)

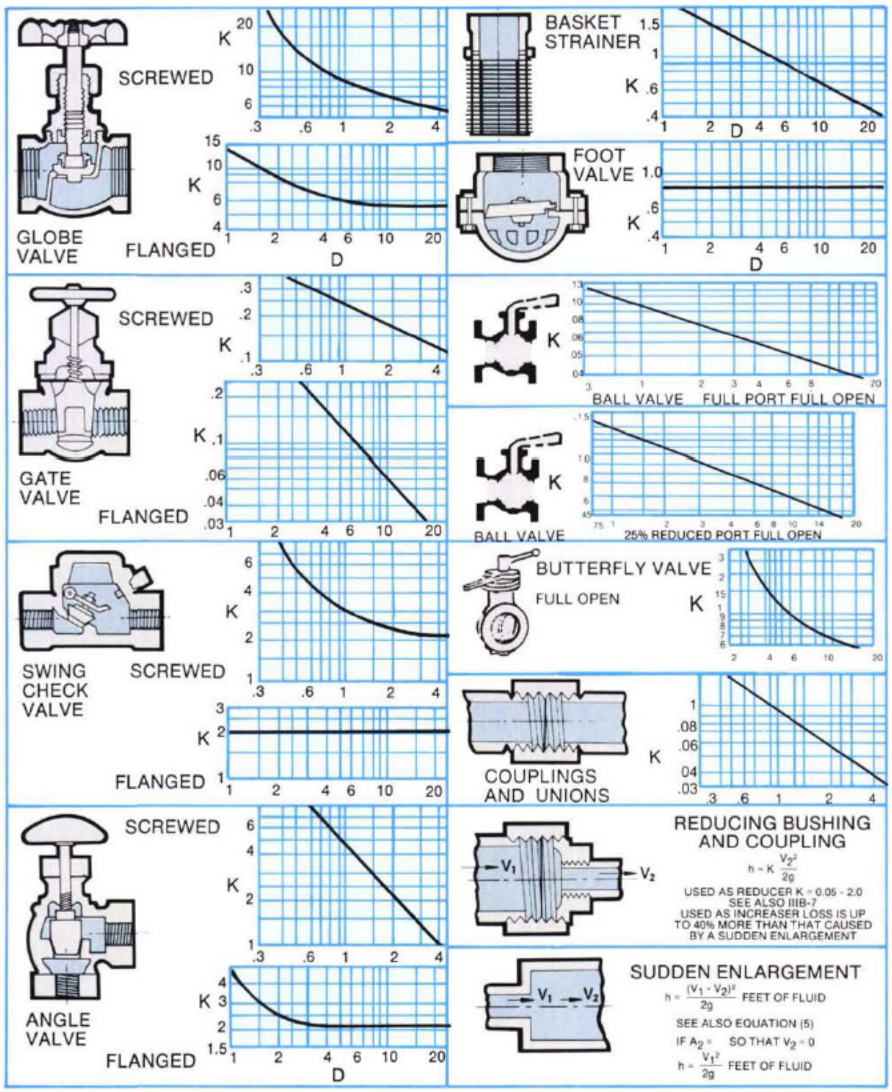
$$h_f = f * \frac{L}{D} * \frac{v^2}{2g}$$

=+=



$h = K \frac{V^2}{2g}$ FEET OF FLUID

Figure 38:



$$h = K \frac{V^2}{2g} \text{ FEET OF FLUID}$$

Figure 39:

Approximate Range of Variation For K		
Fitting		Range of Variation
90 Deg. Elbow	Regular Screwed	± 20 per cent above 2 inch size
	Regular Screwed	± 40 per cent below 2 inch size
	Long Radius, Screwed	± 25 per cent
	Regular Flanged	± 35 per cent
	Long Radius, Flanged	± 30 per cent
45 Deg. Elbow	Regular Screwed	± 10 per cent
	Long Radius, Flanged	± 10 per cent
180 Deg. Bend	Regular Screwed	± 25 per cent
	Regular Flanged	± 35 per cent
	Long Radius, Flanged	± 30 per cent
Tee	Screwed, Line or Branch Flow	± 25 per cent
	Flanged, Line or Branch Flow	± 35 per cent
Globe Valve	Screwed	± 25 per cent
	Flanged	± 25 per cent
Gate Valve	Screwed	± 25 per cent
	Flanged	± 50 per cent
Check Valve	Screwed	± 30 per cent
	Flanged	+ 200 per cent − 80 per cent
Sleeve Check Valve		Multiply flanged values by .2 to .5
Tilting Check Valve		Multiply flanged values by .13 to .19
Drainage Gate Check		Multiply flanged values by .03 to .07
Angle Valve	Screwed	± 20 per cent
	Flanged	± 50 per cent
Basket Strainer		± 50 per cent
Foot Valve		± 50 per cent
Couplings		± 50 per cent
Unions		± 50 per cent
Reducers		± 50 per cent

1. The value of D given in the charts is nominal IPS (Iron Pipe Size).
2. For velocities below 15 feet per second, check valves and foot valves will be only partially open and will exhibit higher values of K than that shown in the charts.

Figure 40:

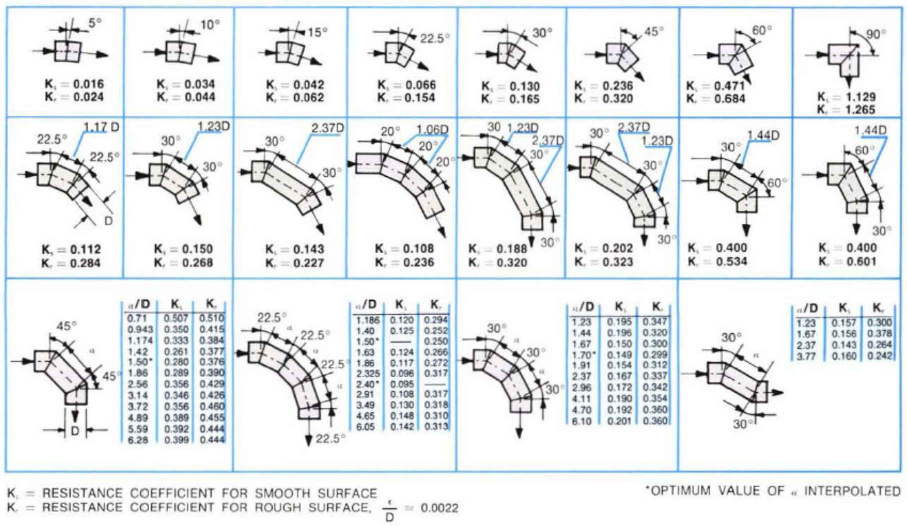


Figure 41:

where:

- hf = Frictional resistance (head loss) in
- L = Length of pipe in
- D = Average internal diameter of pipe in
- v = Average velocity in pipe in
- g = Acceleration due to gravity
- f = Friction factor

The Colebrook Equation (portrayed below) offers a reliable means for computing the **Darcy-Weisbach friction factor friction factor (f)** to be used in Equation (1).

=+= (2)

$$\frac{1}{\sqrt{f}} = -2\log_{10}(\frac{\epsilon}{(3.7*D)} + [\frac{2.51}{Re\sqrt{f}}])$$

=+=

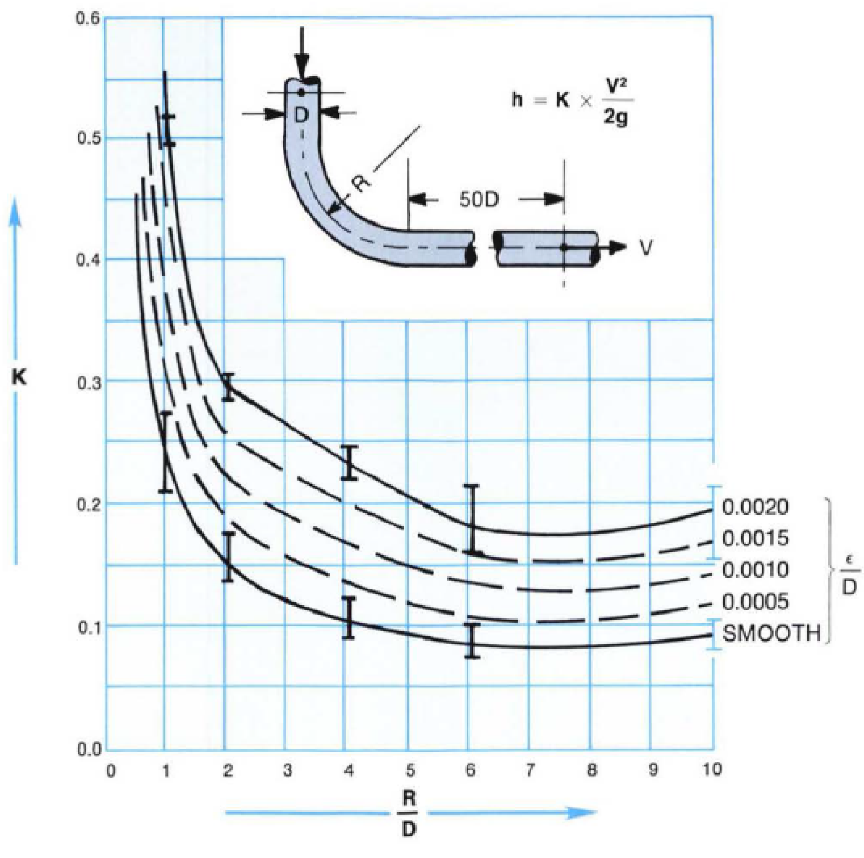
Another common form, which can be solved without iteration, is shown below.

=+=

$$\frac{1}{\sqrt{f}} = -2\log(\frac{\epsilon/D}{3.7} + \frac{5.74}{Re^{0.9}})$$

=+=

The **Reynolds number** (Re) is a non-dimensional ratio of inertial forces to viscous forces and is used to help scale data over a range of pipe sizes, fluid



h = K × V²/2g
D
R
50D
V
K
R/D
0.0020
0.0015
0.0010
0.0005
SMOOTH
ε/D
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0
1
2
3
4
5
6
7
8
9
10

Figure 42:

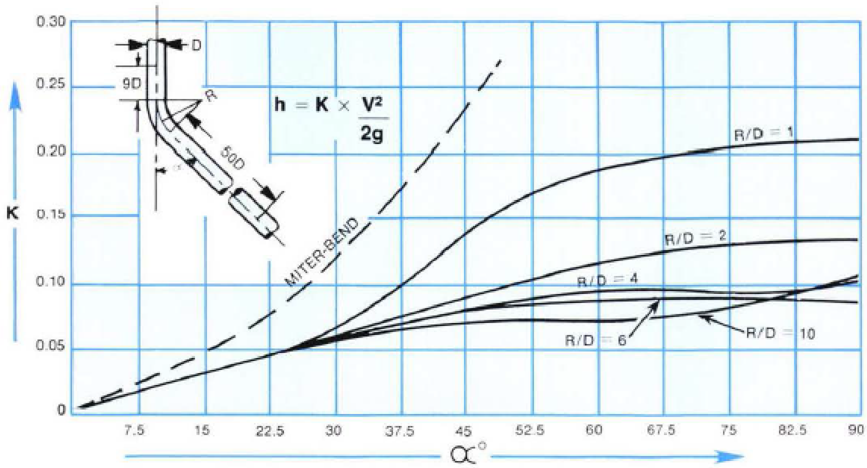


Figure 43:

properties, and flow conditions. It is used as the basis for the Moody Diagram to determine friction factors and pressure/head losses.

The Reynolds number is defined as:

=+=

$$Re = \frac{vD}{} = \frac{vD}{} = \frac{QD}{A}$$

=+=

where, for cylindrical pipes:

- is fluid density
- v is fluid velocity
- D is pipe inner diameter
- is dynamic viscosity
- is kinematic viscosity
- Q is volumetric flow rate
- A is pipe cross-sectional area

At Reynolds numbers less than about 2300, the flow tends to be laminar where it is traveling in a smooth, orderly manner with little mixing. At Reynolds numbers higher than about 4000, the flow is considered turbulent, with eddies forming and irregular motion.

Pipe Roughness

Pipe roughness varies with pipe material, age, usage, fluid transport and lining. This table gives example values for some clean materials.

=|= title: Roughness Values data-us: pipe-roughness-ft.csv data-metric: pipe-roughness-mm.csv =|=

Determining the frictional roughness for old pipe is beyond this tutorial. Deterioration of pipes with age depends on the particular chemical properties of the fluid and the metal with which it is in contact. It is recommended that prior experience or testing be used to determine an accurate value. For commercial installations, it is recommended that 15 percent be added to the values shown above.

References 1, 2, 3, 4, 5, 6 and 7 in Section VJ were studied to obtain the best value of the roughness parameter ( ) and the probable variations in the friction factors for new pipes.

Equation (2) was combined with Equation (1) and solutions carried out for each kind and size of pipe. These were used to construct large-scale logarithmic plots from which the values of (hf) shown in Section IIIB, Tables 1–31 incl., were obtained.

Tables of Friction Loss for Water, Explanation

Frictional resistances for water flowing in new, clean steel pipe (Schedule 40)* or in asphalt-dipped cast iron pipe are given in Section IIIB, Tables 1–31 incl., herein.*

The tables show the discharge in cubic feet per second, the average velocity in feet per second, and the velocity head in feet for any fluid in a circular pipe of the same diameter as that specified in each table for rates of flow in gallons per minute. The values of the friction head (hf) in feet of fluid per 100 feet of pipe apply to any fluid having a kinematic viscosity, v = 0.00001216 square feet per second (1.130 centistokes) which is the value for pure fresh water at 60° F. The friction heads are average values for pipes having the /D values given in the tables, where ( ) is a linear measure of the absolute roughness of the pipe walls and (D) is the internal diameter of the pipe. Further information on the roughness parameter is given in Section IV.

The tabulated values of (hf) are in feet of pure fresh water (60° F) per 100 feet of new clean steel pipe (Schedule 40)* or of new clean asphalt-dipped cast iron pipe as specified.

No allowance has been made for age, differences in diameter resulting from manufacturing tolerances, or any abnormal conditions of interior surface. Any factor of safety must be estimated from the local conditions and the requirements of each particular installation. An example illustrating the use of the tables will be found in Section IIIB.

To learn more about the effects of pipe friction on rotodynamic pumps, refer to ANSI/HI 14.3 – Rotodynamic Pumps for Design and Application.

Valves and Fittings (Minor Losses)

The resistance to flow (head loss) caused by a valve or fitting may be computed from the equation:

=+= (4)

$$h_f = K * \frac{v^2}{2g}$$

=+=

where:

- hf = Frictional resistance (head loss) in of fluid
- v = Average velocity in in a pipe of corresponding diameter
- g =
- K = Resistance coefficient for valve or fitting

Values of (K) for valves and fittings may be referenced below, and in Friction Loss – Water. Reference to the literature will reveal wide differences in the published values of (K) for all types of valves and fittings. The available data are inconclusive. As indicated in Section IIIB, flanged valves and fittings usually exhibit lower resistance coefficients than screwed valves and fittings. The resistance coefficients decrease with the increasing size of most valves and fittings.

Component (minor) losses can be summed together with the pipe losses to determine an overall frictional loss for the system, producing the equation

=+=

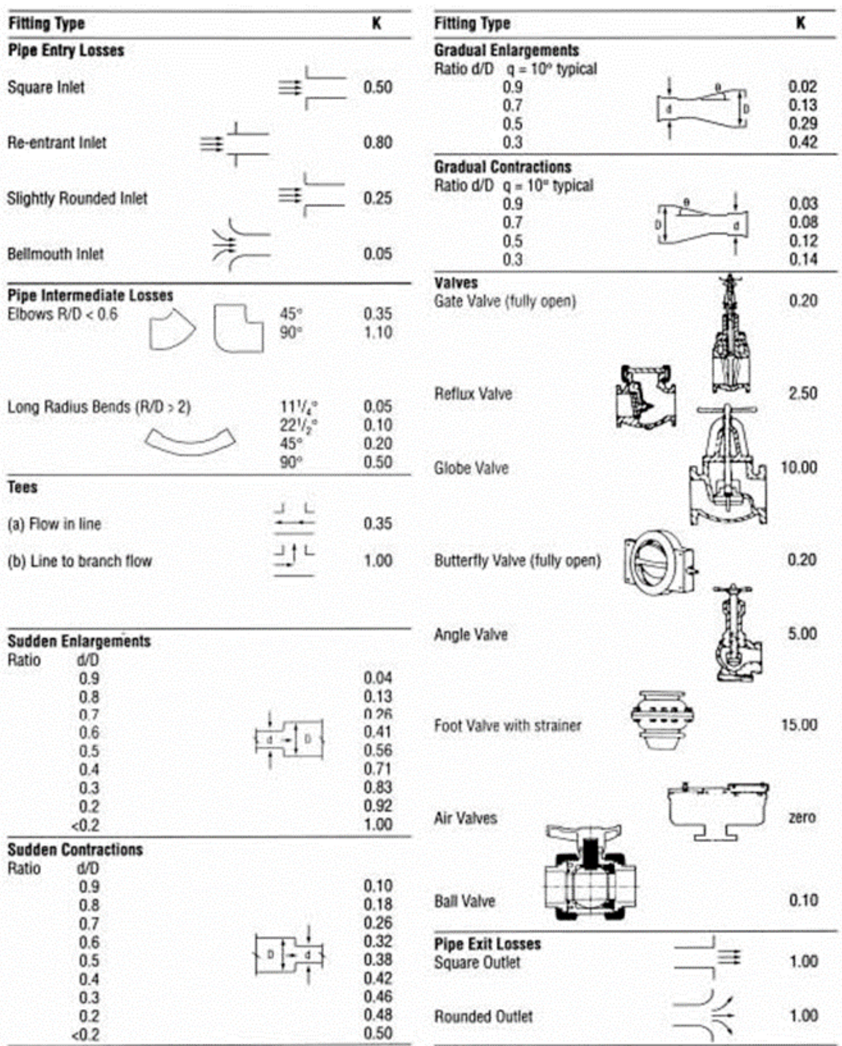
$$h_f = (\frac{fL}{D} + K) * \frac{v^2}{2g}$$

=+=

where:

- f = pipe friction factor
- L = pipe length in
- D = pipe inside diameter in
- ΣK = sum of the minor losses, which includes losses across valves

Cast iron flanged elbows and drainage-type elbows may be expected to approximate pipe bends. Values of the resistance coefficient (K) may be taken from Section IIIB. The solid line curves in Fig. IIIB-5A are given by Reference 12a of Section VI with the range of scatter of the test points as indicated. The broken



Fitting Type		K
Pipe Entry Losses		
Square Inlet		0.50
Re-entrant Inlet		0.80
Slightly Rounded Inlet		0.25
Bellmouth Inlet		0.05
Pipe Intermediate Losses		
Elbows R/D < 0.6	45°	0.35
	90°	1.10
Long Radius Bends (R/D > 2)	$11^1/_4$°	0.05
	$22^1/_2$°	0.10
	45°	0.20
	90°	0.50
Tees		
(a) Flow in line		0.35
(b) Line to branch flow		1.00
Sudden Enlargements		
Ratio	d/D	
	0.9	0.04
	0.8	0.13
	0.7	0.26
	0.6	0.41
	0.5	0.56
	0.4	0.71
	0.3	0.83
	0.2	0.92
	<0.2	1.00
Sudden Contractions		
Ratio	d/D	
	0.9	0.10
	0.8	0.18
	0.7	0.26
	0.6	0.32
	0.5	0.38
	0.4	0.42
	0.3	0.46
	0.2	0.48
	<0.2	0.50

Fitting Type		K
Gradual Enlargements		
Ratio d/D	q = 10° typical	
	0.9	0.02
	0.7	0.13
	0.5	0.29
	0.3	0.42
Gradual Contractions		
Ratio d/D	q = 10° typical	
	0.9	0.03
	0.7	0.08
	0.5	0.12
	0.3	0.14
Valves		
Gate Valve (fully open)		0.20
Reflux Valve		2.50
Globe Valve		10.00
Butterfly Valve (fully open)		0.20
Angle Valve		5.00
Foot Valve with strainer		15.00
Air Valves		zero
Ball Valve		0.10
Pipe Exit Losses		
Square Outlet		1.00
Rounded Outlet		1.00

Figure 44:

line curves may be used as a guide to probable resistance coefficients for intermediate values of the relative roughness factor /D. A value of = 0.00085 feet will be satisfactory for uncoated cast iron and cast steel elbows. Resistance coefficients for pipe bends with less than 90 degree deflection angles as reported by Wasielewski(g) 13 are shown in Fig. IIIB-5B. The curves shown are for smooth surfaces but may be used as a guide to approximating the resistance coefficients for surfaces of moderate roughness such as clean steel and cast iron. Figs. IIIB-5A and IIIB-5B in Section IIIB are not reliable below R/D = 1, where R is the radius of the elbow in feet. The approximate radius of a flanged elbow may be obtained by subtracting the flange thickness from the center-to-face dimension. The center-to-face dimension for a reducing elbow is usually identical with that of an elbow of the same straight size as the larger end.

The resistance coefficients for miter bends as reported by Shubert(g) 12b are shown in Section IIIB, Table 33. The rough pipe used in the Shubert investigation had a relative roughness factor, /D of about 0.0022. Reference 12b of Section VI. should be consulted for information on the variation of the resistance coefficients with variation in the Reynolds Number.

The resistance to flow (head loss) caused by a sudden enlargement may be computed from the equation:

=+= (5)

$$h_f = K\frac{(v_1 - v_2)^2}{2g}$$

=+= =+=

$$= K(1 - \frac{A_1}{A_2})^2\frac{v_1^2}{2g}$$

=+= =+=

$$= K[1 - (\frac{D_1}{D_2})^2]^2\frac{v_1^2}{2g}$$

=+= =+=

$$= K[(\frac{D_2}{D_1})^2 - 1]^2\frac{v_2^2}{2g}$$

=+=

where:

- h = Frictional resistance (head loss) in
- v1 = Average velocity in in the smaller (upstream) pipe
- A1 = Internal cross-sectional area of the smaller pipe in
- D1 = Internal diameter of the smaller pipe in
- v2, A2, D2 = Corresponding values for the larger (downstream pipe)
- g =
- K = Resistance coefficient, usually taken as unity since the variation is almost always less than ±3 per cent.

Equation (5) is useful for computing the resistance to flow caused by conical increasers and diffusers. Values of (K) for conical increasers based on data reported by Gibson(g) 14 are given in Section IIIB, Fig. IIB-6 or may be computed by the equation:

=+= (6)

$$K = 3.50(\tan(\theta/2))^{1.22}$$

=+=

where:

- = total conical angle of the increaser in degrees

Equation (6) applies only to values of between 7.5 and 35 degrees. Noteworthy is the fact that above 50 degrees a sudden enlargement will be as good or better than a conical increaser. Values of (k) for conical diffusers as reported by Reference 11 of Section VI are shown in Section IIIB, Fig. IIIB-6. The values shown include the entrance mouthpiece which accounts in part for the increase over Gibson's values for conical increasers. Resistance coefficients for reducers, as reported by Russell are given in Section IIIB, Fig. IIIB-7.

Friction Factor Diagrams

As previously stated, the resistance to the incompressible flow of any fluid (head loss) in any pipe may be computed from equation (1)

=+=

$$h_f = f * \frac{L}{D} * \frac{v^2}{2g}$$

=+=

Values of (f) may be obtained directly from Fig. IIIA-1 where the pipe is new clean asphalt-dipped cast iron, from Fig. IIIA-2 where the pipe is new clean steel of Schedule 40 wall thickness, or from Fig. IIIA-3 which applies to any size and type of surface. The probable variations in (f) for some classes of new clean pipe are given in Table A. It will be necessary to know the approximate value of the relative roughness factor, /D to enter Fig. IIIA-3 and this can be obtained, for several kinds of pipes, from Fig. llIA-4. If the fluid is fresh water at = 60° F or atmospheric air at 60° F the scales at the top of Figs. IIIA-1-3 incl., may be used. For convenience in air and water computations only, the scale reading is the product of the average velocity in feet/second and the internal diameter in inches, (VD").

For other fluids or temperatures the scales at the bottom of Figs. IIIA-1-3 incl., must be used. The scale reading is the Reynolds Number, Re, given by equation (3).

The data for Figs. IIIA-1-3 incl., were obtained directly from solutions of Equation (2). Figs. IIIA-4 and IIIA-5 were taken from Reference 2 with minor changes.

Values of the kinematic viscosity (v) at various temperatures are given in Fig. IIIA-5 for a number of different fluids. The Reynolds Number also may be obtained directly from Fig. IIIA-5 with the aid of the quantity (VD) mentioned above. The tracer line shows that for kerosene at 175° F flowing with an average velocity of 12.5 ft/sec in a pipe of 4 inches internal diameter, the Reynolds Number would be 3.5 x 10^5. In cases where viscosities are obtained from sources other than Fig. IIIA-5, it is *absolutely essential* that they be expressed in sq ft/sec in order that they may be used with Fig. IIIA-5 or Equation (3). Kinematic viscosities measured in stokes or centistokes may be converted to v in sq ft/sec by the formula:

=+= (7)

$$v = 0.00107639 * stokes$$

$$= 0.0000107639 * centistokes$$

=+=

Further information on viscosity can be found here.

If the Reynolds Number is less than 2000, the flow is laminar and the friction factor for any fluid in any pipe is given by the equation:

=+= (8)

$$f = \frac{64}{R}$$

=+=

If the Reynolds Number is above 4000, the flow will usually be turbulent and the **Moody Diagram** pictured below can be used to determine the friction factor. The range Re = 2000–4000 is called the critical zone in which the flow may be highly unstable and the friction factor indeterminate.

Moody Diagram

(Original data from L.F. Moody, "Friction factors for Pipe Flow", Trans. A.S.M.E., Vol 66, 1944) # Friction Loss of Paper Stock

I. Introduction

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 consists of pressure head, static head, velocity head and total friction head produced by friction between the pulp suspension and the pipe, bends,

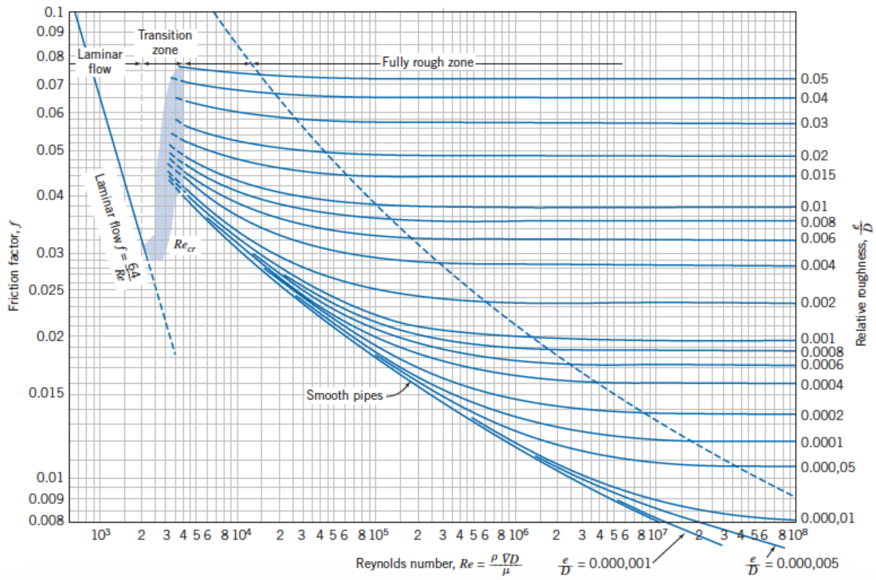


Figure 45:

and fittings. The total friction head is the most difficult to determine because of the complex, non-linear 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 recently published TAPPI Technical Information Sheet (TIS) 408-4 (Reference I), and is applicable to stock consistencies (oven-dried) from 2 to 6 percent. Normally, stock consistencies of less than 2% (oven-dried) are considered to have the same friction loss characteristic as water. This paper only applies to systems using centrifugal pumps which is normal for these consistencies. The method for determining the friction loss of pulp suspensions in pipe, as presented here, is intended to supersede the various methods previously issued.

II. Background

Figure 1 and Figure 2 show typical friction loss curves for two different consistencies (C2 < C1) 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 3.

Regions shown in Fig. 3 may be described as follows:

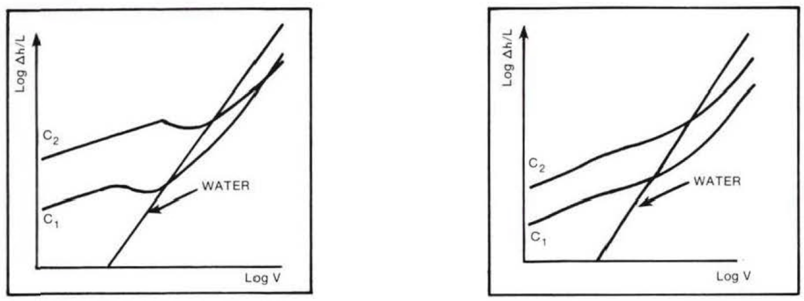


Figure 1—Friction loss curves for chemical pulp ($C_2 > C_1$). **Figure 2**—Friction loss curves for mechanical pulp ($C_2 > C_1$).

Figure 46:

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 at the upper limit of this linear region (Point B) is designated vmax.

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 vw.

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 4, is divided into only two regions: Regions 1 and 3. For this pulp type, the friction loss curve crosses the water curve vw and there is no true vmax ·

III. Design Parameters

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, etc. 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.

IV. Pipe Friction Estimation Procedure

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 3 (Point B) is designated vmax. The value of vmax is determined using Equation (1) and data given in Table I.

=+= (1) [units = us]

$$v_{max} = K'C(ft/s)$$

=+=

=+= (1) [units = metric]

$$v_{max} = 0.3048K'C(m/s)$$

=+=

where:

- K' = numerical coefficient (constant for a given pulp), obtained from Table I
- C = consistency (oven-dried, expressed as a percentage, *not* decimally), 2-6% limit.
- = exponent (constant for a given pulp), obtained from Table I

The following is data for use with Equation (1) to determine velocity limit, vmax ·

=|= title: Table I data: Table1CSV.csv =|=

NOTES:

1. Estimates for pulps based on published literature.
2. Original data obtained in stainless steel and PVC pipe.
3. Stainless steel may be hydraulically smooth although some manufacturing processes may destroy the surface and hydraulic smoothness is lost. PVC is taken to be hydraulically smooth pipe.
4. For cast iron and galvanized pipe, the K9 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 judgement is required.
6. Wood is New Zealand Kraft pulp.

If the proposed design velocity (v) is less than vmax, the value of flow resistance (Δh/L) may be calculated using Equation (2) and data given in Table II and the appendices.

=+= (2) [units = us]

$$h/L = FKvCD\,(ft/100\,ft)$$

=+=

=+= (2) [units = metric]

$$h/L = 0.3048FKvCD\,(m/100\,m)$$

=+=

where:

- F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix A)
- K = numerical coefficient (constant for a given pulp), obtained from Table II
- v = bulk velocity (ft/s)
- C = consistency (oven-dried, expressed as a percentage, *not* decimally), 2-6% limit
- D = pipe inside diameter (in), and
- , , = exponents (constant for a given pulp), obtained from Table II

For mechancial pulps, there is no true vmax · The upper limit of the correlation equation (Equation (2)) is also given by Equation (1) in this case, the upper velocity is actually vw ·

Region 2: The lower limit of Region 2 in Figure 3 (Point B) is vmax and the upper limit (Point 0) is vw. The velocity of the stock at the onset of drag reduction is determined using Equation (3).

=+= (3) [units = us]

$$v_w = 4.00C^{1.40}(ft/s)$$

=+=

=+= (3) [units = metric]

$$v_w = 1.2192C^{1.40}(m/s)$$

=+=

where:

- C = consistency (oven-dried,expressed as a percentage, not decimally).

If v is between vmax and vw, Equation (2) may be used to determine ΔH/L at the maximum point (vmax) · Because the system must cope with the worst flow condition, ΔH/L at the maximum point (v max) can be used for all design velocities between vmax and vw ·

Region 3: A conservative estimate of friction loss is obtained by using the water curve. (Δh/ L)w can be obtained from a Friction Factor vs. Reynolds Number plot (Reference 3, for example), or approximated from the following equation (based on the Blasius equation).

=+= (4) [units = us]

$$(h/L)_w = 0.58v^{1.75}D^{-1.25}\,(ft/100\,ft)$$

=+=

=+= (4) [units = metric]

$$(h/L)_w = 0.58v^{1.75}D^{-1.25}\,(m/100\,m)$$

=+=

where:

- v = bulk velocity (ft/s), and
- 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.

Pertinent equations, in addition to those herein presented, are located in Appendix A. Example problems are located in Appendix B.

V. Head Losses in Valves, Bends and Fittings

The friction head loss of pulp suspensions in valves, bends and fittings may be determined from the basic equation for head loss, provided in the general fluid flow section.

Values of K for the flow of water through various types of bends and fittings are tabulated in numerous reference sources (Reference 3, for example). The loss coefficient for valves may be obtained from the valve manufacturer.

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 percent for each 1 percent 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).

The following is data for use with Equation (2) to determine head loss, ΔH/L.

=|= title: Table II data: Table2CSV.csv =|=

NOTES:

1. Estimates for pulps based on published literature.
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.

Appendix A

The following gives supplemental information to that provided in the main text.

1) Rate of flow, Q —

=+= [units = us] (i)

$$Q = \frac{16.65(T.P.D.)}{C} \, (gpm)$$

=+=

=+= [units = metric] (i)

$$Q = \frac{3.782(T.P.D.)}{C} \, (m^3/h)$$

=+=

where:

- T.P.D. = mill capacity (short tons per day) and

- C = consistency (oven-dried, expressed as a percentage, *not* decimally).

2) Bulk velocity, v —

=+= [units = us] (ii)

$$v = \frac{0.321Q}{A}\,(ft/s),\ or$$

=+=

=+= [units = metric] (ii)

$$v = \frac{278Q}{A}\,(m/s),\ or$$

=+=

=+= [units = us] (iii)

$$v = \frac{0.4085Q}{D^2}\,(ft/s)$$

=+=

=+= [units = metric] (iii)

$$v = \frac{354Q}{D^2}\,(m/s)$$

=+=

where:

- Q = rate of flow
- A = inside area of pipe
- D = inside diameter of pipe

3) Multiplication Factor, F (included in Equation (2)) —

=+= (iv)

$$F = F_1 * F_2 * F_3 * F_4 * F_5,$$

=+=

where:

- F1 = correction factor for temperature. Friction loss calculations are normally based on a reference pulp temperature of 95°F. The flow resistance may be increased or decreased by 1 percent for each 1.8°F below or above 95°F, respectively. This may be expressed as follows (where T = pulp temperature :

=+= [units = us] (v)

$$F_1 = 1.526 - 0.00556T$$

=+=

=+= [units = metric] (v)

$$F_1 = 1.34808 - 0.010008T$$

=+=

- F2 = correction factor for pipe roughness. This factor may vary due to manufacturing processes of the piping, surface roughness, age, etc. Typical values for PVC and stainless steel piping are listed below (please note that these are typical values; experience and/or additional data may modify the above factors):

=+=

$$F_2 = 1.0\, for\, PVC\, piping$$

$$F_2 = 1.25\, for\, stainless\, steel\, piping$$

=+=

- F3 = correction factor for pulp type. Typical values are listed below (Note: This factor has been incorporated in the numerical coefficient, K, for the pulps listed in Table II. When using Table II, F3 should not be used.):

=+=

$$F_3 = 1.0\, for\, pulps\, that\, have\, never\, been\, dried\, and\, resulurried$$

$$F_3 = 0.8\, for\, pulps\, that\, have\, been\, dried\, and\, resulurried$$

=+=

- F4 = 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 F4 = 1.0, beating caused the freeness to decrease to 636 CSF and F4 to decrease to 0.96. Progressive beating decreased the freeness to 300 CSF and increased F4 to 1.37 (see K values in Table II). Some engineering judgement may be required.
- F5 = design safety factor. This is usually specified by company policy with consideration given to future requirements.

Appendix B

The following is an example that illustrates the method for determining pipe friction loss in each of the three regions shown in Figure 3.

Example

Solution:

a) The bulk velocity is:

=+= [units = us]

$$v = \frac{0.4085Q}{D^2}$$

=+=

=+= [units = metric]

$$v = \frac{354Q}{D^2}$$

=+=

=+= [units = us] (iii)

$$= \frac{0.4085(1100)}{6.065^2} = 12.22ft/s$$

=+=

=+= [units = metric] (iii)

$$= \frac{354(249.84)}{154.051^2} = 3.72m/s$$

=+=

b) It must be determined in which region (1, 2 or 3) this velocity falls. To obtain an initial indication, determine Vmax.

=+= [units = us] (1)

$$v_{max} = K'C,$$

=+=

=+= [units = metric] (1)

$$v_{max} = 0.3048 * K'C,$$

=+=

and

- K' = 0.59 (from Table I),
- = 1.45 (from Table I),

=+= [units = us]

$$v_{max} = 0.59(2.0^{1.45}) = 1.61ft/s.$$

=+=

=+= [units = metric]

$$v_{max} = 0.3048 * 0.59(2.0^{1.45}) = 0.49m/s.$$

=+=

c) Since v exceeds vmax, 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, vW, must be calculated.

=+= [units = us] (3)

$$v_w = 4.00C^{1.40},$$

=+=

=+= [units = metric] (3)

$$v_w = 1.2192C^{1.40},$$

=+=

=+= [units = us]

$$v_w = 4.00(2.0^{1.40}) = 10.56ft/s.$$

=+=

=+= [units = metric]

$$v_w = 1.2192(2.0^{1.40}) = 3.22m/s.$$

=+=

d) v exceeds vw, indicating that it falls in Region 3. The friction loss is calculated as that of water flowing at the same velocity.

=+= (4) [units = us]

$$(h/L)_w = 0.58v^{1.75}D^{-1.25}$$

=+=

=+= (4) [units = metric]

$$(h/L)_w = 0.58v^{1.75}D^{-1.25}$$

=+=

=+= [units = us]

$$= 0.58(12.22^{1.75})(6.065^{-1.25}) = 4.85\, ft\, head\, loss/100\, ft\, of\, pipe$$

=+=

=+= [units = metric]

$$= 0.58(12.22^{1.75})(6.065^{-1.25}) = 4.85\, m\, head\, loss/100\, m\, of\, pipe$$

=+=

This will be a conservative estimate, as the actual friction loss curve for pulp suspensions under these conditions will be below the water curve.

References

(1) TAPPI Technical Information Sheet (TIS) 408-4. Technical Association of the Pulp and Paper Industry, Atlanta. Georgia (1981).

(2) K. Molier and G. G. Duffy. TAPPI 61 , 1, 63 (1978).

(3) Hydraulic Institute Engineering Data Book, First Edition. Hydraulic Institute. Cleveland. Ohio (1979).

(4) K. Molier and G. Elmquist. TAPPI 63, 3, 101 (1980).

(5) W. Brecht and H. Heller. TAPPI 33, 9, 14A (1950).

(6) R. E. Durst and L. C. Jenness. TAPPI 39, 5, 277 (1956).

(7) K. Molier. G. G. Duffy and A. L. Titchener. APPITA 26, 4, 278 (1973).

(8) G. G. Duffy and A. L. Tichener. TAPPI 57, 5, 162 (1974).

(9) G. G. Duffy, K. Molier, P. F. W. Lee and S. W. A. Mine, APPITA 27, 5, 327 (1974).

(10) G. G. Duffy, TAPPI 59, 8, 124 (1976).

(11) G. G. Duffy. Company Communications Goulds Pumps. Inc. (1980-1981).

Calculation of Friction Loss for Other Fluids

The pipe friction charts, Figs. IIIC-2 thru IIIC-13 incl., show friction-loss moduli for the incompressible flow of viscous fluids, including water, in several sizes of new clean steel or wrought iron pipes having Schedule 40 wall thickness. Each chart covers the losses for a single size of pipe based on the kinematic viscosity in centistokes (cs). As viscosity is frequently given in Seconds Saybolt Universal, (SSu), corresponding rounded values at 100 °F are shown except in the case of low viscositites where no accurate SSU equivalents exist. Further information on viscosity and commonly accepted viscosity values for liquids will be found in the viscosity section. No allowance has been made for abnormal conditions of interior surface or installation nor for the deterioration with age. (See the general fluid flow section.)

Friction loss moduli for laminar flow are shown by the 45-degree lines in the upper left hand portion of each chart. Moduli for turbulent flow are shown by the steeper curves in the lower right hand portion. Both of these regions represent stable states of flow. A diagonal line separates the regions of laminar and turbulent flow and represents the critical zone, a region in which it is difficult to predict the state of flow and hence, the friction loss. The critical zone usually represents a region of unstable flow. The critical zone line gives approximate moduli on the high side for this region of unstable flow.

The bottom scale of each chart represents flow in gallons per minute, gpm. An auxiliary top scale shows the average velocity in the pipe in feet per second. Read vertically from the gpm scale to find the corresponding velocity in feet per second. The vertical scales, labeled "Friction Loss Modulus for 100 Feet of Pipe", represent values of the ratio

=+= (10)

$$M = \frac{p}{s}$$

=+=

where:

- M = Friction loss modulus for 100
- Δp = Pressure loss in
- s = Specific gravity of fluid at

The loss due to pipe friction may be obtained as follows:

=+= (11)

$$p = M * s$$

=+=

and

=+= (12)

$$h_f = 2.31M$$

=+=

where:

- hf = Friction head loss in

The other quantities are listed under Equation (10).

To use the charts, proceed as follows:

a) Select the chart for the size of pipe in question.

b) Follow the vertical line representing the flow rate to its intersection with the desired viscosity curve, and read the modulus at the left.

c) If the vertical line representing the flow rate does not intersect the viscosity line in either turbulent or laminar flow, use the intersection with the critical zone line.

d) Compute the friction loss in pressure drop or head, as desired, from Equations (11) or (12) above. These equations are repeated on each chart.

Volume of Tanks

Conversion Tables

Source

Ambler Thompson and Barry N. Taylor. "Guide for the Use of the International System of Units (SI)," Special Publication (NIST SP), National Institute of Standards and Technology, Gaithersburg MD, 20899.

Mechanical Friction in Line Shafts

Barometric Pressure – Effects of Altitude

Other Useful Information

Water Properties

=|= title: Water Properties at Various Temperatures data-us: water-properties-us.csv data-metric: water-properties-metric.csv =|=

=/= title: Specific Gravity vs. Temperature for Water data-us: water-properties-other.csv data-metric: water-properties-orig.csv x: 2 series: 5 series_title_index: 0 =/=

Water Saturation Properties

=|= title: Sat. Properties for Water (Liquid) data-us: liquid-water-us.csv data-metric: liquid-water-metric.csv =|=

=|= title: Sat. Properties for Water (Vapor) data-us: vapor-water-us.csv data-metric: vapor-water-metric.csv =|=

=/= title: Water Vapor Saturation Curve
data-us: vapor-water-us.csv data-metric: vapor-water-metric.csv x: 1 series: 2 series_title_index: 0 =/=

Auxiliary Data

Reference States, Default for Fluid

Enthalpy H =

Entropy S =

=|= title: Additional Fluid Properties data-us: auxiliary-us.csv data-metric: auxiliary-metric.csv =|=

Equation of States

The uncertainty in density of the equation of state is 0.0001% at 1 atm in the liquid phase, and 0.001% at other liquid states at pressures up to

The uncertainty in pressure in the critical region is 0.1%.

The uncertainty of the speed of sound is 0.15% in the vapor and 0.1% or less in the liquid, and increases near the critical region and at high temperatures and pressures.

The uncertainty in isobaric heat capacity is 0.2% in the vapor and 0.1% in the liquid, with increasing values in the critical region and at high pressures.

The uncertainties of saturation conditions are 0.025% in vapor pressure, 0.0025% in saturated liquid density, and 0.1% in saturated vapor density. The uncertainties in the saturated densities increase substantially as the critical region is approached.

Source

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved February 19, 2019).

Viscosity

Definitions and Methods of Measurement

The **viscosity** of a fluid (liquid or gas) is that property which tends to resist a shearing force. Since motion or flow of a fluid is produced by shearing forces, viscosity is associated with fluid motion. There is no relation between the viscosity and the specific gravity of most liquids. For instance, molasses having the same specific gravity (1.48) and the same Brix rating (90) may vary in viscosity from 128,000 to 303,000 Seconds Saybolt Universal (SSU). In rotodynamic pumps, fluid viscosity can have a significant impact on performance. ANSI/HI 9.6.7 acts as a guideline that explains these effects.

There are two basic viscosity parameters: **dynamic (*or absolute*) viscosity** and **kinematic viscosity**. The dynamic viscosity may be defined with the aid of Fig. 1 which shows two parallel plane surfaces of area (A) separated a distance (d) and the space between completely filled with fluid. A force (F) is applied to and in the plane of the upper surface, causing it to move with a velocity (v) parallel to the lower fixed surface. The velocity distribution will be linear over the distance (d) and experiments show that the slope of the velocity line (v/d) will be directly proportional to the unit shearing force (= F/A) for all "true" or "Newtonian" fluids. The proportionality factor ( ) is the dynamic viscosity. The foregoing may be expressed by the equations

=+=

$$= \frac{F}{A} = \frac{v}{d}$$

=+=

=+=

$$= \frac{}{v/d}$$

=+=

Therefore, the dimensions of the dynamic viscosity are

=+=

$$force \frac{time}{length^2}$$

=+=

The kinematic viscosity ( ) may be obtained by dividing the dynamic viscosity ( ) by the mass density ( ). The mass density is the specific weight (w) divided by the acceleration of gravity (g). These relationships may be expressed by the equation

=+=

$$= \frac{}{w/g} =$$

=+=

Therefore, the dimensions of kinematic viscosity are

=+=

$$\frac{length^2}{time}$$

=+=

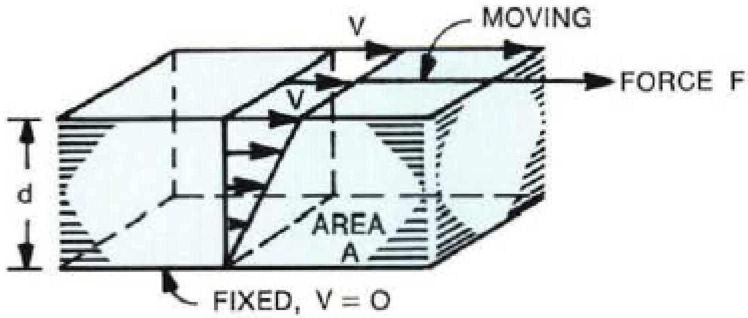


Figure 1

Figure 47:

The distinction between the dynamic and the kinematic viscosity should be carefully noted so that the correct parameter will be used as required in computations. Some useful relationships are as follows:

1 pound = 444,823 dynes

1 foot = 30.4800 centimeters

1 square foot = 929.034 square centimeters

1 dyne-second per sq cm = 1 poise = 100 centipoises

1 sq cm/sec = 1 stoke = 100 centistokes

1 lb-sec/sq ft = 478.801 poises = 47,880.1 centipoises

lb-sec/sq ft = (/47,880.1)centipoises = 0.0000208855 centipoises

sq ft/sec = sq cm/sec / 929.034 = 0.00107639 stokes

= / = /(w/g)

sq ft/sec = 0.000671970(/w)

where:

- = centipoises, and
- w =
- g =

The viscosities of most fluids vary appreciably with changes in temperature. The influence of change in pressure usually is negligible.

The viscosities of fluids, such as mineral oil and water, are unaffected by the magnitude and kind of motion to which they may be subjected as long as the temperature remains constant. Thus the ratio of shear stress to shear rate is a constant for all shear rates, is independent of time, and zero shear rate exists only at zero shear stress; such a fluid is said to be **Newtonian.**

When the ratio of shear stress to shear rate increases as the shear rate increases, reversibly and independent of time, a fluid is said to be **dilatent.**

When the shear stress to shear rate ratio is constant for shear rates above zero, is independent of time, but when shear occurs only for shear stress above a fixed minimum greater than zero, a fluid is said to be **plastic.**

When the ratio of shear stress to shear rate decrez. ses as shear rate increases, reversibly and independent of time, and zero shear rate occurs only at zero shear stress, a fluid is said to be **pseudo-plastic.**

When the ratio of shear stress t-0 shear rate decreases as shear rate increases and is time dependent in that this ratio increases back to its "rest" value gradually with lapse of time at zero shear rate and stress, and decreases to a limit value gradually with lapse of time at constant shear rate, a fluid is said to be **thixotropic.**

When the shear stress to shear ratio rate is constant for all shear rates at any given instant of time, but increases with time, a fluid is said to be **rheopectic.**

Viscosity is measured by an instrument called a **viscosimeter**. A definite volume of fluid is allowed to flow through a capillary tube or orifice of specified proportions and the time of efflux noted. Instruments of the capillary type, such as the Ostwald, Bingham, and Ubbelohde viscosimeters are used primarily for fluids of low viscosity, such as water. Instruments of the orifice type are used commercially for more viscous fluids such as petroleum products, and the time of efflux of the sample is taken as a measure of the viscosity. The Saybolt viscosimeter is commonly used in the United States, the Saybolt Universal for fluids of medium viscosity and the Saybolt Furol for those of high viscosity. The viscosity is expressed in Seconds Saybolt Universal (SSU) or Seconds Saybolt Furol (SSF). The relationship between Saybolt Universal viscosities and kinematic viscosities in centistokes is given in "ASTM Conversion Tables for Kinematic and Saybolt Universal Viscosities" or by the ASTM Standard, Designation: D446-85a*. Similar information for Saybolt Furol viscosities may be obtained from the ASTM Standard, Designation: D2161-87. The respective British counterparts of the Saybolt Universal and Saybolt Furol viscosimeters are the Redwood and Redwood Admiralty viscosimeters. The Engler viscosimeter is used extensively on the continent of Europe. Viscosimeters such as the Brookfield are particularly useful with non-Newtonian fluids. There are many other viscosimeters for special purposes, discussion of which is beyond the scope of this Manual. Viscosity conversion tables for use with the above described viscosimeters are shown in Tables IIC-3 and 4. A viscosity blending chart for use with oils is shown in IIC-2. Let oil (A) have the higher viscosity and oil (B) the lower viscosity. Mark the viscosity of (A) and (B) on the right and left hand scales, respectively, and draw a straight line connecting the marks as shown. The viscosity of any blend of (A) and (B) will be shown by the intersection of the vertical line representing the percentage composition and the line described above. See examples 1 and 2.

*American Society for Testing Materials, 1916 Race St., Philadelphia. Pa. 19103.

Viscosity Blending Chart

Viscosity Conversion Tables

The following tables will give an approximate comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

=|= title: Conversions data: visc-conv-1.csv =|=

=|= data: visc-conv-2.csv =|=

=+= *

$$Kinematic\,Viscosity\,(in\,centistokes) = \frac{Absolute\,Viscosity\,(in\,centipoises)}{Density\,(in\,g/cm^3)}$$

=+=

When the Metric System terms centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following expression can be used which will be sufficiently accurate for most calculations:

=+=

$$Kinematic\,Viscosity\,(in\,centistokes) = \frac{Absolute\,Viscosity\,(in\,centipoises)}{Specific\,Gravity}$$

=+=

When the English System units are used, the density must be used rather than the specific gravity.

For values of 70 centistokes and above, use the following conversion:

=+=

$$SSU = centistokes * 4.635$$

=+=

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:

=|= title: Conversion Factors data:SSU-conv.csv =|=

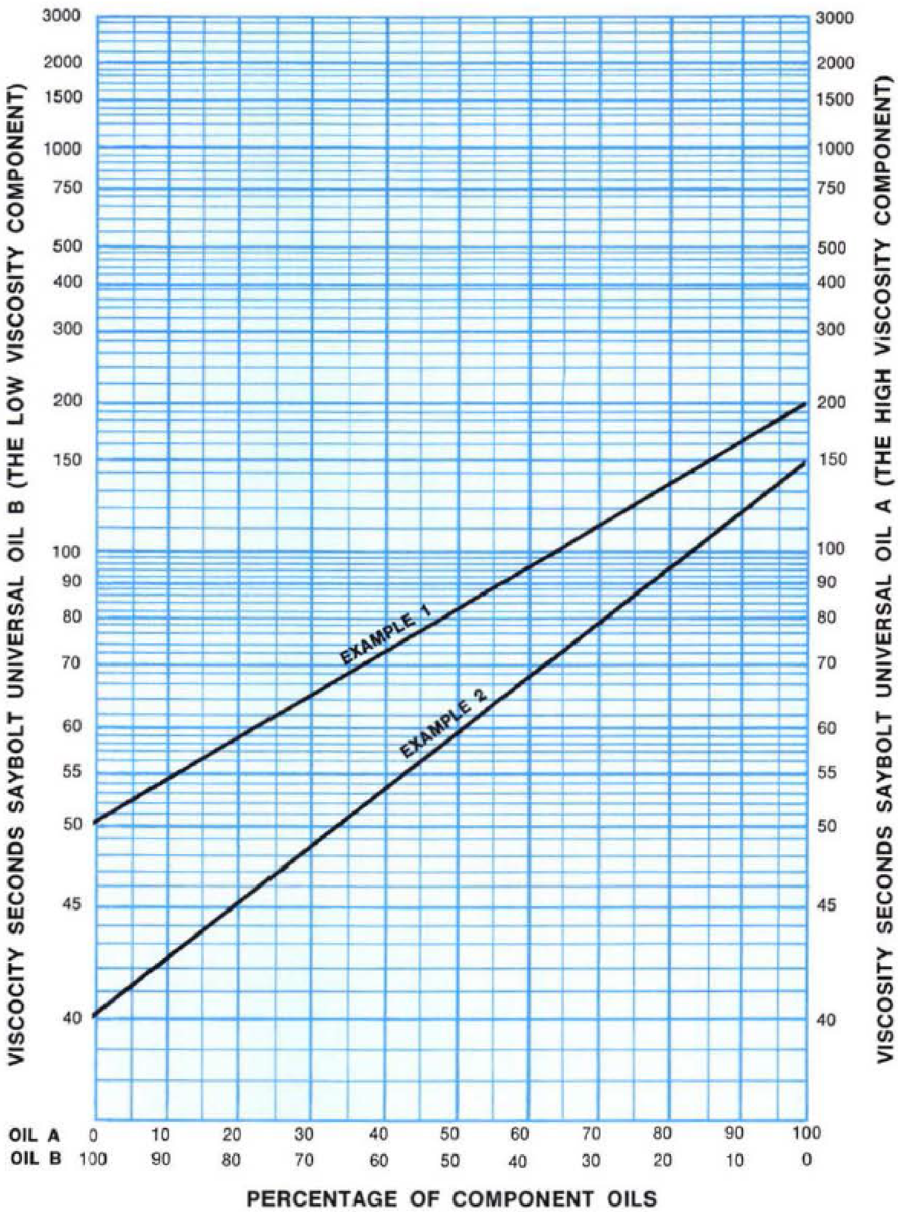

VISCOCITY SECONDS SAYBOLT UNIVERSAL OIL B (THE LOW VISCOSITY COMPONENT)
VISCOSITY SECONDS SAYBOLT UNIVERSAL OIL A (THE HIGH VISCOSITY COMPONENT)
3000
2000
1500
1000
750
500
400
300
200
150
100
90
80
70
60
55
50
45
40
EXAMPLE 1
EXAMPLE 2
OIL A 0 10 20 30 40 50 60 70 80 90 100
OIL B 100 90 80 70 60 50 40 30 20 10 0
PERCENTAGE OF COMPONENT OILS

Figure 48:

Properties of Other Fluids

Degrees A.P.I. vs. Specific Gravity for Oil

The relation of Degrees A.P.I. to Specific Gravity (g) is expressed by the following formula:

=+=

$$Degrees\,A.P.I. = \frac{141.5}{g} - 131.5$$

=+=

=+=

$$g = \frac{141.5}{131.5 + Degrees\,A.P.I.}$$

=+=

The following tables are based on the weight of 1 gallon (U.S.) of oil with a volume of 231 cubic inches at 60°F in air at 760 mm pressure and 50% humidity. Assumed weight of 1 gallon of water at 60°F in air is 8.32828 pounds.

To determine the resulting specific gravity by mixing oils of different specific gravities:

=+=

$$D = \frac{md_1 + nd_2}{m + n}$$

=+=

where:

- D = density or specific gravity of mixture
- m = proportion of oil of d1 density
- n = proportion of oil of d2 density
- d1 = specific gravity of m oil
- d2 = specific gravity of n oil

=|= title: Degrees A.P.I. vs. Specific Gravity data: IIB1-1.csv =|=

=|= data: IIB1-2.csv =|=

Temperature-Volume Relation for Oil

Specific Gravities vs. Degrees Baumé

Calculated from the formula, specific gravity 60°/60° F = 140 / (160 - Deg. Bé) =|= title: Spec. Gravities at 60°/60° F. Corresp. to Deg. Bé for Liquids Lighter than Water data: IIB2-1.csv =|=

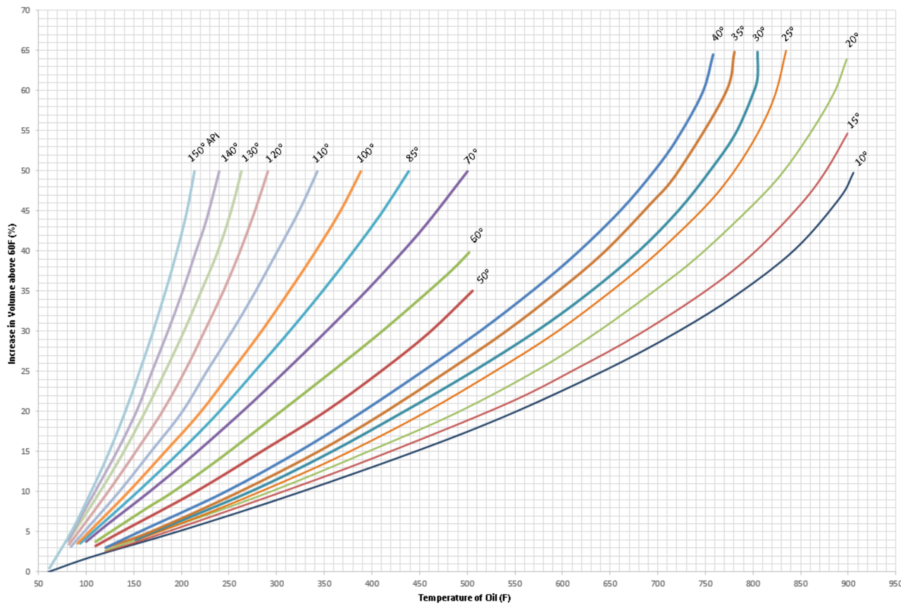


Figure 49:

Calculated from the formula, specific gravity 60°/60° F = 145 / (145 - Deg. Bé) =|= title: Spec. Gravities at 60°/60° F. Corresp. to Deg. Bé for Liquids Heavier than Water data: IIB2-2.csv =|=

Solids and Slurries

Useful Formulas

a. The formula for specific gravity of a solids-liquids mixture or slurry, Sm is:

=+=

$$S_m = \frac{S_s * S_1}{S_s + C_w(S_1 - Ss)}$$

=+=

Where:

- Sm = specific gravity of mixture or slurry
- S1 = specific gravity of liquid phase
- Ss = specific gravity of solids phase
- Cw = concentration of solids by weight
- Cv = concentration of solids by volume

Example: If the liquid has a specific gravity of 1.2 and the concentration of solids by weight is 35% with the solids having a specific gravity of 2.2 then:

=+=

$$S_m = \frac{2.2 * 1.2}{2.2 + .35(1.2 - 2.2)} = 1.43$$

=+=

b. Basic relationships among concentration and specific gravities of solid liquid mixtures are shown below.

In Terms Of	S_s, S_m, S_1	C_v	C_w
C_v	$\frac{S_m - S_1}{S_s - S_1}$		$C_w \frac{S_m}{S_s}$
C_w	$\frac{(S_m - S_1)}{(S_s - S_1)} \times \frac{S_s}{S_m}$	$C_v \frac{S_s}{S_m}$	

Figure 50:

Where pumps are to be applied to mixtues which are both corrosive and abrasive, the predominant factor causing wear should be identified and the materials of construction selected accordingly. This often results in a compromise and in many cases can only be decided as a result of test or operational experience. ANSI/HI 12.1-12.6 – Rotodynamic Centrifugal Slurry Pumps contains more information regarding the operation and applications of slurry pumps.

For any slurry pump application a complete description of the mixture components is required in order to select the correct type of pump and materials of construction.

=+=

$$C_w = \frac{weight\,of\,dry\,solids}{weight\,of\,dry\,solids\;+weight\,of\,liquid\,phase}$$

=+=

=+=

$$C_v = \frac{volume\,of\,dry\,solids}{volume\,of\,dry\,solids\;+volume\,of\,liquid\,phase}$$

=+=

A nomograph for the relationship of concentration to specific gravity of dry solids in water is shown in Figure IIB-5.

c. Slurry flow requirements can be determined from the expression:

=+= [units = us]

$$Q_m = \frac{4 * dry\, solids\, (in\, tons\, per\, hour)}{C_w * S_m}$$

=+=

=+= [units = metric]

$$Q_m = \frac{0.9085 * dry\, solids\, (in\, tons\, per\, hour)}{C_w * S_m}$$

=+=

Where:

- Qm = slurry flow
- 1 ton = 2000 lbs

Example: If 2,400 tons of dry solids is processed in 24 hours in water with a specific gravity of 1.0 and the concentration of solids by weight is 30% with the solids having a specific gravity of 2.7 then:

=+=

$$S_m = \frac{2.7 * 1.0}{2.7 + .3(1 - 2.7)} = 1.23$$

=+=

=+= [units = us]

$$Q_m = \frac{4 * 100}{.3 * 1.23} = 1,084\, GPM$$

=+=

=+= [units = metric]

$$Q_m = \frac{.9085 * 100}{.3 * 1.23} = 246\, m^3/h$$

=+=

d. Abrasive wear:

Wear increases rapidly when the particle hardness exceeds that of the metal surfaces being abraded. Always select metals with a higher relative hardness to that of the particle hardness. There is little to be gained by increasing the hardness of the metal unless it can be made to exceed that of the particles. The effective abrasion resistance of any metal will depend on its position on the mohs or knoop hardness scale. The relationships of various common ore minerals and metals is shown in Figure IIB-4.

Wear increases rapidly when the particle size increases. The life of the pump parts can be extended by choosing the correct materials of construction.

Sharp angular particles cause about twice the wear of rounded particles.

Austinetic maganese steel is used when pumping large dense solids where the impact is high.

Hard irons are used to resist erosion and to a lesser extent impact wear.

Elastomeric materials are used when pumping concentrations of fine material but total head is usually restricted to about

Castable ceramic materials have excellent resistance to cutting erosion but impeller tip velocities are usually restricted to

Classification of pumps according to particle size is shown in Figure IIB-6.

Approximate Comparison of Hardness Values of Common Ores and Minerals

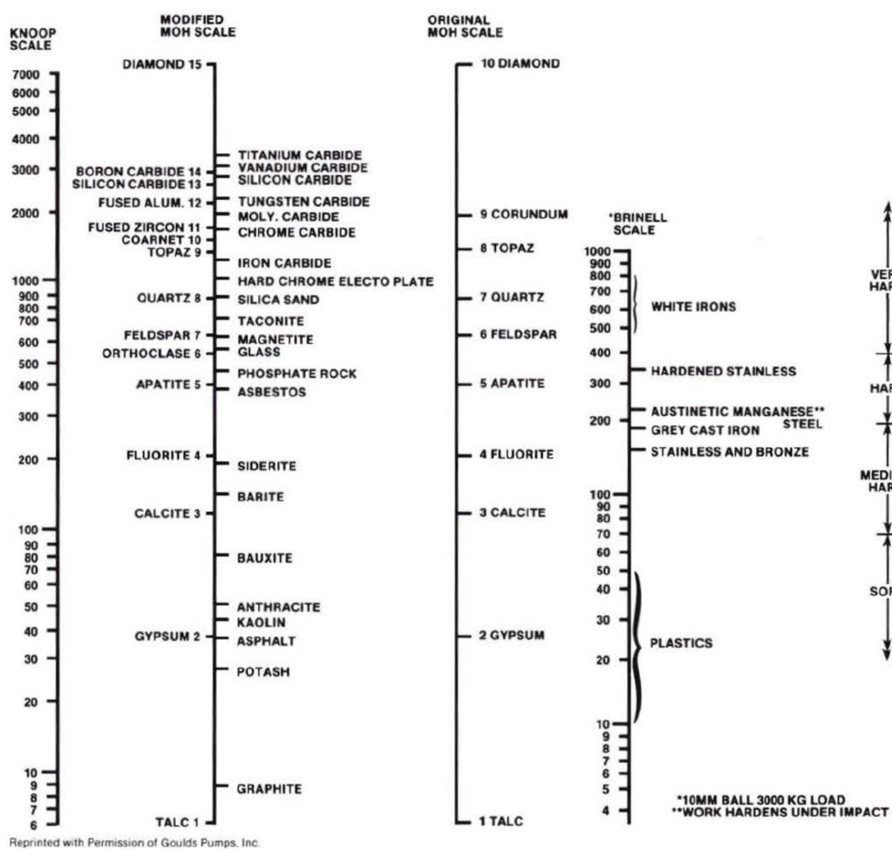


Figure 51:

Nomograph of the Relationship of Concentration to Specific Gravity in Aqueous Slurries

Classification of Pumps According to Solid Size

Vapor Pressure of Liquid H2

=|= title: Data Points data-us: VP-H2.csv data-metric: VP-H2-met.csv =|=

=/= title: Liquid Hydrogen Vapor Pressure Plot data-us: VP-H2.csv data-metric: VP-H2-met.csv x: 1 series: 2 series_title_index: 0 =/=

Vapor Pressure of Helium

=|= title: Data Points data-us: VP-helium.csv data-metric: VP-helium-met.csv =|=

=/= title: Helium Vapor Pressure Plot data-us: VP-helium.csv data-metric: VP-helium-met.csv x: 1 series: 2 series_title_index: 0 =/=

(Based on water having 1.00 specific gravity at 68°F., corresponding to a weight of 62.344 lb./cu. ft., and 1 psi equalling 2.310 feet.)

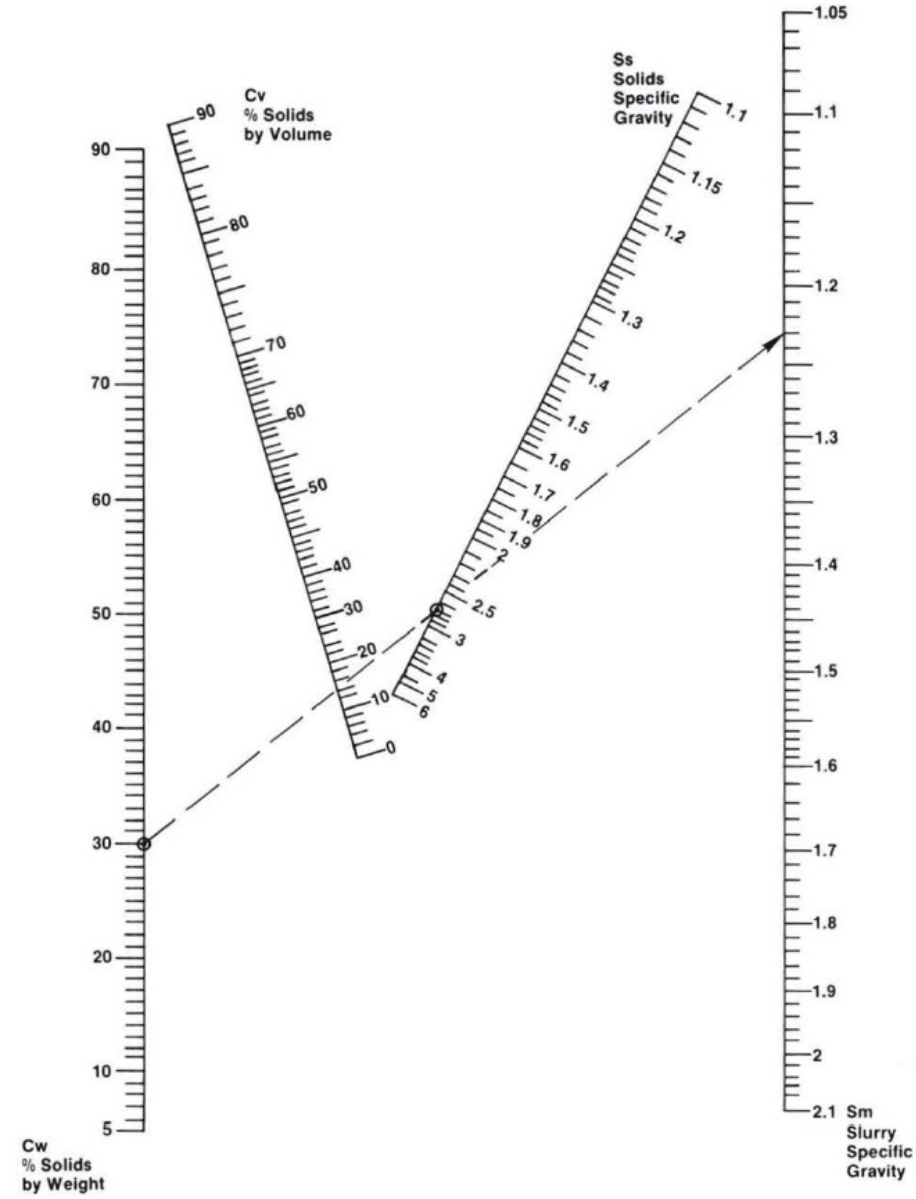
Sources

H2

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved July 31, 2019).

Helium

Eric W. Lemmon, Mark O. McLinden and Daniel G. Friend, "Thermophysical Properties of Fluid Systems" in NIST Chemistry WebBook, NIST Standard Reference Database Number 69, Eds. P.J. Linstrom and W.G. Mallard, National Institute of Standards and Technology, Gaithersburg MD, 20899, https://doi.org/10.18434/T4D303, (retrieved July 31, 2019).



Cv
% Solids
by Volume
Ss
Solids
Specific
Gravity
90
80
70
60
50
40
30
20
10
5
Cw
% Solids
by Weight
0
1.1
1.15
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2
2.5
3
4
5
6
1.05
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2
2.1
Sm
Slurry
Specific
Gravity

Figure 52:

Sieve/Mesh Aperture Size Information				Grade
Aperture		Tyler Mesh	US Sieve Size	
Inch	mm			
160	4064	-	-	Very Large Boulders
80	2032	-	-	Large Boulders
40	1016	-	-	Medium Boulders
20	508	-	-	Small Boulders
10	254	-	-	Large Cobbles
3	76.2	-	-	Small Cobbles
2	50.8	-	-	
1.5	38.1	-	-	Very Coarse Cobbles
1.050	26.67	-	-	Coarse Gravel
0.883	22.43	-	-	
0.742	18.85	-	-	
0.624	15.85	-	-	Medium Gravel
0.524	13.31	-	-	
0.441	11.20	-	-	
0.371	9.423	-	-	
0.312	7.925	2.5	-	Fine Gravel
0.263	6.680	3	-	
0.221	5.613	3.5	3.5	
0.185	4.699	4	4	
0.156	3.962	5	5	Very Fine Gravel
0.11	2.794	6	6	
0.131	3.327	7	7	
0.093	2.362	8	8	
0.078	1.981	9	10	Very Coarse Sand
0.065	1.651	10	12	
0.055	1.397	12	14	
0.045	1.143	14	16	Coarse Sand
0.039	0.991	16	18	
0.0328	0.833	20	20	
0.0276	0.701	24	25	Medium Sand
0.0232	0.589	28	30	
0.0195	0.495	32	35	
0.0164	0.417	35	40	
0.0139	0.354	42	45	
0.0117	0.297	48	50	
0.0098	0.250	60	60	Fine Sand
0.00825	0.210	65	70	
0.0070	0.177	80	80	
0.0059	0.149	100	100	
0.0049	0.125	115	120	
0.0041	0.105	150	140	
0.0035	0.088	170	170	Silt Slimes
0.0029	0.074	200	200	
0.0025	0.063	250	230	
0.0021	0.053	270	270	Pulverized
0.0017	0.044	325	325	
0.0015	0.037	400	400	
-	0.025	*500		
-	0.02	*625		
-	0.01	*1250		
-	0.005	*2500		
-	0.001	*12500		Mud Clay
-	0.0005			
-	0.00024			

Dredge Pump

Sand & Gravel Pump

Sand Pump

Slurry Pump

Severe Duty Slurry Pump

* Theoretical Values

Micron = 0.001mm
Mil = 0.001in

Note: This tabulation is for general guidance only since the selection of pump type and materials of construction also depends on the local head to be generated and the barasivity of the slurry i.e. concentration, solids, specific gravity, etc.

Figure 53: