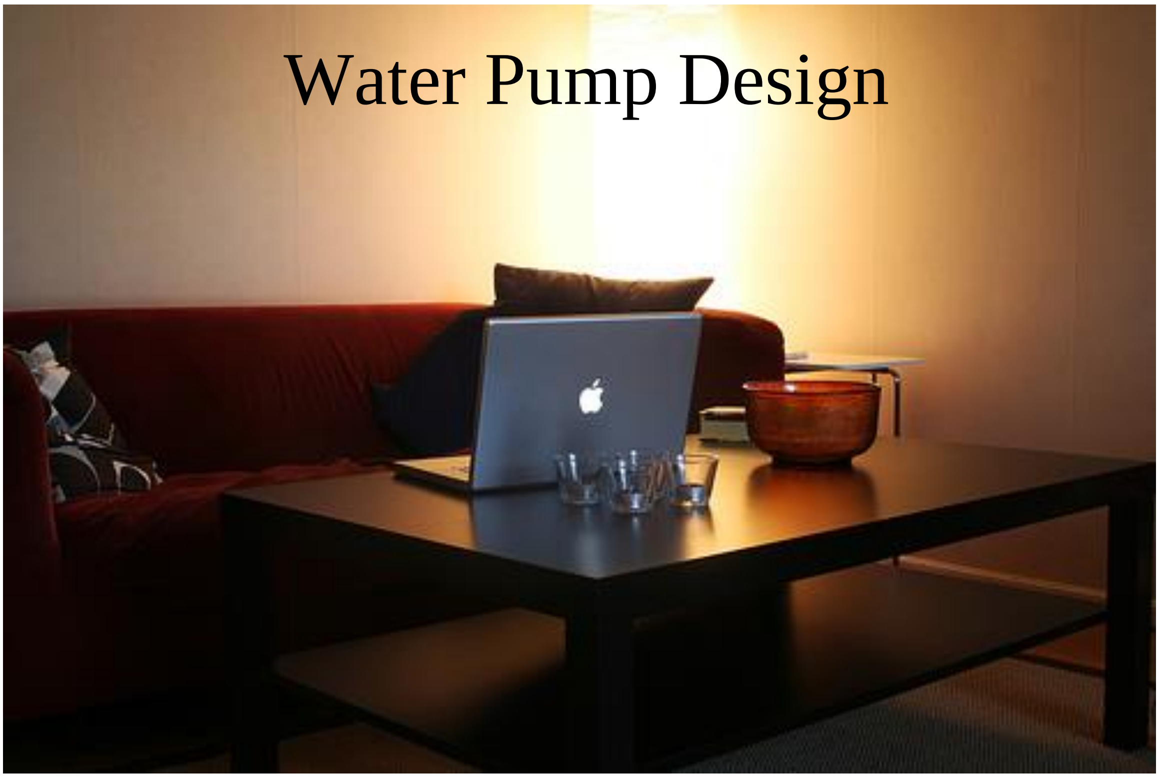

Water Pump Design

By

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PDHLibrary Course No 2023013
2 PDH HOURS

WATER PUMP DESIGN

Types of Pumps

Turbine Pumps

In wells, the pumps are nearly always turbine pumps. In very shallow wells, or wells with artesian pressure, end suction centrifugal pumps can be used, but these situations are rare. There are two basic types of turbine pumps. Figure 1 shows a schematic of a submersible pump. At the bottom of the pump assembly is the motor. It is important that submersible motors always be covered with water because the water provides cooling for the motor. Above the motor is a screen that allows water to enter the pipe. Sizing of this screen is important in well design, since the screen must have enough openings to allow the maximum amount of water into the pump bowls, while maintaining very low velocity. The low velocity is important to reduce the potential for sand and other sediment particles to reach the pump bowls. Above the screen are the pump bowls. These bowls take the energy from the motor and impart that energy into the water, increasing the pressure (or head) through each bowl. While some turbine pumps have a single bowl (or stage), many of them are multi-stage pumps. Each stage has a single impeller which adds additional head to the water, but the flow is the same through all stages. At the top of the pump assembly is the discharge head. The discharge head is connected to the transmission or distribution system piping to convey the water into the overall system. The discharge head can be above ground or below ground.

The other option for a well pump is a vertical turbine pump. The primary difference between these two types of pumps is the location of the motor. As shown in Figure 1, the motor for a submersible pump is at the bottom. With a vertical turbine pump, the motor is at the top. Figure 2 shows a vertical turbine pump. This figure does not show a screen, but if a screen is provided, it is at the very bottom of the pump assembly. Above the screen are the pump bowls. As with a submersible pump, there can be a single bowl (or stage) or multiple stages. Above the bowls are sections of pipe that extend to the floor of the pump building. Vertical turbine pumps are almost always placed in a building. There are two common exceptions to this – in climates where it never freezes and in applications such as irrigation where the pump only operates during non-freezing times and is drained for the winter. Above the floor is the discharge head for the pump and above the discharge head is the motor. Vertical turbine pumps are also very commonly used in treatment facilities, with the bowls set in a below ground storage tank (called a clearwell) and the discharge head and motor above the floor. Vertical turbine pumps are sometimes used in a booster pump station, but only if the booster pump station includes a below-ground reservoir so the pump bowls can still be set in a tank configuration.

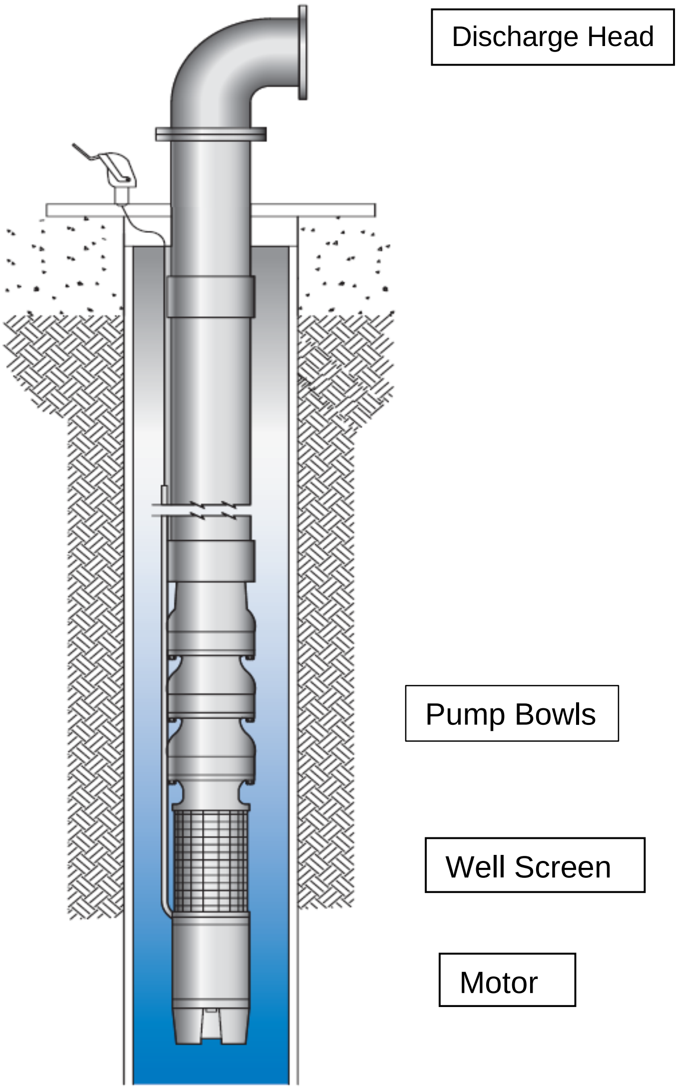

Figure 1 – Schematic of submersible pump

Figure 2 – Schematic of vertical turbine pump

Figure 3 – Vertical Turbine Pump in Service

Horizontal Booster Pumps

In a booster station, the pumps are usually connected to the distribution system piping, so the suction and discharge side are both under pressure. There are a number of different types of horizontal pumps used in this application. Figure 4 shows a schematic of an end suction centrifugal pump and Figure 5 shows a photograph of three end suction centrifugal pumps in service. The suction side of these pumps is at the end of the pump. These pumps can draw water from a reservoir below the centerline of the suction inlet, but they are much more efficient when the inlet has a positive head (or positive pressure). In Figure 5, these pumps take water from a pressure zone on the suction side which provides positive pressure and pump it to a storage tank at a higher elevation. The design of an end suction centrifugal pump limits it to a single impeller, unlike a turbine pump which can have multiple bowls, each with one impeller. Therefore, these pumps cannot typically produce as much increase in head as a turbine pump. The design of this pump also requires a 90-degree change in direction from the suction to the discharge side of the pump, so the piping must be designed to accommodate this change in direction.

Figure 4 – End Suction Centrifugal Pump

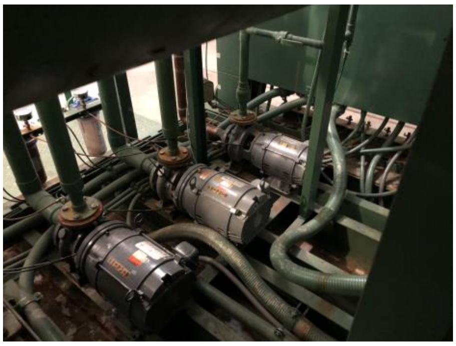

Figure 5 – Installed End Suction Centrifugal Pumps

Another type of horizontal pump is a split case pump (see Figure 6). Similar to an end suction centrifugal pump, this pump has a single impeller. With a split case pump, the suction line and the discharge line are 180-degrees from each other, but there is a horizontal offset so the suction line is not directly aligned with the discharge line.

Figure 6 – Horizontal Split Case Pump

Vertical Booster Pumps

Booster pumps can also be vertical turbine pumps. The pumps in Figure 7 are multi-stage vertical turbine pumps. These pumps are similar to the turbine pumps previously described, except these are specifically designed for booster stations and include suction and discharge pipes that are 180-degrees apart and on the exact same horizontal and vertical alignment.

Figure 7 – In-line Vertical Turbine Pump

Pump Curves

The performance of any pump is defined by a pump curve. The pump curve typically has head (energy) added to the water on the vertical axis and the flow rate on the horizontal axis. Head is typically expressed in terms of feet and flow in terms of gallons per minute (gpm). Pump curves will typically also show the efficiency of the pump at various points along the pump curve, the horsepower (HP) required and the Net Positive Suction Head (NPSH) required. Pump curves often show multiple curves on one graph. This can represent two different options. One option is it shows the performance of a turbine pump with multiple stages. The other option is it shows the performance of a pump with different size impellers.

Parallel Curves

Pump curves from the same pump, with either different sizes of impellers or a different number of impellers, are typically approximately parallel to each other. Figure 8 shows one example of a pump curve. The red angle identifies the design point for this pump – 1500 gpm at 150 feet of head. This pump curve shows the curve for several different impeller diameters. The top curve is for an impeller diameter of 13.5 inches and the bottom curve is an impeller diameter of 10". The blue line represents the curve for an impeller diameter of 12.375 inches. Depending on the manufacturer and the specifics of the pump, the 12.375-inch impeller may be a standard, or it may be a custom-made impeller. The process used to create a custom impeller is relatively simple and is often called "trimming" the impeller. The process starts with a larger standard impeller and the diameter of the impeller is reduced to the desired size to match the necessary head and flow requirements.

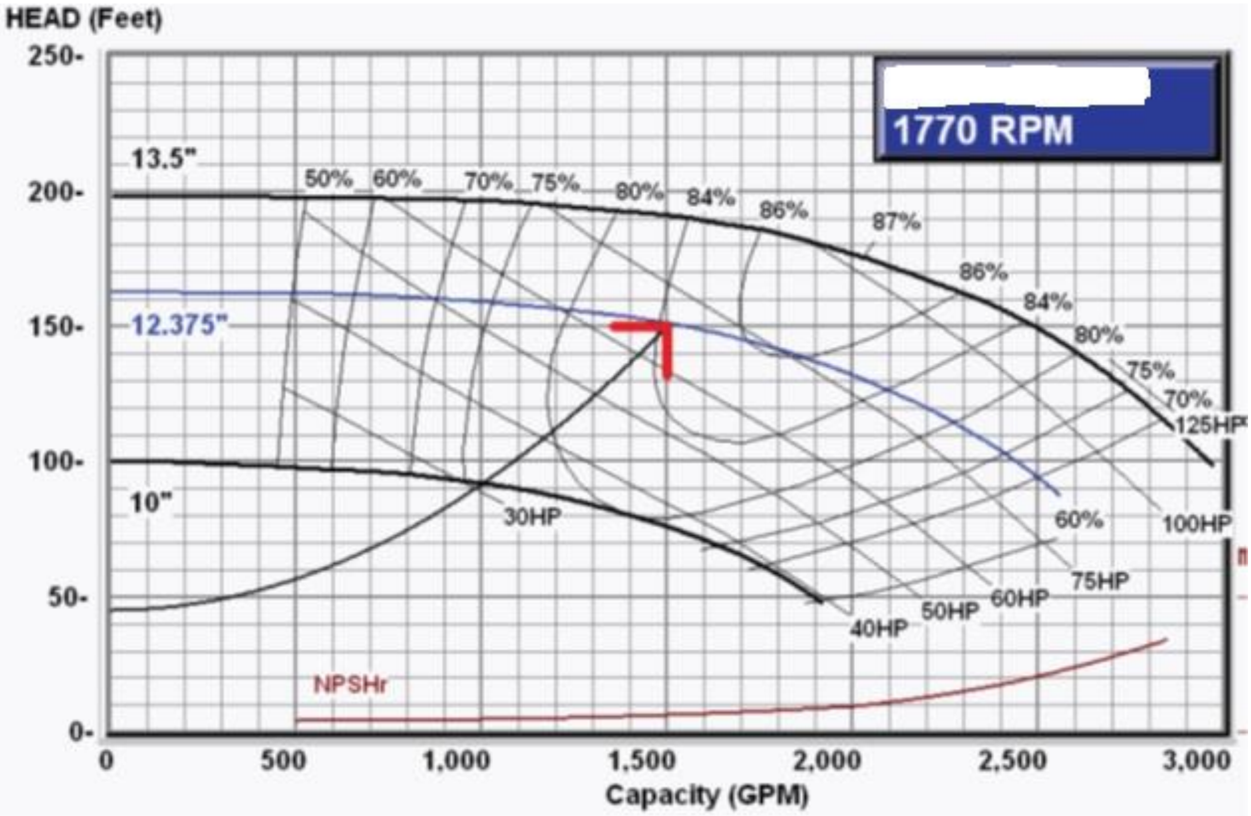

Figure 8 – Pump Curve with Design Point

Figure 9 shows an example of a series of pump curves that represent a turbine pump with multiple impellers. It includes 10 different combinations. This particular pump has two different impeller sizes – a standard size and a "trimmed" or smaller size. The label on each pump curve defines the total number of impellers and the number of trimmed impellers. For example, curve 4-1 has a total of four impellers, one of which is trimmed. Curve 3 has 3 impellers, all the standard size.

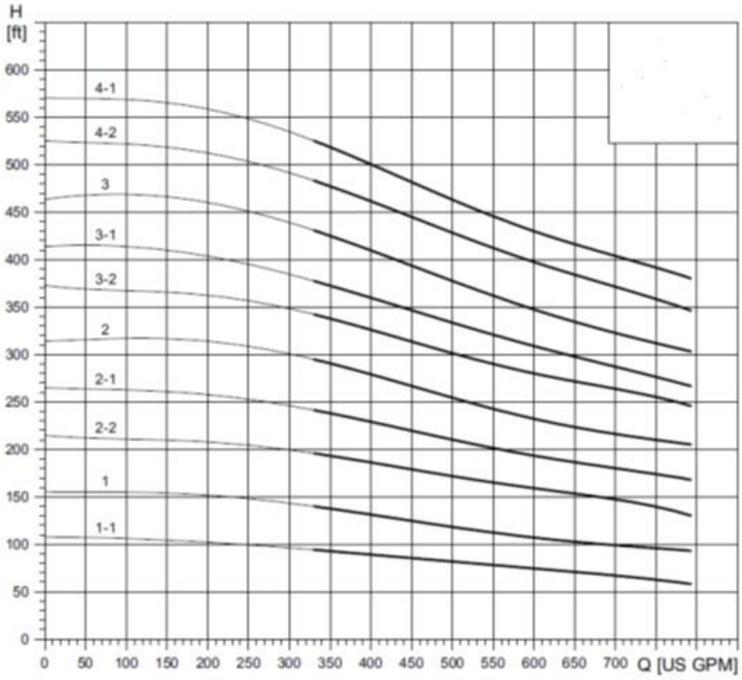

Figure 9 – Pump Curve for multiple impellers

Pump Curve Components

The information presented on a typical pump curve includes more than just the head vs. flow performance of the pump, and even this curve has useful information. Figure 10 highlights two important pieces of information shown on the pump curve. Circled in red on the left side of the pump curve is the shut off head point. The shut off head is the head produced by the pump at a flow of 0 gpm. For this pump, the shut off head is about 162 feet. If the static head from the pump exceeds 162 feet, the pump won't be able to produce any water under these conditions. For example, if the water level in the clearwell is elevation 1050 and the water level in the storage tank ranges from elevation 1200 to 1230, the tank will never fill beyond elevation 1212 (1050 + 162). When the tank is empty, the pumping head (not including friction loss) is 150 feet, so the pump could potentially produce 1500 gpm (depending on the magnitude of the friction loss). But as the tank starts to fill, the flow through the pump will be reduced and will go to 0 flow at elevation 1212, with the tank only 40% full.

Circled in blue on the right side of Figure 10 is the pump runout point. This point is about 2550 gpm at about 89 feet of head. This is also an undesirable operating point. While a pump can function beyond the runout point, it can create some undesirable conditions. Beyond the runout point, cavitation can occur within the pump and result in premature wear on the impeller and the bearings. It can also cause the motor to exceed the amperage rating of the motor which can result in overheating of the motor. This can result is shorter motor life or even failure of the motor. With these problems, it is important to be sure the pump does not operate beyond the pump runout point.

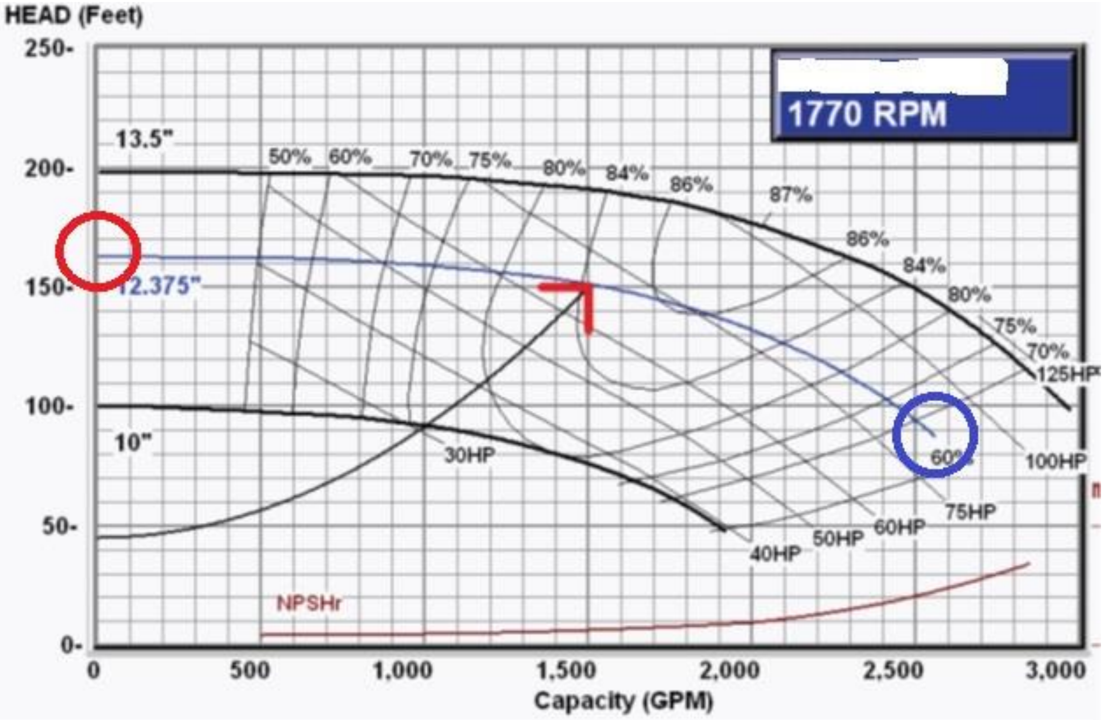

Figure 10 – Pump Curve with Shutoff Head and Pump Runout

Figure 11 highlights some additional points of importance on a pump curve. Circled in green is the point of highest pump efficiency. The pump efficiency is different for different impeller sizes. For the 12.375-inch impeller, the point of maximum efficiency, circled in green, is about 1800 gpm at 141 feet of head and 86% efficiency. With the larger 13.5-inch impeller, the point of maximum efficiency is about 2040 gpm at 175 feet of head and 87% efficiency. The blue circle shows the runout point on the pump curve. Note the efficiency at this point is only about 67%, much lower than the point of maximum efficiency. The pump curve also has a series of curves labeled 40HP, 50 HP, 60 HP, etc. These curves represent the motor horsepower requirements for various points on the pump curve. It is important to understand these motor horsepower requirements to select the proper motor size. For example, at the design point (the red angle), the motor horsepower requirements are between 60 HP and 75 HP, so a 75 HP motor would be indicated. However, it is possible the pump could operate anywhere along this pump curve. This may not be the case if the physical conditions of the system prevent operating at significantly lower head, which would be the case if most of the system head was associated with the elevation head required to pump to a storage tank. If much of the system head is associated with friction loss, the pump could operate anywhere on the pump curve. If the pump is operating near the pump runout point, this point is between 75 HP and 100 HP, so a 100 HP motor would be required. Unless the designer is positive the pump couldn't operate beyond the 75 HP curve, this pump should be provided with a 100 HP motor.

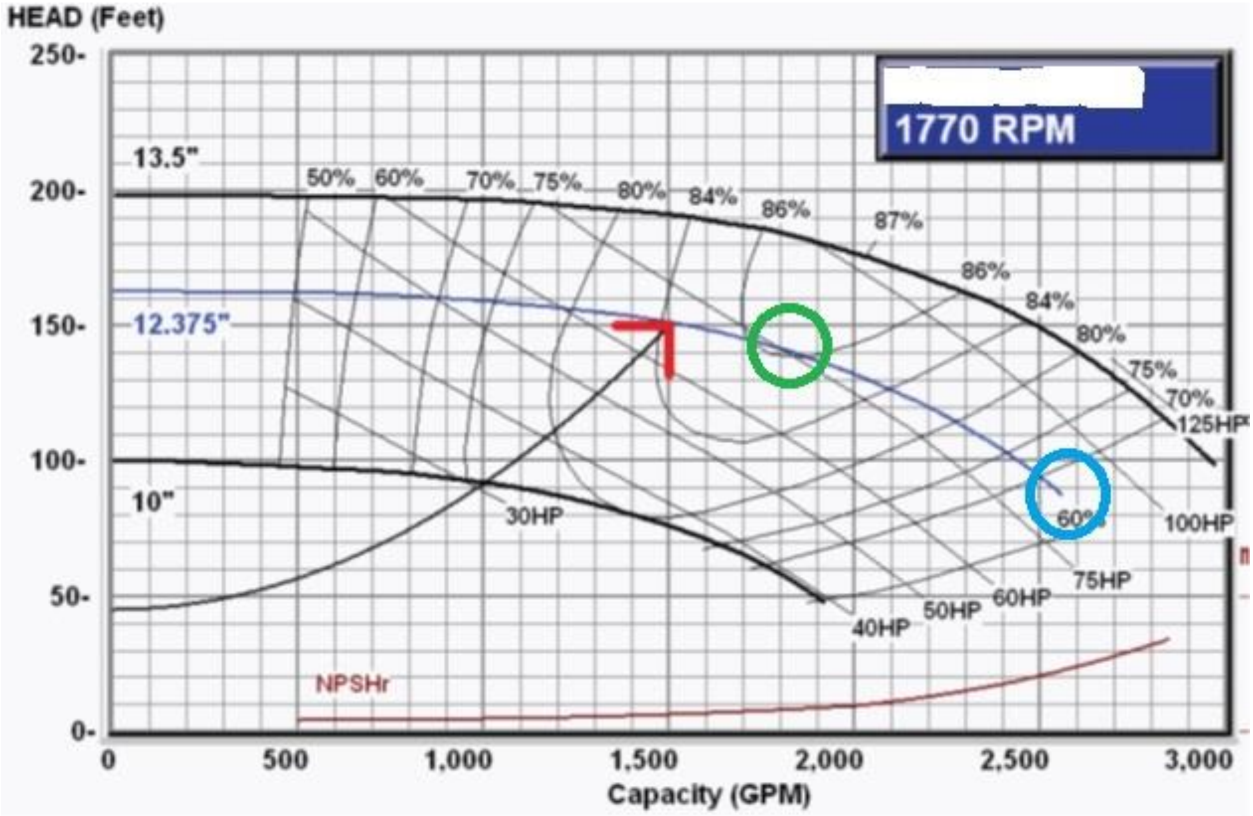

Figure 11 – Pump Curve with Point of Maximum Efficiency

Net Positive Suction Head

Net Positive Suction Head (NPSH) required can be an important design parameter. NPSH required is specific to each pump. The pump suction needs to be continuously submerged so that no air is drawn into the pump. If air is sucked into the pump impeller, the capacity is reduced. In addition, the air is likely to cause cavitation. This can sometimes create significant noise in the pump, and frequently causes cavitation. The cavitation will start to destroy the impeller and lead to premature failure of the pump.

The NPSH available includes a variety of factors, including:

- The elevation between the minimum water level at the source (an adjacent tank or reservoir or the well depth) and the centerline of the pump impeller. This can be positive if the pump is below the tank level or negative if the pump is above the tank level.
- The atmospheric pressure, which is 14.7 psi or 33.9 feet at sea level. This pressure decreases as the elevation increases, so knowing the elevation at the project site can be important.
- The minimum suction pressure of the pump, if applicable.
- The friction losses between the water source and the pump impeller. While bends, tees, valves and other components are usually considered to be “minor losses,” these losses can be important in calculation of NPSH.

Figure 12 shows the NPSH requirements for the pump. These are sometimes shown on the same graph as the pump curve and sometimes are shown separately. The orange circle highlights the NPSH requirement of 7 feet at the design flow. The purple circle highlights the NPSH requirement of 34 feet at pump runout. This indicates that not only are motor requirements potentially much higher near pump runout, but NPSH requirements are also potentially much higher. When comparing the NPSH available to the NPSH required, it is recommended that a factor of safety of at least 2 or 3 feet be provided to avoid the problems noted with air entering the pump bowl.

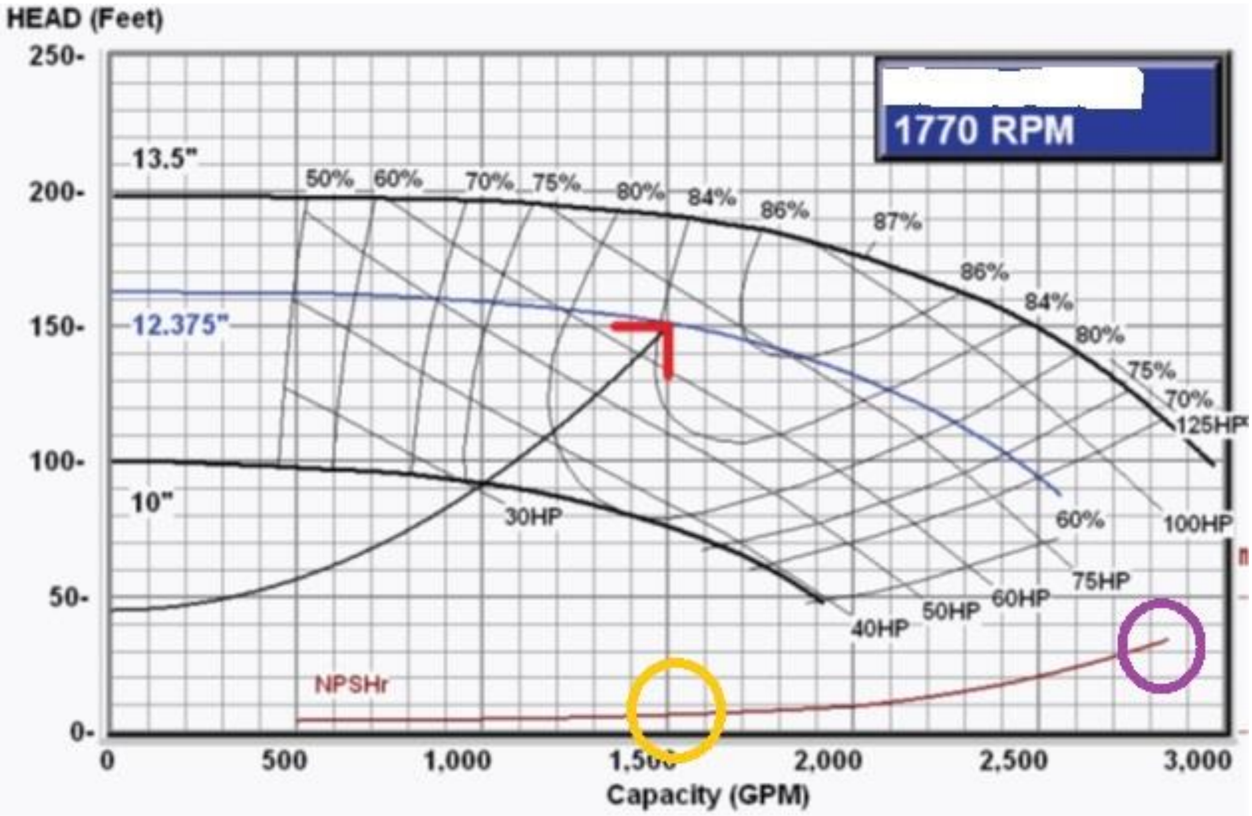

Figure 12 – Pump Curve with NPSH Requirements

System Curves

A system curve defines the relationship between flow and head in the distribution or transmission system. When used with a pump curve, the system curve can aid in identifying the operating point for the pump. A system curve is typically developed based on water flowing from the pump to a storage tank. The size and interconnection of the pipes between the pump and the tank are the primary factors. However, the system demand between the pump and the tank also influences the system curve. During periods of high demand, most (or all) of the water from the pump is used in the system and does not reach the tank, so the overall friction loss in the system is relatively low. During periods of low demand, most (or all) of the water from the pump reaches the tank, so there are higher flows and higher overall friction losses.

The simplest system is a pump connected to a reservoir on the suction side and to a single pipe on the discharge side that conveys water into a storage tank. In this system, the system curve is a function of:

- Size, length and friction characteristics of the pipe.
- Water level in the reservoir on the suction side.
- Water level in the storage tank on the discharge side.

A spreadsheet can be easily developed for the head loss calculation at a series of different flows, then the results plotted.

To illustrate an example of a system curve, the system includes:

- A reservoir on the suction side where the water level can vary from elevation 100 feet to elevation 120 feet.
- 5000 feet of 12-inch diameter PVC pipe, meeting the requirements of AWWA C900, DR 18. For this example a friction coefficient of 140 will be used.
- A storage tank at the end of the 12-inch diameter pipe where the water level can vary from elevation 220 feet to elevation 250 feet.

Figure 13 shows three different system curves for this example. The system curve in green, labeled minimum system curve, is the curve that results from the following conditions:

- Reservoir on the suction side at elevation 120.
- Storage tank at the end of the 12-inch diameter pipe at elevation 220.
- These two values yield a total static head difference of 100 feet, which is the head with zero flow, which represents no friction loss in the pipeline.

The system curve in red, labeled the average system curve, is the curve that results from the following conditions:

- Reservoir on the suction side at elevation 110.
- Storage tank at the end of the 12-inch diameter pipe at elevation 235.
- These two values yield a total static head difference of 125 feet.

The system curve in blue, labeled the maximum system curve, is the curve that results from the following conditions:

- Reservoir on the suction side at elevation 100.
- Storage tank at the end of the 12-inch diameter pipe at elevation 250.
- These two values yield a total static head difference of 150 feet.

These three curves illustrate that the system curve is not a single curve, but a series of curves. The other major parameter that is not included in the curves in Figure 13 is the impact of system demand. As previously noted, when system demand is high, friction loss is lower so the curves in Figure 13 will be flatter. The curves in Figure 13 represent a condition of zero demand between the pumps and the storage tank so friction loss is a maximum.

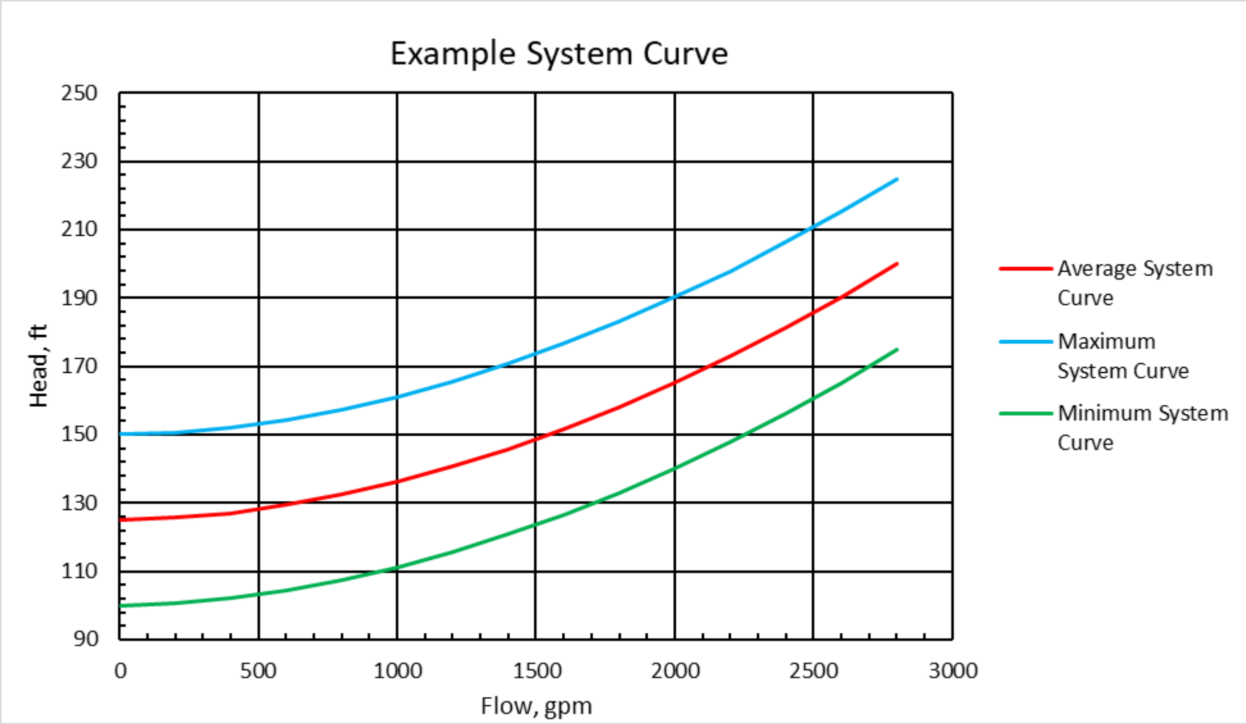

Figure 13 – Example System Curve

For a distribution system, it is more complicated to develop a system curve due to the large number of pipes and junctions. The system curve is unique for each pump location.

Some distribution system software packages provide the ability to create a system curve. However, the curve is created at a specific time interval, so it only presents the system curve for the head conditions at that time. Not only does the system curve change with differing head conditions, it also changes with varying demand conditions.

Minimum and maximum curves can be developed, but the variables need to include not only reservoir levels but also demand (such as maximum hour and minimum hour) to represent the full range of system curve possibilities.

Figure 14 shows the model for a small water system. The area circled in blue represents the water plant clearwell and high service pump. The area circled in green represents the elevated storage tank serving the community.

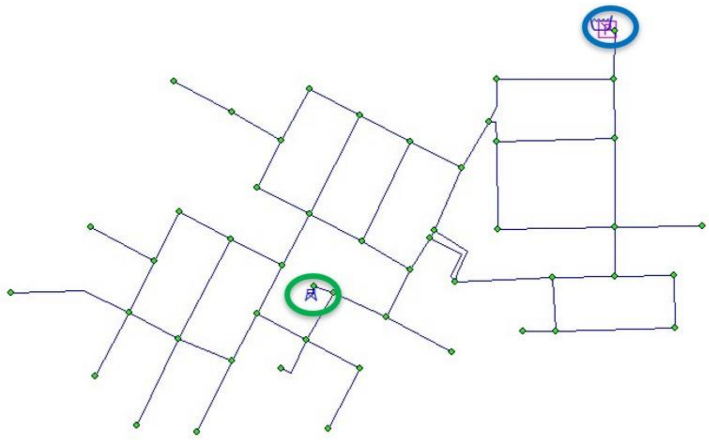

Figure 14 – System model

For the water system shown in Figure 14, the curves in Figure 15 represent the system curves for different demand conditions. The red curve represents the system curve during average demand conditions and the green curve represents the system curve during high demand conditions. Note while the curves have similar shapes, they are not parallel. The impact of friction loss on the system curve is higher during periods of high flows. During periods of low overall flows, the overall friction loss is lower, so the difference between the two values is lower.

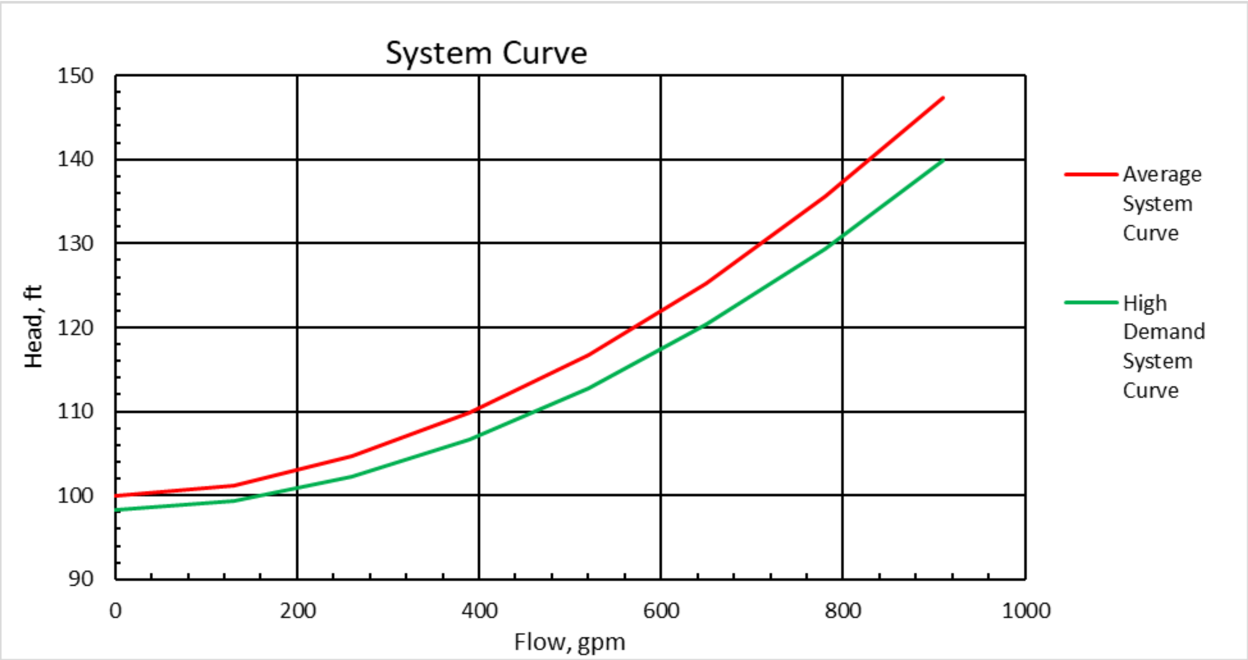

Figure 15 – System Curves

Combining System Curves and Pump Curves

Figure 16 shows the pump curve from Figure 12, with the 12.375-inch diameter impeller, superimposed on the system curves. Under the average system conditions, the pump will produce about 1500 gpm at 150 feet of head. However, under maximum system conditions, the pump will only produce about 900 gpm at 160 feet of head. With minimum system conditions, the pump will produce about 1900 gpm at 135 feet of head. This pump will often be considered to be a 1500 gpm pump, since that is probably the design condition, but it could produce anywhere from 900 gpm to 1900 gpm, depending on the levels in the reservoir and storage tank.

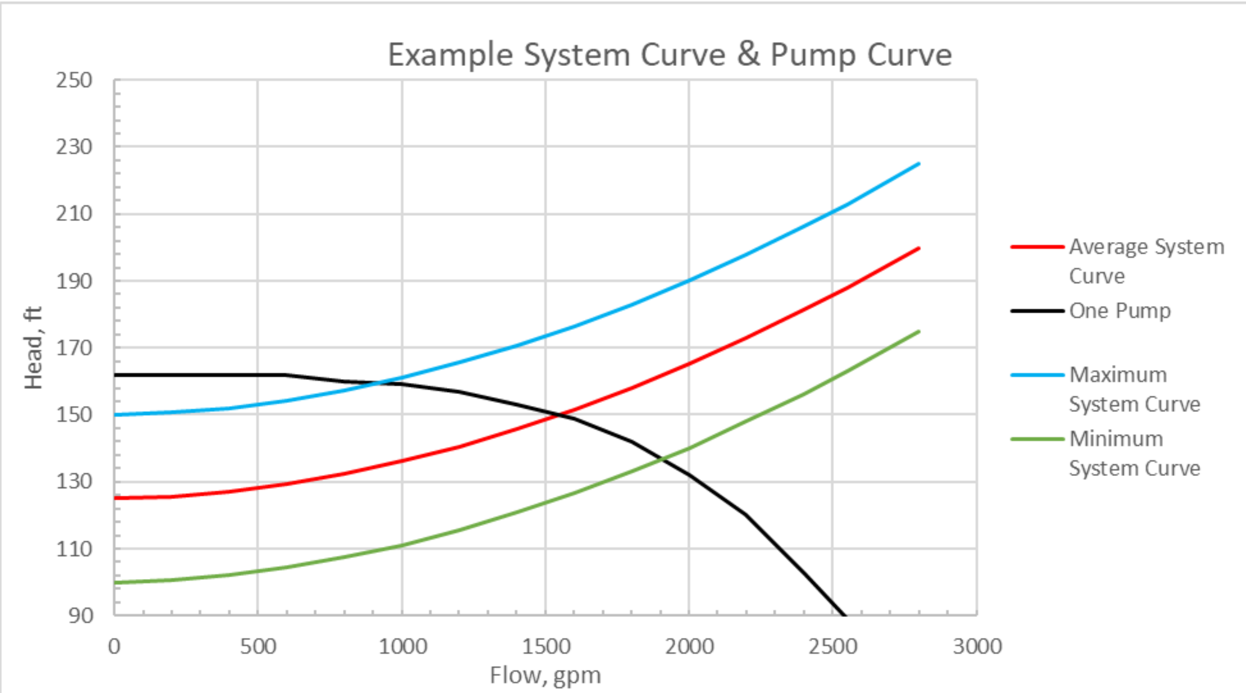

Figure 16 – Example System Curve with Pump Curve Superimposed

In many booster pump systems, more than one pump is installed. There is a common misconception that if one pump produces 1500 gpm, operating a second, identical pump will increase the flow from the pump station to 3000 gpm. To illustrate how to determine the impact of a second operating pump, Figure 17 shows the same curves as Figure 16, but includes the pump curve for a second pump.

The difference between operating one pump and operating two pumps is the flow rate on the pump curve is doubled at each head condition. For example, at 150 feet of head, one pump will produce 1500 gpm and two pumps will produce 3000 gpm. This is what typically convinces individuals that adding a second pump will double the flow rate. The problem with this assumption is it ignores the friction loss in the system associated with the higher flows. For the average system conditions, one pump produces 1500 gpm and two pumps will produce about 1800 gpm, an increase of only 20% with the second pump. For the minimum system curve, one pump produces 1900 gpm and two pumps will produce about 2400 gpm, an increase of about 26%. For the maximum curve, one pump produces about 900 gpm and two pumps will produce about 1000 gpm, an increase of only about 11%. The increase in flow with two pumps is a function of the pump curve and the system curve. This example is specific to this system and these pumps and is not representative of all situations. In some cases, operating a second pump will increase the flow substantially, but it will never double the flow unless the friction loss in the piping is essentially zero.

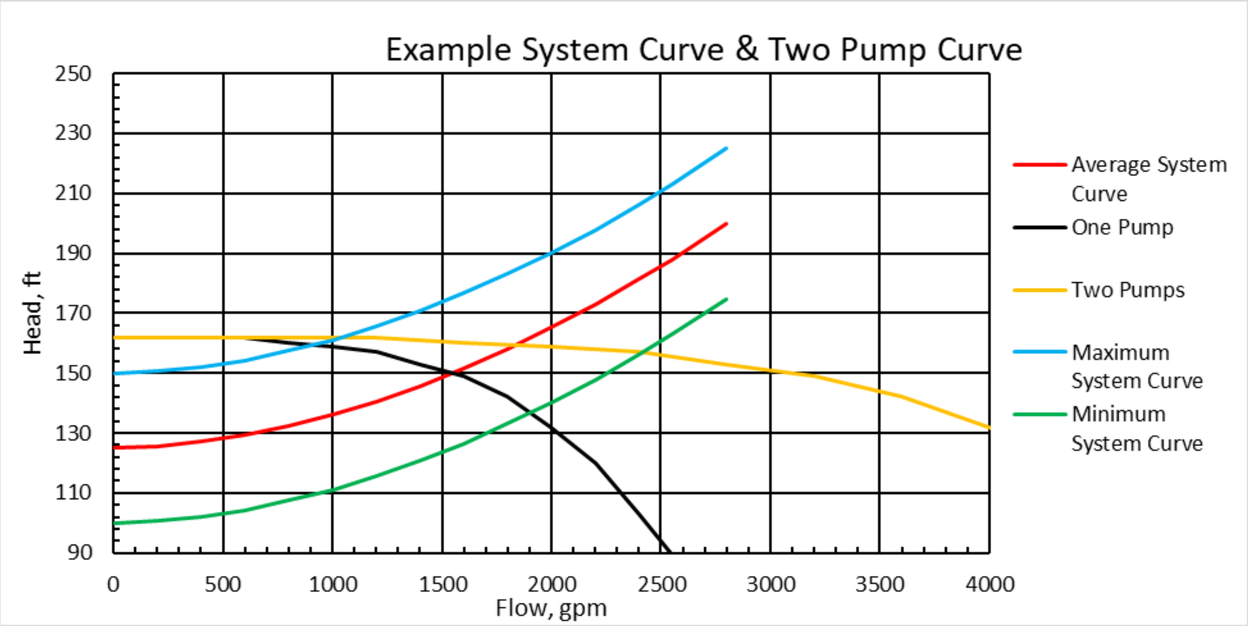

Figure 17 – Example System Curve with Two Pumps Operating

<u>Variable Frequency Drives</u>

In many instances, the motors for water pumps are equipped with a variable frequency drive (VFD). It is important if a VFD is to be used the motor be designed for use with a VFD. Not all motors are suitable for VFD use. A VFD changes the frequency of the power reaching the motor, from 60 Hz to something smaller. For example, setting the VFD at 90% changes the frequency to 54 Hz. This changes the speed of the motor in direct proportion. If the motor speed was 1800 rpm at 60 Hz, at 54 Hz, the motor speed will be 1800 * (54/60) = 1620 rpm.

The change is speed impacts the head and the flow from the pump. Flow is directly proportional to speed and head is directly proportional to the square of the speed. Power (HP) is directly proportional to the cube of the speed. If a pump can produce 1500 gpm at 150 ft of head at full speed, and the VFD is set at 54 Hz, it will produce 1500 * (54/60) = 1350 gpm at a head of 150 * $(54/60)^2$ = 121.5 ft.

There are a number of reasons to use a VFD, including:

- It allows for adjustment to the pump curve for periods of lower demands, such as:
  - Nighttime operation
  - Winter operation
  - Operation before a development is completed, so demands are lower than the ultimate design flow.

- It can reduce power consumption by more closely matching flows to demands.
- It provides significant protection against water hammer during pump starting and stopping. When using a VFD, the motor is started slowly and stopped slowly. This slow start and stop reduces the sudden change in velocity that can create significant water hammer in the system.
- It reduces demand charges associated with across-the-line motor starting. When a motor is started using the full-power, the in-rush current is commonly 5 to 6 times the current associated with operating the motor at full speed. While this in-rush current only occurs for a very short period of time, this large current creates a significant demand on the electrical system. Most pump stations are charged for electrical power in two separate ways. Similar to the way electricity is charged for a typical household, the electrical use (measured in kilowatt-hours, kwh) is measured and billed at a specific charge. There is also commonly a second charge on the electrical bill for a booster pump station, called a demand charge. This charge is based on the maximum instantaneous demand for the pump station. In most pump stations, this is a result of the starting of a pump, especially if one or more other pumps are also running. If multiple pumps are started at the same time, the demand charge is even higher. By using a VFD to start the motor slowly, the in-rush current can be limited to approximately the same value as the full-power operating current. In many cases, the cost of a VFD can be recovered in only a year or two just by reducing the high demand charge associated with across-the-line motor starting.

The use of a VFD will modify the pump curve in a very predictable manner. As noted, the flow will change as a direct proportion of the change in speed and the head will change in relation to the square of the change in speed. Figure 18 shows the resultant pump curves when the speed is changed to 95% of full speed and 90% of full speed. When the VFD is set at 95%, at 1500 gpm the available head from the pump drops from 150 feet to about 135 feet. When the VFD is set at 90%, the available head drops to 120 feet.

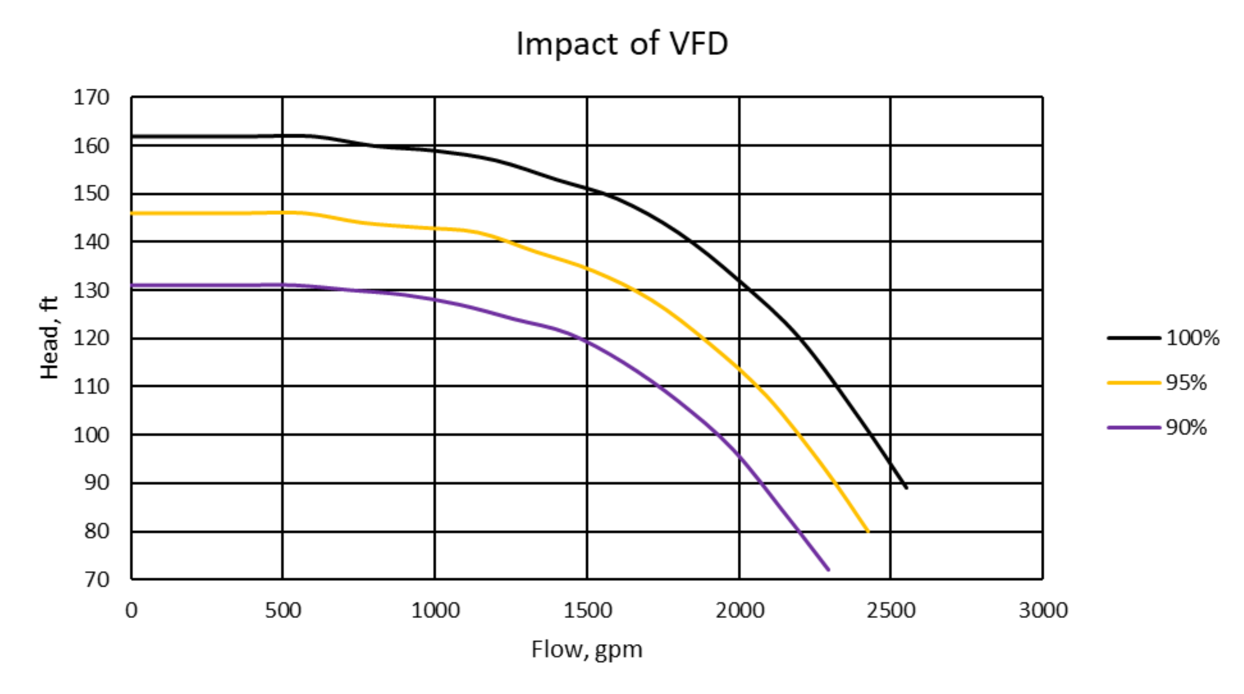

Figure 18 – Impact of VFD on Pump Curve

Figure 19 shows the various pump curves superimposed on the system curves from Figure 13. The design point for this system was 1500 gpm at 150 feet of head, based on the Average System Curve. If the VFD is set at 95% with the same system conditions, the pump would provide about 1200 gpm at 140 feet of head. If the VFD is set at 90%, the pump would provide about 650 gpm at 130 feet of head.

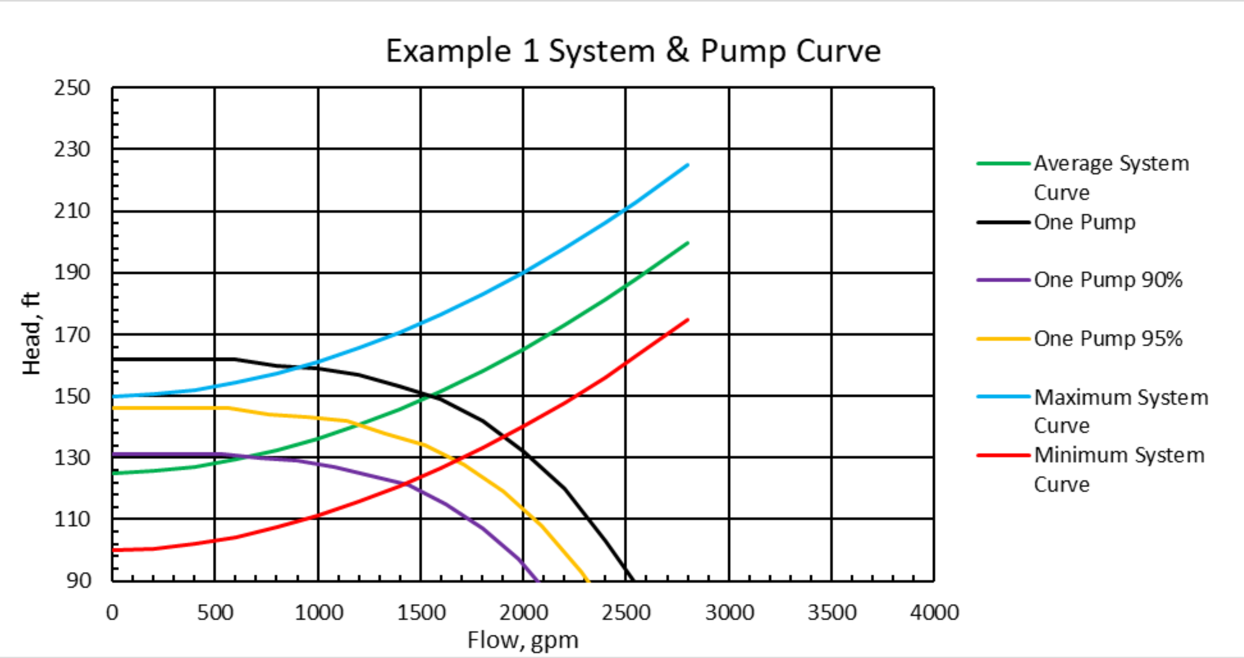

Figure 19 – System Curves and Pump Curves with VFD

Sometimes changes in other parts of a water system will result in needed changes to the pumps. The use of VFDs (either existing ones or new ones) can help a system to adjust to the new system curve resulting from these changes. Figure 20 shows the system previously shown in Figures 14 and 15, with a revised system curve due to construction of a new elevated tank in a different location. This new location reduced the friction losses from the pump to the tank. With the reduced friction losses, the system curve is now flatter. The existing pump, operating at 100% (the blue line in Figure 20) does not intersect the system curve within the operating range of the pump. Extrapolating the pump curve, it would intersect around a flow rate of 320 gpm. However, this is beyond the runout point on the pump curve and operating the pump consistently beyond the runout point is definitely not recommended. Using a VFD to reduce the motor speed to 80% (48 Hz), the pump would produce about 210 gpm at about 100 feet of head and stay within the range of the pump curve. This is one approach to addressing the revisions to the system without pump replacement.

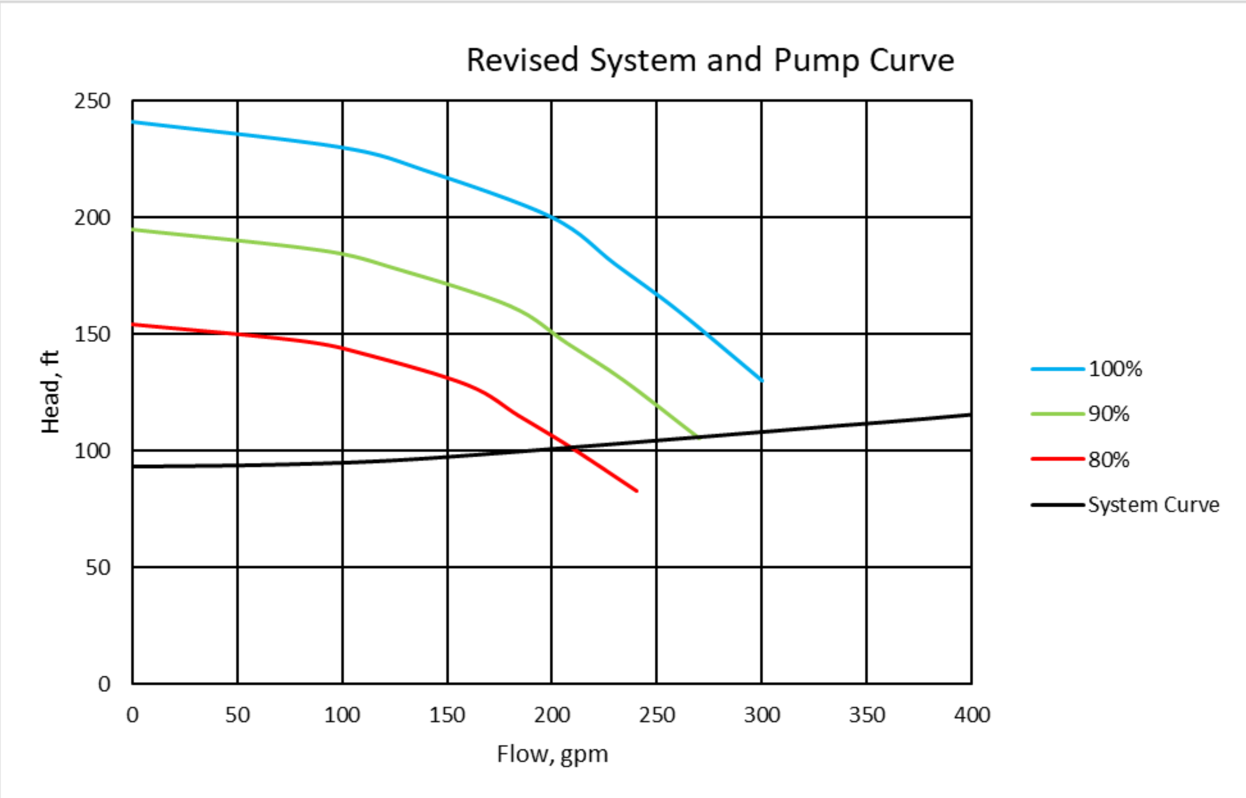

Figure 20 – Revised System and Pump Curves

Other Pump Controls

Besides the use of a VFD, there are several other pump controls that should be considered. They include:

- A check valve on the downstream side of each pump. The check valve prevents water in the discharge pipe from flow backwards through the pump.
- Surge control on the downstream side of the pump. This can take the form of a control valve specifically designed for surge control or a variety of other approaches to controlling surge pressures.
- A foot valve on the drop pipe for a centrifugal pump. A foot valve is a form of a check valve, but it is located on the drop pipe in a well.

Pump Selection Summary

The first part of pump selection is often finding a pump that fits the geometric conditions. Frequently the geometry of the pump station is pre-determined. Sometimes, it's not possible to find a particular pump style that will provide the necessary hydraulic demands. In that case, it's necessary to re-consider the pump station geometry.

The second part of the selection is defining the system curve. Here the pump curve information was presented first, in order to define the parameters, but the system curve is usually fixed by the existing or proposed system, with little flexibility (except perhaps pipe size for a proposed system).

The third step is to review different pump curves to determine which pumps will meet the desired design conditions. The need for a VFD should also be considered, both for early development/low flow conditions and for pump starting considerations.

Once design conditions have been established, pump suppliers can be very helpful in selecting a pump. However, not all suppliers are going to be able to provide an "optimum" pump, so coordination with multiple suppliers for each pump is important.

Water Pump Design - Quiz

Updated: 6/24/2023

1. Which of the following functions do pumps serve in a treatment facility?
    a. Move process water through the plant
    b. Move finished water into the distribution system
    c. Backwash filters
    d. All of the above

2. Where is the motor located in a submersible pump?
    a. At the top of the well
    b. At the bottom of the well
    c. Above the well screen
    d. Above the pump bowls

3. What is the angle of offset between the suction and discharge of an end suction centrifugal pump?
    a. 0 degrees
    b. 90 degrees
    c. 180 degrees
    d. The angle is variable

4. What units are typically used to express head of a pump?
    a. Feet
    b. Gallons per minute
    c. Horsepower
    d. Percentage

5. For the pump in Figure 8, if the design point is 2300 gpm at 160 feet of head, what size motor is required?
    a. 60 HP
    b. 75 HP
    c. 100 HP
    d. 125 HP

6. For the pump curve in Figure 10, with a 12.375-inch impeller, what is the flow at runout?
    a. 0 gpm
    b. 1500 gpm
    c. 2000 gpm
    d. 2550 gpm

7. For the pump curve in Figure 10, with a 10-inch impeller, what is the required motor horsepower?
    a. 30 HP
    b. 40 HP
    c. 50 HP
    d. 60 HP

8. Which of the following factors impact the NPSH required?
    a. Elevation difference between suction reservoir and discharge tank.
    b. Minimum suction pressure of the pump
    c. Friction losses between the water source and pump impeller
    d. All of the above

9. Which of the following conditions will result in the highest head conditions on the system curve?
    a. Low reservoir level at the pump and high storage tank level
    b. High reservoir level at the pump and low storage tank level
    c. Low reservoir level at the pump and low storage tank level
    d. High reservoir level at the pump and high storage tank level

10. In Figure 16, what is the estimated flow when the maximum system curve is appropriate?
    a. 900 gpm
    b. 1500 gpm
    c. 1900 gpm
    d. 2500 gpm