A SunCam online continuing education course

Centrifugal Pump Selection

by

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Course Outline:

What is a Centrifugal Pump?
Common Applications
Pump Design Steps
Design Criteria
Design Flow Rates
Number of Pumps and Speed Control
Process Flow Diagram
Intake Design and NPSH
Discharge Design
System Curves
Pump Curves and Selection
Hydraulic Profile
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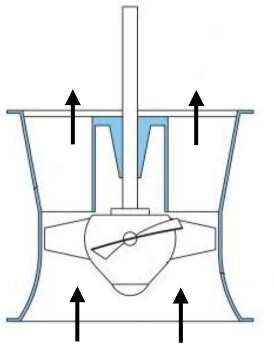
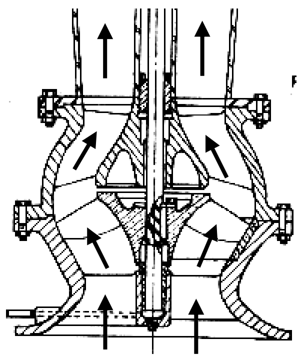
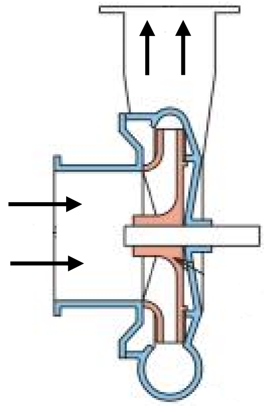
Examination

What is a Centrifugal Pump?

Pumps are mechanical devices that move fluids. Centrifugal type pumps use spinning impellers as the means to move fluids. The fluid enters the pump near the center of the impeller and is accelerated outward (radially) to a discharge outlet. The flow turns approximately 90 degrees through the pump. For this reason, centrifugal pumps are often called radial flow pumps.

See Figure 1 and Table 1 for a comparison of the three types of rotodynamic pumps: axial flow, mixed flow, and radial flow.

Figure 1: Example impellers for the three types of rotodynamic pumps.
Source: https://commons.wikimedia.org/wiki/File:Pump_Impellers-1.jpg CC BY-SA 3.0

Table 1: Comparison of Rotodynamic Pumps			
Pump Type	Section View	Flow Change	Performance
Axial Flow		0^o	High Flow Rate Low Pressure
Mixed Flow		40^o to 60^o	Medium Flow Rate Medium Pressure
Radial Flow (Centrifugal)		90^o	Low Flow Rate High Pressure
Sources:	https://commons.wikimedia.org/wiki/File:Axial_flow_pump-diagram.jpg (modified) by Jonasz, CC BY-SA 3.0 https://commons.wikimedia.org/wiki/File:Centrifugal_pump-tech_diagram (modified) by Jonasz, CC BY-SA 3.0		

Common parts of a centrifugal pump are labeled in Figures 2 and 3.

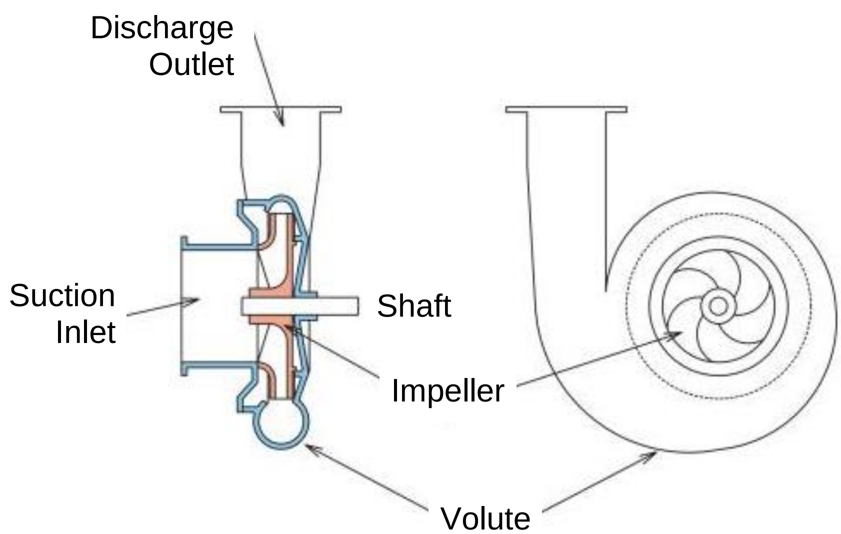


Figure 2: Major components of a typical centrifugal pump.
https://commons.wikimedia.org/wiki/File:Centrifugal_pump-tech_diagram (modified) by Jonasz, CC BY-SA 3.0

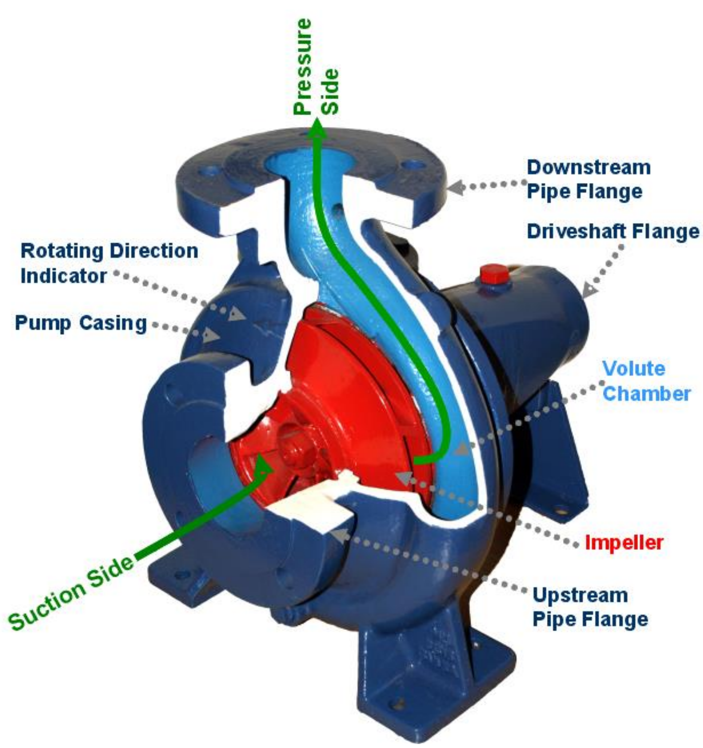


Figure 3: Labels for an end-suction centrifugal pump with an enclosed impeller.
A coupling and motor are attached to the driveshaft to turn the shaft and impeller.

The end-suction type pump is the most common type of centrifugal pump. Several other types of centrifugal pumps are shown in Figures 4 through 8.

Figure 4: A horizontal split-case centrifugal pump for finished water pumping. Flow enters on the left and splits to either side of the impeller. The impeller receives flow in the center from both sides and discharges to the right.

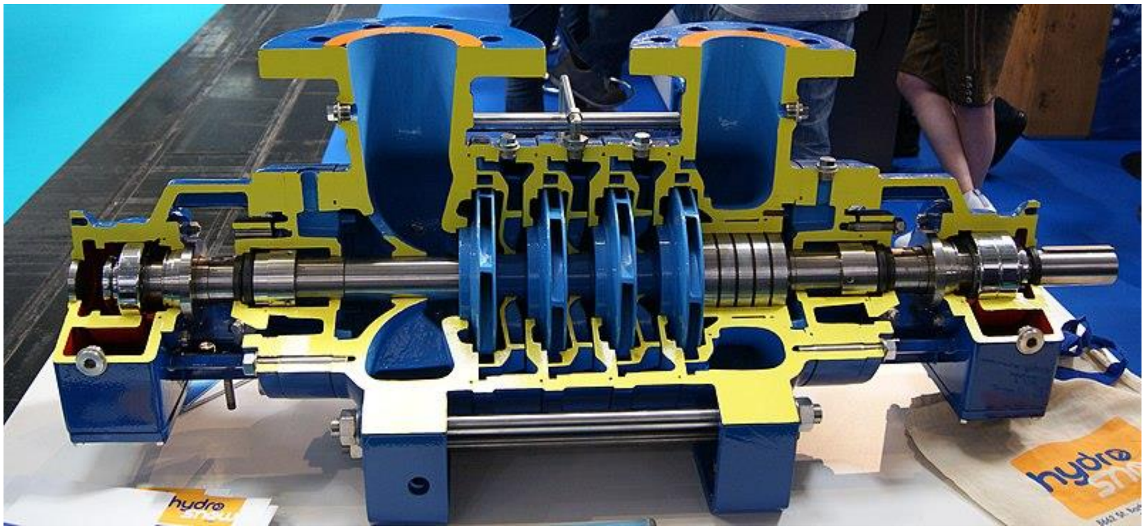


Figure 5: A multi-stage centrifugal pump with four impellers.
Flow enters on the upper left and discharges on the upper right.
https://commons.wikimedia.org/wiki/File:Innsbruck-Hydrosnow-model_pump by Asurnipal, CC BY-SA 4.0

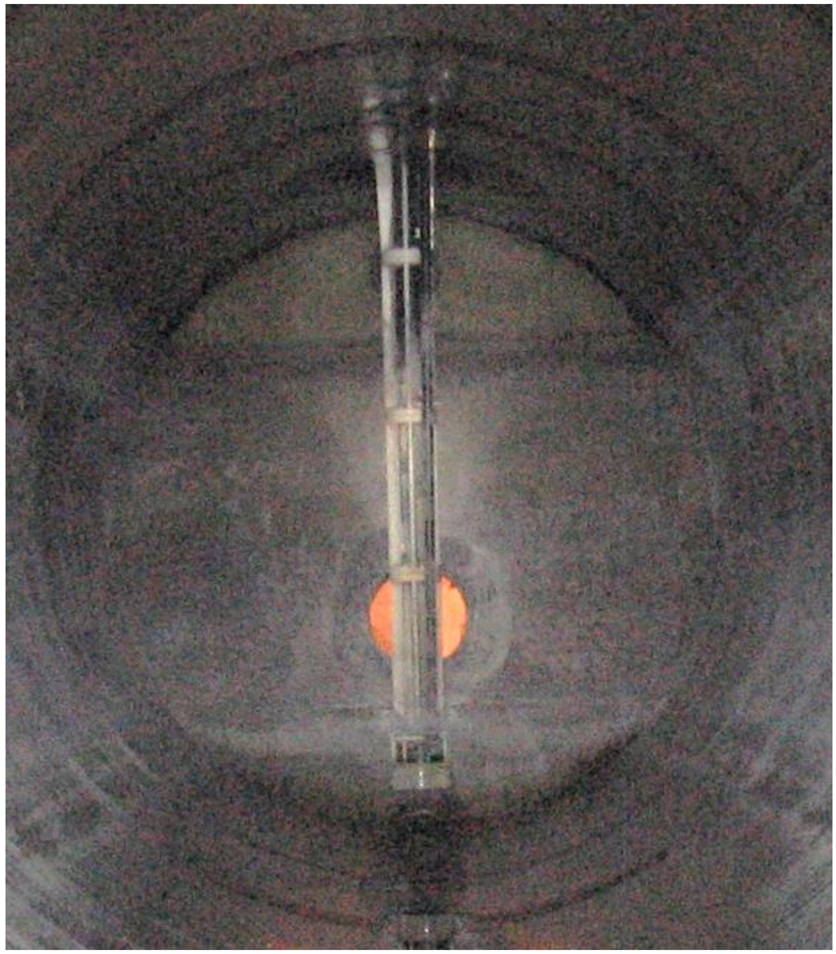


Figure 6: A vertical suspended column centrifugal pump for chemical pumping. The motor is mounted safely above the tank. The impeller is submerged near the bottom of the tank. A small discharge pipe rises to the left of the enclosed vertical shaft.

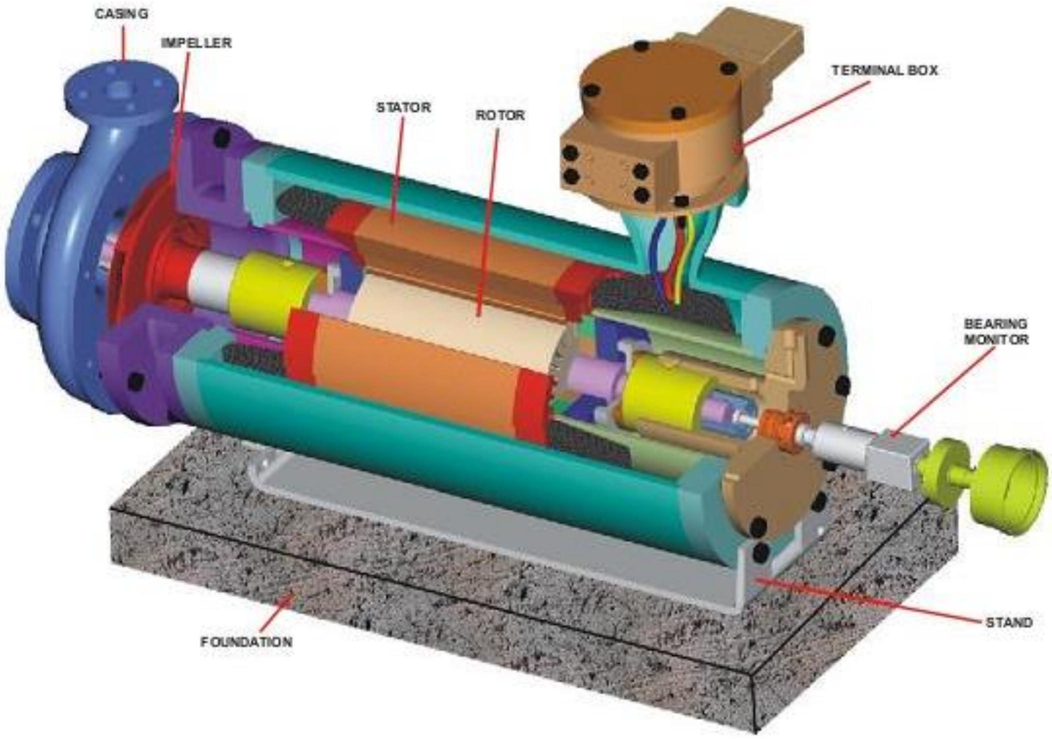
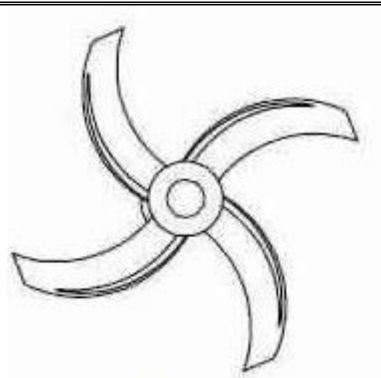
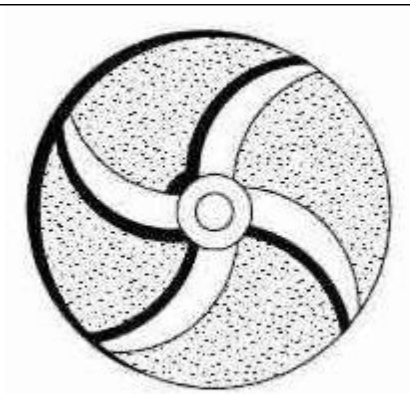
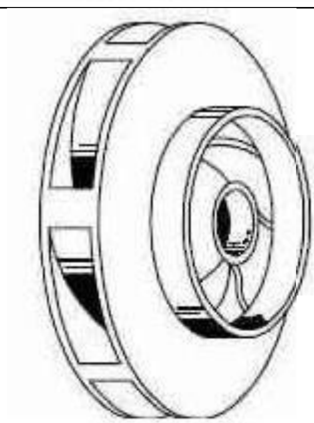


Figure 7: A canned centrifugal pump, which is a sealless pump.
https://commons.wikimedia.org/wiki/File:Canned_motor_pump.jpg by Mukund275, CC BY-SA 3.0

Figure 8: Examples of submersible centrifugal pumps.
These types of pumps are common for wastewater wet wells and sumps.
https://commons.wikimedia.org/wiki/File:NS_non_clog_submersible_pumps.jpg by Mukund275, CC BY-SA 3.0
https://commons.wikimedia.org/wiki/File:Tauchpumpe_Abwasser.jpg by Dergreg, CC 1.0

There are three different types of impeller constructions, as summarized in Table 2. The physical difference is the presence or absence of a backing plate, called a shroud. The closed impeller has a shroud on both sides of the impeller.

Table 2: Comparison of Impeller Types

Type	Example	Flow Capacity	Efficiency	Solids Handling	Ease of Cleaning
Open		Low	Low	Very Good (exept stringy material)	Very Good
Semi-open or Semi-enclosed		Avg	Avg	Good	Good
Closed or Enclosed		High	High	Poor	Poor

Source: Farokhzad, Motlagh, Farhadi (2013) Modal Analysis of a Semi-Closed Impeller of Centrifugal Water Pump, 2. 205-213.

Common Applications

Centrifugal pumps are the most common type of pump being used today because they offer the following benefits:

- Design simplicity
- Wide range of capacity
- High efficiency
- Consistent flow rate
- Ease of operation and maintenance

Centrifugal pumps are commonly used in many industries around the world, including the following:

- Chemical
- Construction
- Food & Beverage
- Manufacturing
- Mining & Minerals
- Oil & Gas
- Power Generation
- Pulp & Paper
- Water & Wastewater

Many types of engineers regularly encounter centrifugal pumps, including:

- Chemical Engineers
- Civil Engineers
- Environmental Engineers
- Industrial Engineers
- Marine Engineers
- Mechanical Engineers
- Mining Engineers
- Nuclear Engineers
- Petroleum Engineers

Engineers are expected to select a pump that is appropriate for the application and of the correct size (pump model and impeller diameter) to handle the design conditions. This course will help prepare you for these tasks.

The Hydraulic Institute (HI) Standards are the most commonly accepted guidelines and specifications for the design of pumping systems. Centrifugal pumps are covered in the series entitled “Rotodynamic Pumps”.

Wastewater Pumps

Pumps handling raw wastewater shall be capable of passing solid spheres of at least 3 inches (80 mm) in diameter, per the *Ten States Standards*. And pump suction and discharge openings shall be at least 4 inches (100 mm) in diameter. An exception to these requirements is if a grinder pump is chosen or if a macerator/grinder is installed upstream of the pumps.

Increasingly in recent years, pumps that can pass 3-inch solids are still becoming clogged due to rags and wipes being flushed into the sewer. The following pumps have shown good performance when rags are present in the wastewater:

- Chopper pumps or cutter pumps with cutting bars adjacent to the vane edges.
- Screw impeller pumps that can pass rags and achieve high efficiency.
- An adaptive self-cleaning impeller that can move axially (i.e. Flygt N-pump).

Pump Design Steps

The design of a pumping system can be accomplished in the following steps:

1. Define design criteria
2. Choose the number of pumps and speed control
3. Create a process flow diagram
4. Intake design
5. Discharge design
6. System curves
7. Pump curves
8. Pump selection
9. Create a hydraulic profile
10. Quality review of calculations
11. Design of ancillary features

The order of these design steps can be modified. Pump design requires an iterative approach. For example, the number of pumps and pipe sizes are assumed and then checked and modified based on the final pump selection. Any change in pipe size or arrangement requires modifying the system curve, which impacts the final pump selection. These inter-dependencies increase the chance for oversights and mistakes and make the final quality review of high importance. Calculations should be kept well organized to assist in this effort.

The following sections address each of these design steps. Additional guidance can be found in the reference documents in the Helpful References section.

Design Criteria

Defining the design criteria is the first step in ensuring a successful pump selection. Design criteria are specific goals. The following are example design criteria to consider when designing a pumping system:

1. Flow and pressure capacity match demands;
2. Pump type and materials suitable for fluid type;
3. Pass solid sphere of specified diameter;
4. Avoid clogging or ragging;
5. Starts per hour not excessive;
6. Compatible with selected speed control;
7. Avoid NPSH and cavitation problems;
8. Allow future change to a larger or smaller impeller;
9. Minimize energy consumption;
10. Minimize capital costs and lifecycle costs;
11. Allow proper maintenance access and clearance for pump removal;
12. Provide an installed redundant pump;
13. Choose a pump with readily available parts; and
14. Provide common spare parts and/or a spare pump on the shelf.

It is recommended to gain stakeholder input to ensure important goals are not missed. Stakeholders may include staff from management, operations, maintenance, and consultants. Although the design criteria should be defined at the start of the design process, it is important to review the criteria throughout the design process to confirm nothing is forgotten and to avoid redesign.

Design Flow Rates

It is critical to define the flow rates for pump selection. The required flow rates are often called the flow demands or the design flow rates. When combined with pressure requirements, these are called the system demands or design conditions.

Each pumping system has a unique combination of flow sources and discharge requirements that should be identified and reviewed when defining the design flow rates. The design flow rates should be defined for the overall pumping system, regardless of the number of pumps. After deciding the number of pumps and the piping arrangement, the flow rates per pump can be specified.

Common flow rates to define are as follows:

- Minimum design flow (MDF), or minimum hourly flow (MHF): This is the smallest flow rate expected to be maintained by the pumping system.
- Average design flow (ADF), or average daily flow (ADF): This is the average flow calculated as the volume of fluid divided by the time period (such as the number of days or months).
- Maximum design flow (MDF), or maximum day design flow (MDDF): This is the largest of the various calculated or measured flow rates, typically measured over days or months.
- Peak design flow (PDF), peak hourly flow (PHF), or instantaneous peak flow (IPF): This is the highest flow rate (measured in a short interval) to be maintained by the pumping system. Often this value is estimated by multiplying the average design flow by a peak factor. For example, in a wastewater lift station, a peak factor of 2 to 4 is commonly used.
- Ultimate design flow (UDF), ultimate average flow (UAF), or ultimate peak flow (UPF): This is the estimated flow rate to be experienced in the future, taking into account predicted changes or growth in the system or flow sources. Often the pumping system is designed with the flexibility to meet the ultimate design flows. For example, impellers or pumps may be upgraded or an additional pump may be installed.

Pumping systems are typically designed so that the “firm capacity” meets or exceeds the PHF. The firm capacity is the discharge flow rate with all the pumps running except one of the largest pumps. This requires the pumping system to be designed with an installed spare large pump.

Note that the pressure requirements are typically calculated later in the design process when developing the system curve. However, in some cases, pressure requirements are part of the initial design criteria. For example, a finished water pumping system is to maintain a pressure of 80 psi in the drinking water distribution system at all times. In this example, a design criteria would be a discharge pressure of 80 psi at all flow rates.

Field Measurements

When designing for the rehabilitation or replacement of an existing pumping system, it is helpful to measure the actual flow rates to define the ADF and PHF. This should be done over a long period to help ensure the extreme flow events are captured. For example, public water and wastewater systems experience seasonal flow variations, so a year or more of flow rates should be analyzed.

The following are common methods for using field measurements to define flow rates:

1. Install a flow meter on the suction or discharge piping.
2. Use level sensor readings in tanks or wet wells in combination with the pump on and off status. Level readings show the rise rate when pumps are off and fall rate when pumps are on, and thereby allow calculation of the volume discharged over each pumping cycle. The volume divided by the cycle time is the flow rate.
3. Use fluid source data. For example, for a wastewater lift station, use water consumption data by summing the water meter totals for each property discharging to the lift station. This approach assumes the wastewater flow is equal to or less than the water utilized, which is nearly always the case for residential and commercial properties. A peak factor can estimate inflow and infiltration contributions.

4. Use pump run times and known pump flow rates. The pump flow rate can be confirmed by either a draw-down test (measuring the change in level in the tank or wet well) or by checking the discharge pressure and finding the corresponding operating point on the pump curve. See Figure 9 for an example. Note to subtract any elevation difference or significant head loss between the pump and gauge. The ADF equals the pump run time (total number of minutes the various pumps were "on") multiplied by the pump flow rate (from the pump curve) divided by the measuring period.

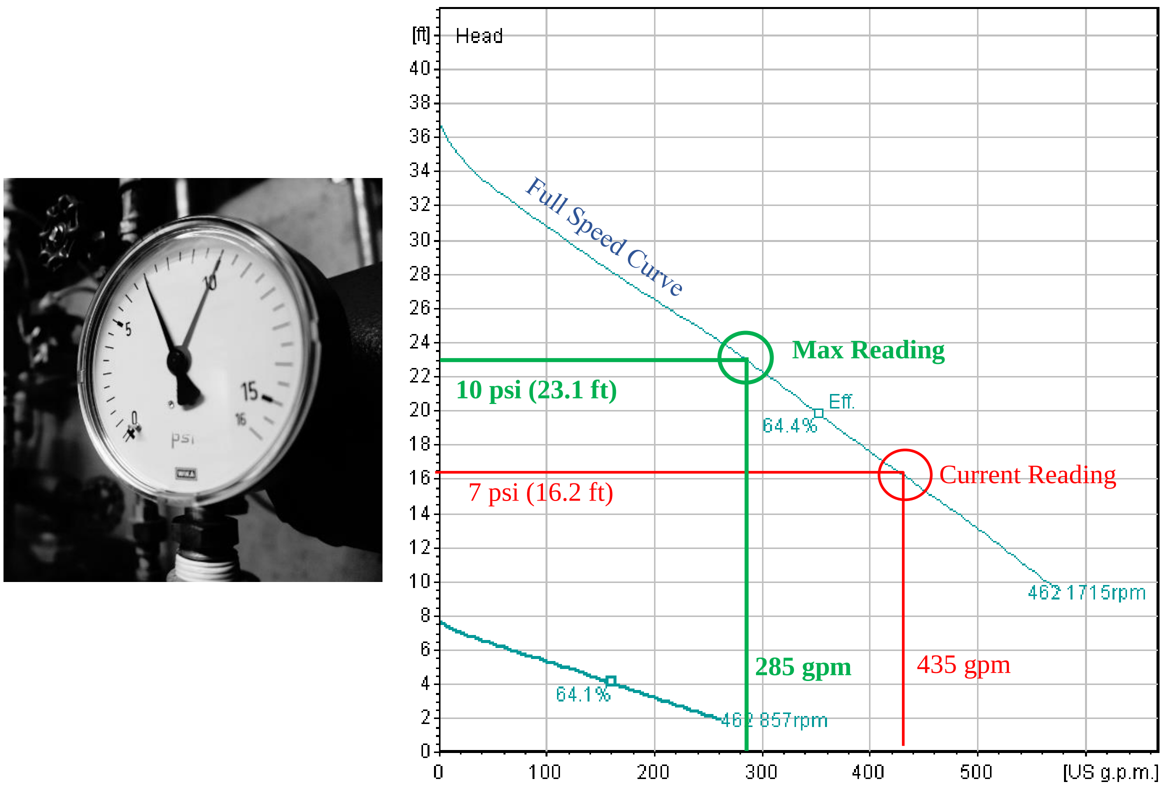


Figure 9: Left: A discharge pressure reading of 7 psi (16.2 ft) and a max pointer reading of 10 psi (23.1 ft). Right: Pump curve with corresponding operating points. These pressure readings also help define the discharge pressure range, which can be used for developing (or checking) the system curve.

Example 1:
Engineer Laura is asked to list the design criteria for upgrading a wastewater booster pump station. The station must reliably pump a peak hour flow (PHF) of 3,600 gpm, and average flow (ADF) of 900 gpm, and a minimum hour flow (MHF) of 400 gpm. Hydraulic modeling indicates that the discharge pressure is 40 psi at the PHF, 20 psi at the ADF, and 15 psi at the MHF. Two end-suction type pumps with variable frequency drives (VFDs) are to be utilized to achieve this range.

Solution:
Laura creates the following Table 3 to summarize the design criteria. Note that 1 psi equals 2.31 feet of head.

Table 3: Booster Pump Station Design Criteria

Fluid Type	Raw wastewater	
Pump Type	End-suction centrifugal	
Number of Pumps	3 (2 duty + 1 standby)	
Drive Type	Variable frequency	
Flow Conditions	Flow Rate (gpm)	Discharge Pressure
Peak Hour (PHF)	3,600	40 psi (92.3 ft)
Average	900	20 psi (46.2 ft)
Minimum Hour (MHF)	400	15 psi (34.7 ft)

Number of Pumps and Speed Control

Early in the design process, it is helpful to choose the number of pumps and the type of speed control. The assumed number of pumps can be used to performing an initial pump selection, which can be revisited and modified to confirm the ideal number.

A duplex pump arrangement is the simplest design. There is one duty (or lead) pump and one standby (or lag) pump. Each pump is the same and each pump can operate at the PDF.

A duplex pump arrangement has the following advantages:

- Simplicity in design, construction, and maintenance,
- Lowest construction cost, and
- Smallest footprint pump station.

Duplex arrangements are common for wet well or sump pump applications for which a pump will cycle on and off based on floats or level switches. In these applications, the pump is not required to match a broad range of influent flow conditions. This is in contrast to an in-line booster pump station application which requires operating over a broad range of flows and pressures. A booster pump station typically requires multiple pumps and variable speed drives to be able to operate at the MDF and the PDF.

Using three or more pumps is generally beneficial under the following conditions:

- In-line booster or repump applications,
- Large flow rates, such as a PHF greater than 5,000 gpm,
- Peak factor great than 4 (such as for combined sewer systems), and a
- Large pressure demand range, such as greater than 20 psi.

Three or more pumps offers the following benefits:

- Ability to maintain the water level in wet well or tank applications,
- Ability to cover a greater range of flows and pressures,
- Can pump at average flow if two pumps are out of service, and
- Increase in pumping efficiency with associated energy savings.

Large pumping stations, and especially booster stations, may have a combination of small pumps and large pumps. The small pumps stay in the high-efficiency range during low flow events, while the larger pumps can efficiently handle flows during large events. This combination of small and large pumps requires a more complex control logic. Also, a redundant small and redundant large pump may be required.

<u>Speed Control</u>
A decision needs to be made if the pumps will be constant speed or variable speed controlled. Constant speed pumps mean the pump will output a relatively constant discharge regardless of the influent flow. This is often acceptable for wet well, sump, or tank applications, for which the water level will fall when the pump is on and rise when the pump is off. The pump on and pump off levels need to be far enough apart to allow proper pump cycling. Often the storage volume needs to be larger than a variable speed control scenario.

Variable speed controls are used to maintain flow, pressure, or a fixed water level. This is achieved by adjusting the speed of the pump in small increments based on instrument readings. The following are benefits to variable speed control:

- For large flow applications, variable speed pumping may allow a given flow range to be achieved with fewer pumps than a constant speed alternative.
- Variable speed pumping is often used to optimize pump performance and minimize power use.
- Variable speed pumping can reduce the storage volume.

Several types of variable speed pumping equipment are available, including variable voltage and frequency drives, eddy current couplings, and mechanical variable speed drives. The most common is the variable frequency drive (VFD). This equipment adds a small amount of energy loss, typically 3% to 5%.

Constant speed controls offer the following benefits:

- Simplicity in maintenance and operation,
- Lower construction costs, and
- Controls have a longer life compared with VFD equipment, especially when installed outdoors.

Process Flow Diagram

Early in the design process, it is important to make a schematic drawing of the overall pumping system. This may start as a back-of-the-envelope sketch with boxes and lines. As the design develops, a more formal diagram should be developed and drawn in CAD. A process flow diagram (PFD) is a simple schematic showing major components such as pumps and tanks, and lines representing the piping. This schematic helps to define the piping arrangement and eventually create a system curve for pump selection.

See Figures 10, 11, and 12 for examples of PFDs. Note that these examples have valve and instrument details that would be developed later in the design.

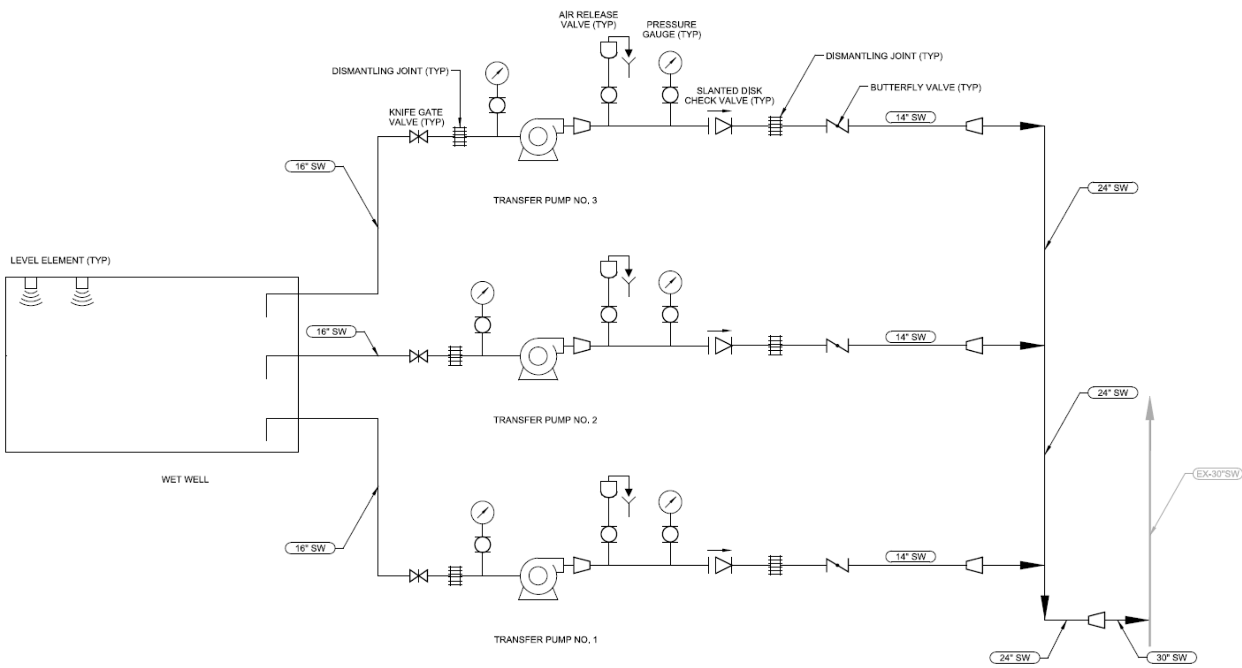


Figure 10: Example PFD of a pump station with three centrifugal pumps drawing from a wet well and discharging into an existing 30” SW pipe. Here, SW means “settled water” as the fluid type.

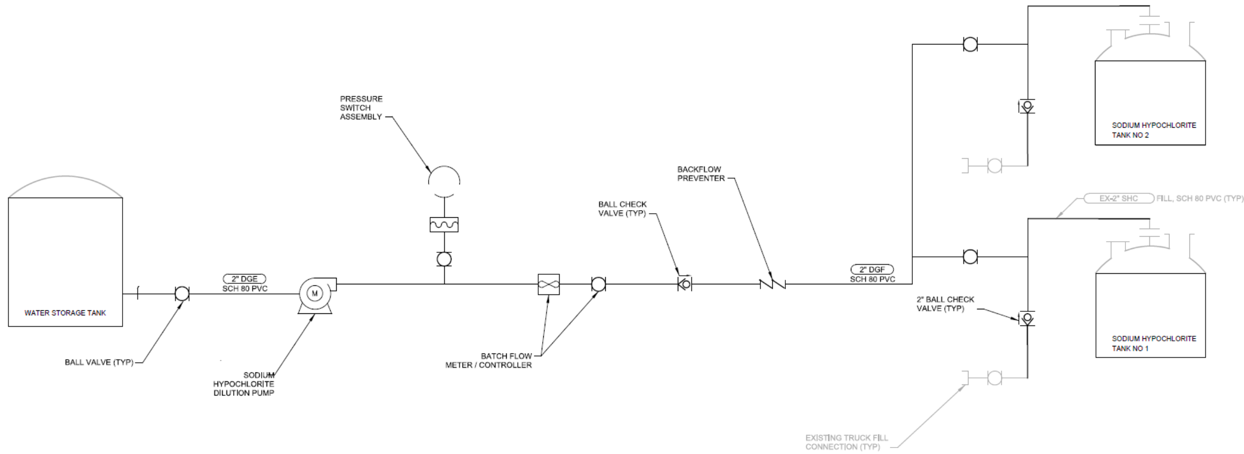


Figure 11: Example PFD of a chemical dilution system with a single pump.

Process flow diagrams are often given to electrical and controls engineers to create instrumentation and controls diagrams (P&IDs). See Figure 12 for an example.

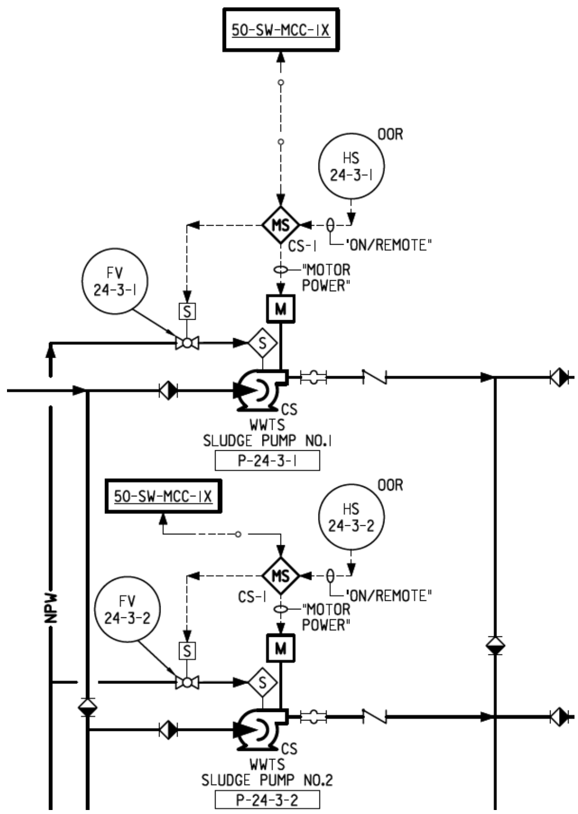


Figure 12: Example P&ID with symbols indicating the controls design.

Intake Design and NPSH

Perhaps the most important part of a pumping system is the intake design. A proper intake design protects the pump from vortices, entrained air, and cavitation. The intake design concerns the suction piping and any inlets that guide flow into each pump. In the cases of submersible pumps and vertical pumps, the inlet is the opening at the bottom of the pump. In the case of dry well pumps, the inlet is the opening or bell in the wet well or tank. In the case of in-line booster pumps, the intake is the tee or wye from the manifold pipe.

The pump manufacturer should provide recommendations for the intake design, although the following principles should still be reviewed during design. Important intake dimensions are as follows, per *HI Standard 9.8* on intake design:

- Bell diameter sized so velocity at maximum pump flow is approximately 5.5 ft/s (1.7 m/s).
- Clearance between multiple inlet bells (or pump volutes) to be a minimum of 0.25 times the bell diameter. For submersible pumps, confirm the minimum pump spacing and wall distance with the manufacturer.
- Distance from the center of the inlet bell to the nearest wall of the wet well should be a minimum of 0.75 times the bell diameter.
- Distance from the inlet bell to the flat floor should be from 0.3 to 0.5 times the bell diameter.
- Minimum inlet submergence (S), which is the depth in inches below the minimum water surface, based on bell diameter (D) in inches and maximum flow rate (Q_{max}) in gpm:

$$S = D + \frac{0.574 * Q_{max}}{D^{1.5}}$$

For wet well and dry well pumps, water levels should be defined by the engineer to ensure pumps are protected from entrained air.

Example 2:
Engineer Henry is designing a duplex wet well lift station. He needs to confirm the depth of the wet well using the pipe invert elevation, pump on and off levels, and provide a 3 foot inlet submergence.

Solution:
Henry creates Figure 13 with critical elevations defined. A total wet well depth of 20'-0" is required.

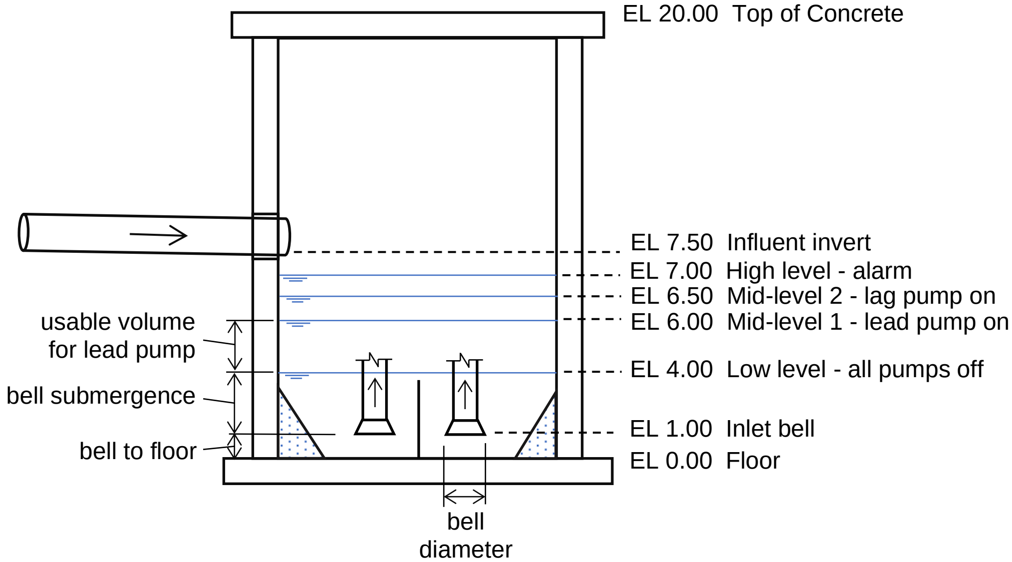


Figure 13: Wet well elevation with important elevations defined for Example 2.

<u>NPSH</u>

To avoid cavitation, the net positive suction head available (NPSHa) should be greater than the net positive suction head required (NPSHr). The NPSHa is based on the details of the intake design, while the NPSHr is from the pump manufacturer.

Often the pump curve will include an NPSHr curve. The engineer should confirm the NPSHr value at the maximum pump operating flow rate is less than the calculated NPSHa. If the NPSHr is greater than NPSHa, the following options are available to correct the issue:

- Increase inlet submergence,
- Increase suction pipe size,
- Use long radius elbows on the suction piping, or
- Choose a different pump.

The NPSHa formula is as follows, with definitions and an example in Table 4:

$$NPSHa = H_{bar} + h_s - h_{vap} - h_{fs} - h_m - h_{vol} - h_a - FS$$

Table 4: NPSHa Definitions and Calculation

Term	**Example (ft)**	**Definition**
H_{bar}	+33.96	Atmospheric pressure, which is 14.7 psi (33.96 ft) at sea level.
h_s	+2.50	Minimum static head at pump. From pump impeller to low water level for wet well or tank applications.
H_{vap}	-1	Vapor pressure of water, at 75 deg F, expressed in feet.
h_{fs}	-0.50	Suction pipe friction losses at the max pump operating flow rate (intersection of the pump curve and low head system curve).
Σh_m	-1.96	Suction pipe minor losses at max pump operating flow rate.
h_{vol}	-2	Partial pressure of dissolved gases such as air in water (customarily ignored as insignificant) and organics in wastewater (estimated at 2 ft).
h_a	-0	Acceleration head for positive displacement pumps only. See Section 1 of *Cameron Hydraulic Data* for the formula.
FS	-5	Factor of Safety, which can range from 2ft to 5ft, or 20% to 35% of NPSHr.
NPSHa	**26.0**	**Sum the above terms**

Note that checking the NPSH is different than checking the minimum inlet submergence. Both of these calculations should be done for each size pump.

Discharge Design

The discharge design consists of the following:

- Sizing the discharge piping,
- Selecting a material, thickness or class, and lining for the piping,
- Selecting the type of check valve and isolation valve,
- Determining the route of the piping, including all fittings (can estimate fittings during preliminary design and confirm during final design), and
- Defining the discharge elevation, water level, or tie-in pressure.

The main tool for sizing the discharge piping is the velocity at the maximum pump rate. The following are *Ten States Standards* recommendations for sizing discharge and force main pipes for wastewater applications:

Minimum velocity: 2 fps, to prevent solids buildup and clean the pipe

Maximum velocity: 8 fps, to avoid high head loss and prevent damage to valves and pipe lining.

If there are more than two pumps, the branches to each pump will be smaller than the pipe downstream of the tee connections. Each pump branch should be sized based on the maximum flow and velocity from the corresponding pump. And downstream of the point the branches combine, the pipe should be designed based on the overall firm capacity of the pumping system, or the UDF, whichever is greater.

See Figure 14 for an example of small pipes at each pump branch with a common larger discharge pipe.

Figure 14: Three horizontal split-case pumps drawing from a suction manifold on the right and sending flow into a discharge manifold on the left.

System Curves

Creating a system curve is the most technically difficult part of the pump selection process. An engineer should perform system curve calculations for each application. A system curve is a plot of the total dynamic head (Y-axis) versus the flow rate (X-axis).

Ideally, the “system” starts with the static liquid elevation at the suction side to the static liquid elevation at the discharge side. However, the system can start or end at any point in the piping, provided that the flow and pressure are known at these points. For example, for a lift station, the system typically starts at the water level in the wet well and ends at the discharge pipe tie-in to the existing force main.

The total dynamic head (TDH) represents the energy required to move the wastewater to the destination at a given flow rate. The pumps provide that energy of course. Another word for TDH is “loss”. Engineers use the unit of feet (or meters) for TDH for simplicity in calculations.

A system curve is created by calculating the system losses at several different flow rates and then plotting the points. The following are typical steps for creating a system curve:

1. Gather the following information:
   a) Design flow rates (i.e. ADH and PHF),
   b) Number and type of duty pumps,
   c) Suction and discharge pipe size, material, route, and tie-in elevations,
   d) Suction and discharge fittings and valves, and
   e) Water level elevations or tie-in pressures for the suction and discharge.
2. Define the inputs and outputs (water levels or pressures) at the extreme low and high static operating conditions. For example with a lift station:
   a) Lowest static head: high water level on suction and low tie-in pressure.
   b) Highest static head: low water level on suction and high tie-in pressure.
3. Tabulate and sum the following headloss values for the extreme operating conditions:
   a) minor losses,
   b) pipe friction losses, and
   c) static head change
4. Plot the TDH versus Flow for the two conditions. See Figure 15 for an example.

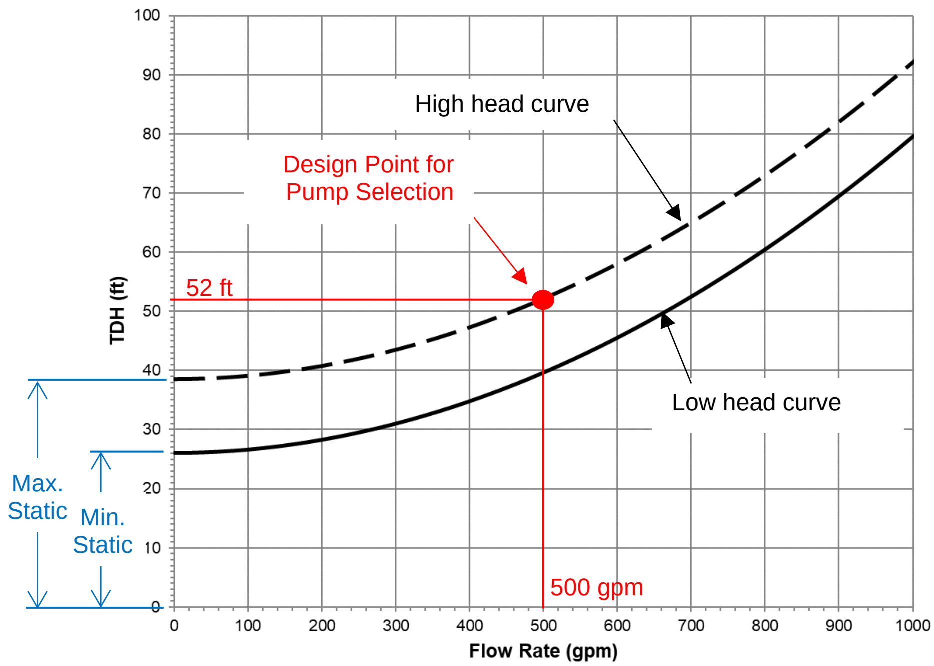


Figure 15: System curves with the operating point in red.

Most engineering firms have standard excel templates or programs for calculating headloss values. Please download the Free Software that comes with this course for a template for performing headloss calculations.

For minor losses (H_{Lm}), the K-value method (shown below) is utilized, which requires finding the resistance coefficient (K) for each type of fitting or valve in the system.

$$\boldsymbol{H_{Lm}} = \sum K \frac{V^2}{2g}$$

where: K = resistance coefficient
V = velocity (typically in fps)
g = acceleration of gravity (32.2 ft/s^2 or 9.81 m/s^2)

For pipe friction losses (H_{Lf}), the Hazen Williams Equation (shown below) is by far the most used formula. Not only is it simple and easy to use, but its use is also required by many regulatory agencies.

$$H_{Lf} = Le_{ft} / 100 * 0.2083 (100 / c)^{1.852} * q_{(gpm)}^{1.852} / d_{(in)}^{4.8655}$$

where: *Le* = pipe length, feet
c = friction coefficient, 120 (cast iron), 140 (ductile iron), 150 (PVC)
q = flow rate, gpm
d = pipe diameter, inches

Coefficients for friction losses and K values for minor losses can be obtained from *Flow of Fluids Through Valves, Fittings & Pipe* by Crane, *Cameron Hydraulic Data* by Flowserve, or other documents listed under Helpful References.

Example 3:
Gloria is a consulting engineer working on a sewer system for a new neighborhood. She is designing a duplex lift station with submersible centrifugal pumps, as shown in Figure 16. The water level varies from EL 4.0 to 6.0. The discharge tie-in pressure varies from 9 to 13 psi with pipe centerline EL 12.0. The PHF is 400 gpm. The total length of 6" ductile iron pipe (DIP) is 400 feet. Calculate the static head, TDH at the PHF, and plot example system curves.

Solution:
First, Gloria calculates the static head change at the two extreme operating conditions:

- Lowest static head:
    - Water level from 6.0 to 12.0, plus low FM pressure of 9 psi
    - $\mathbf{H_{St\_low}}$ = (12.0' – 6.0') + 9 psi * 2.31 ft/psi = **26.8 ft**
- Highest static head:
    - Water level from 4.0 to 12.0, plus high FM pressure of 13 psi
    - $\mathbf{H_{St\_high}}$ = (12.0' – 4.0') + 13 psi * 2.31 ft/psi = **38.0 ft**

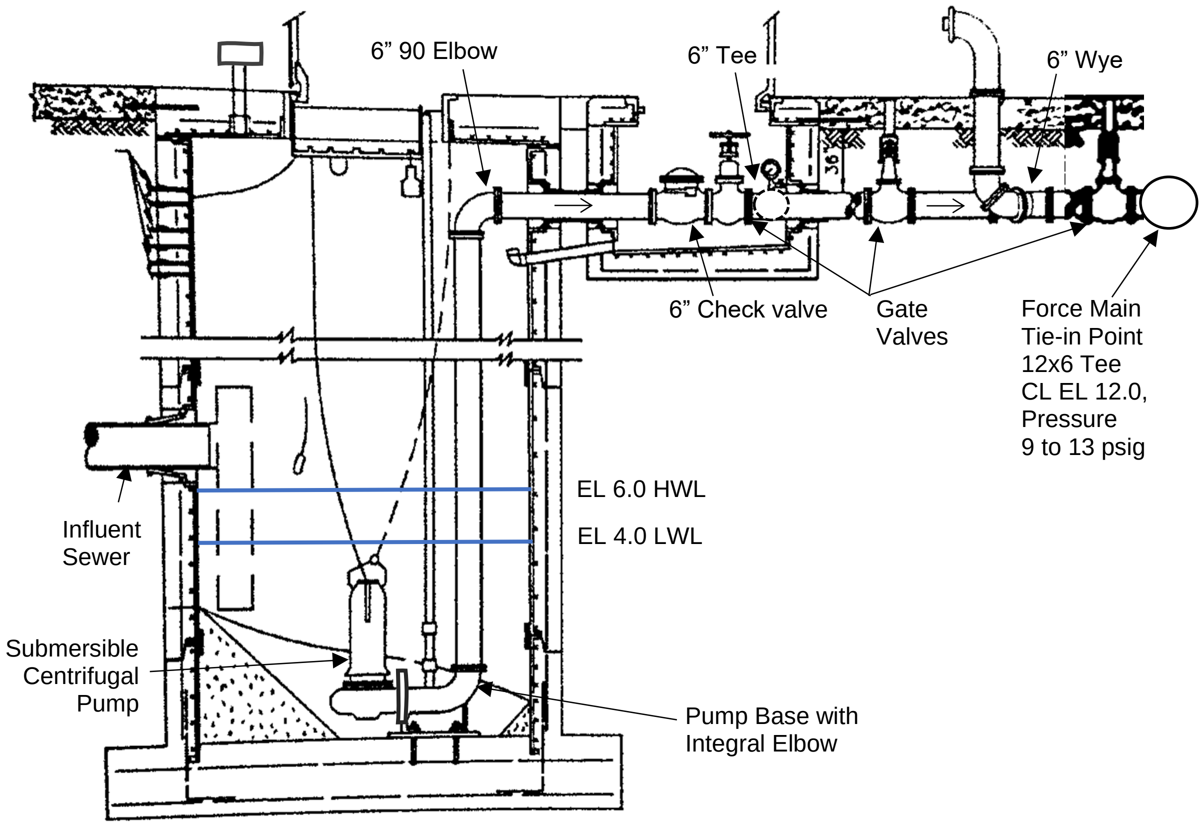


Figure 16: Duplex lift station for Example 3.

Next, Gloria calculates the minor losses at the PHF. Gloria uses the K-value Method to calculate the minor losses at 400 gpm. First, she finds the resistance coefficient, K, for each fitting and valve, then sums the K-values in Table 5.

Table 5: Fittings and Coefficients

Fitting	K-value
Base elbow	0.50
90º elbow	0.30
Check valve	3.00
Gate valve	0.19
Tee – straight thru	0.60
Gate valve	0.19
Wye – straight thru	0.60
Gate valve	0.19
Tee – out branch	1.80
Sum K-values	**7.37**

The minor losses are calculated with the K-value Method equation:

$$\boldsymbol{H_{Lm}} = \sum K \frac{V^2}{2g} = \sum K \frac{\left(\frac{Q_{cfs}}{A_{ft^2}}\right)^2}{2 * 32.2 \frac{ft}{s^2}} = \sum K \frac{\left(\frac{\frac{400_{gpm}}{448.8_{gpm/cfs}}}{\pi\left(\frac{3}{12}ft\right)^2}\right)^2}{64.4\frac{ft}{s^2}} = 7.37 * 0.32 = \mathbf{2.36ft}$$

The pipe friction losses are calculated with the Hazen Williams equation:

$$\boldsymbol{H_{Lf}} = 400\text{ft}/100 * 0.2083 \left(\frac{100}{140}\right)^{1.852} (400\text{gpm})^{1.852}/(6in)^{4.8655} = \mathbf{4.82ft}$$

Gloria sums the static head, minor losses, and friction losses for the total dynamic head, TDH, at 400 gpm. Note that the TDH_{high} value of 45.18 would be used for pump sizing.

$$\boldsymbol{TDH_{low}} = H_{St\_low} + H_{Lm} + H_{Lf} = 26.8\text{ft} + 2.36\text{ft} + 4.82\text{ft} = \mathbf{34.0ft}$$

$$\boldsymbol{TDH_{high}} = H_{St\_high} + H_{Lm} + H_{Lf} = 38.0\text{ft} + 2.36\text{ft} + 4.82\text{ft} = \mathbf{45.2ft}$$

Gloria plots the system curves in Figure 17. The calculations are provided in a tab called "Example 3" in the Free Software that comes with the course.

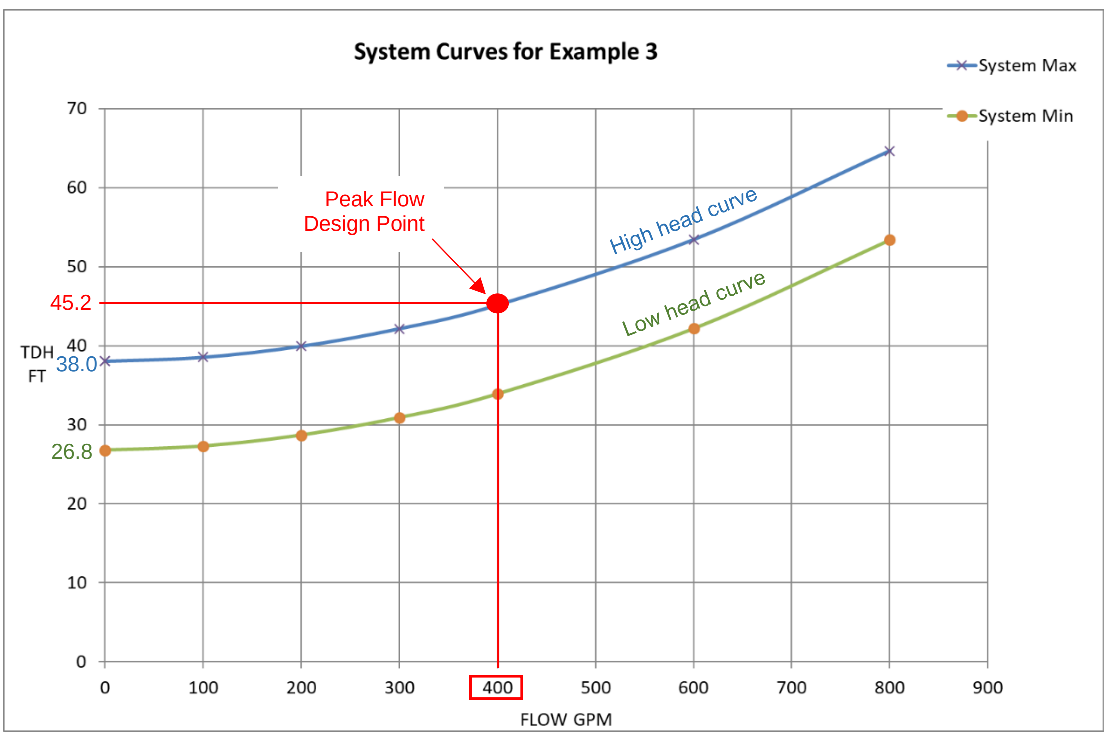


Figure 17: System curves for Example 3 with the design point in red.

When there is more than one duty pump, the calculations become a bit more complicated because the flow rate is not the same through all the piping. If the pumps are the same size and the piping at each branch is the same or only one fitting different, then the flow rate through each branch will be the total flow divided by the number of pumps in operation. The minor losses and friction losses through each pipe branch are calculated with the branch flow rate instead of the total flow rate. System curves can be plotted for each combination of pumps running (one pump running, two pumps running, etc).

Example 4:
Jim is an Engineer-In-Training (EIT) working under Juan, a Professional Engineer. Juan asked Jim to assist him in designing a reclaim pump station with two duty pumps and one standby pump, as shown in Figure 18. The suction water level is to be maintained at a constant EL 4.0. The discharge tank water level is a constant EL 12.0. The PHF is 2,000 gpm. Each branch has 50 feet of 10" DIP. The pipeline to the discharge tank has 4,000 feet of 12" DIP. Jim is to calculate the TDH at the PHF and plot the system curve with two pumps in operation, for Jim's review.

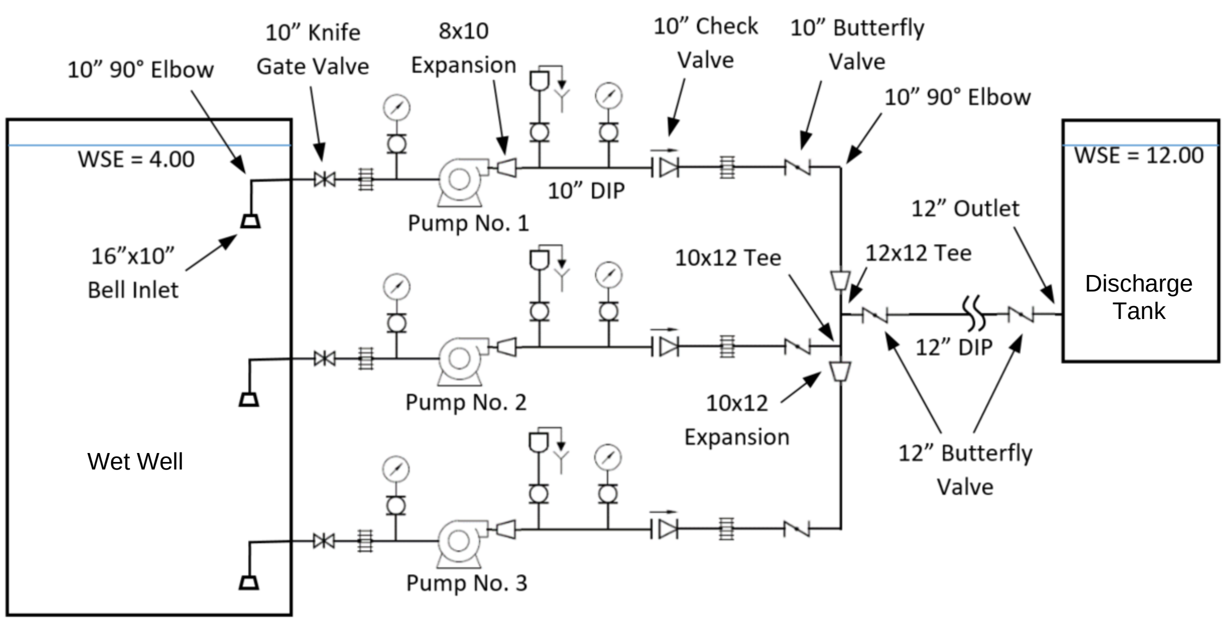


Figure 18: Process flow schematic for Example 4.
Labeled fittings are the same for all branches.

Solution:
Jim determines that since the pipe branches are nearly identical, with two pumps running, the flow through each branch will be 1,000 gpm. Since the suction and discharge water levels are the same, Jim decides that calculating a single system curve is acceptable. To be conservative, he chooses the bottom branch for the calculations, since it has one more fitting than the other two branches.

To start, Jim calculates the static head:

$$\boldsymbol{H_{St}} = 12.0' - 4.0' = \mathbf{8.0\ ft}$$

Jim calculates the minor losses at the PHF. Jim uses the K-value Method to calculate the minor losses at 1,000 gpm in the 10" branch and 2,000 gpm in the 12" pipeline. K-values are summarized separately in Tables 6 and 7. Note that joints, couplings, and small taps have insignificant loss contributions.

Table 6: Branch Coefficients	
Fitting	K-value
Inlet, submerged	0.78
16"x10" Bell	0.20
90º elbow	0.30
Knife gate valve	0.15
8"x10" expansion	0.15
Check valve	3.00
Butterfly valve	0.50
90º elbow	0.30
10"x12" expansion	0.15
Sum K-values	**5.53**

Table 7: Pipeline Coefficients	
Fitting	K-value
Tee – straight thru	0.60
Tee – out branch	1.80
Butterfly valve	0.50
Butterfly valve	0.50
Outlet, wall opening	1.00
Sum K-values	**4.40**

The minor losses are calculated for the branch and pipeline, and summed:

$$\boldsymbol{H_{Lm\_br}} = \sum K_{br} \frac{\left(\dfrac{\dfrac{1000_{gpm}}{448.8_{gpm/cfs}}}{\pi \left(\dfrac{5}{12} ft\right)^2} \right)^2}{64.4 \dfrac{ft}{s^2}} = 5.53 * 0.26 = \mathbf{1.44ft}$$

$$\boldsymbol{H_{Lm\_pl}} = \sum K_{pl} \frac{\left(\frac{\frac{2000_{gpm}}{448.8_{gpm/cfs}}}{\pi\left(\frac{6}{12}ft\right)^2}\right)^2}{64.4\frac{ft}{s^2}} = 4.40 * 0.50 = \mathbf{2.20ft}$$

$$\boldsymbol{H_{Lm}} = \boldsymbol{H_{Lm\_br}} + \boldsymbol{H_{Lm\_pl}} = \mathbf{1.44 + 2.20 = 3.64ft}$$

The pipe friction losses are calculated with the Hazen Williams equation:

$$\boldsymbol{H_{Lf\_br}} = 50\text{ft}/100 * 0.2083\left(\frac{100}{140}\right)^{1.852}(1{,}000\text{gpm})^{1.852}/(10in)^{4.8655} = \mathbf{0.3ft}$$

$$\boldsymbol{H_{Lf\_pl}} = 4{,}000\text{ft}/100 * 0.2083\left(\frac{100}{140}\right)^{1.852}(2{,}000\text{gpm})^{1.852}/(12in)^{4.8655} = \mathbf{32.5ft}$$

$$\boldsymbol{H_{Lf}} = \boldsymbol{H_{Lf\_br}} + \boldsymbol{H_{Lf\_pl}} = \mathbf{0.3 + 32.5 = 32.8ft}$$

Jim sums the static head, minor losses, and friction losses for the total dynamic head, TDH, at 2,000 gpm.

$$\boldsymbol{TDH} = H_{St} + H_{Lm} + H_{Lf} = 8.0\text{ft} + 3.64\text{ft} + 32.8\text{ft} = \mathbf{44.4ft}$$

Jim plots the system curve shown in Figure 19. The calculations are provided in a tab called “Example 4” in the Free Software that comes with the course.

Jim schedules a meeting with Juan to present the results and to provide Juan the calculations for his review.

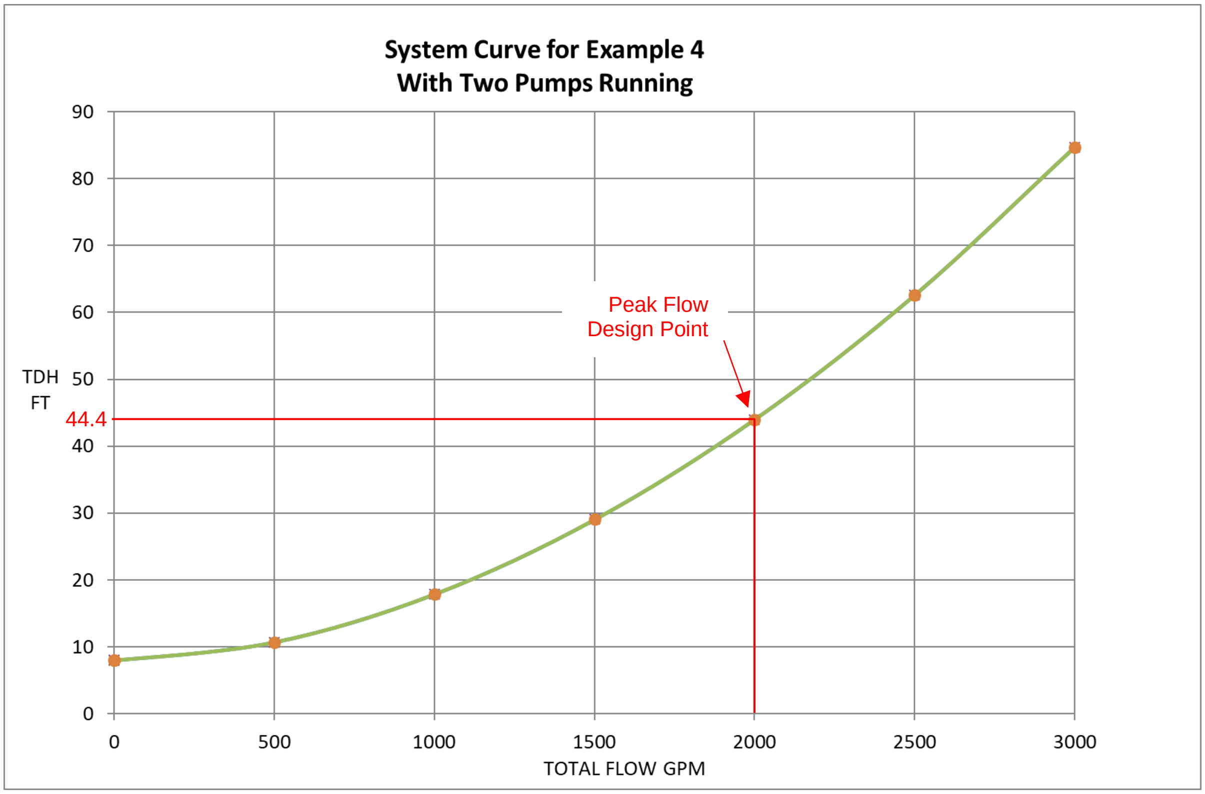


Figure 19: System curve for Example 4 with the design point in red.

Example 5:
Continuing from Example 4, Juan asks Jim to create a table that summarizes the design criteria for the pumps. Jims asks if there is a minimum flow rate. Juan says that in his experience the minimum flow would be around 500 gpm.

Solution:
Jim selects end-suction centrifugal type pumps because they are commonly used for reclaimed water, are reliable, and are economical. He selects variable frequency drives for pump speed adjustments which will allow maintaining a water level in the wet well.

Jim recognizes that the minimum flow rate condition would be with one pump running versus the peak flow rate condition with two pumps running. With one pump running, 100% of the flow goes through a single pump branch. The previously calculated system curve was with 50% of the flow through a pump branch. Therefore, a new system curve is calculated for the one pump running condition, as shown in Figure 20. At 500 gpm, the TDH is 11.0.

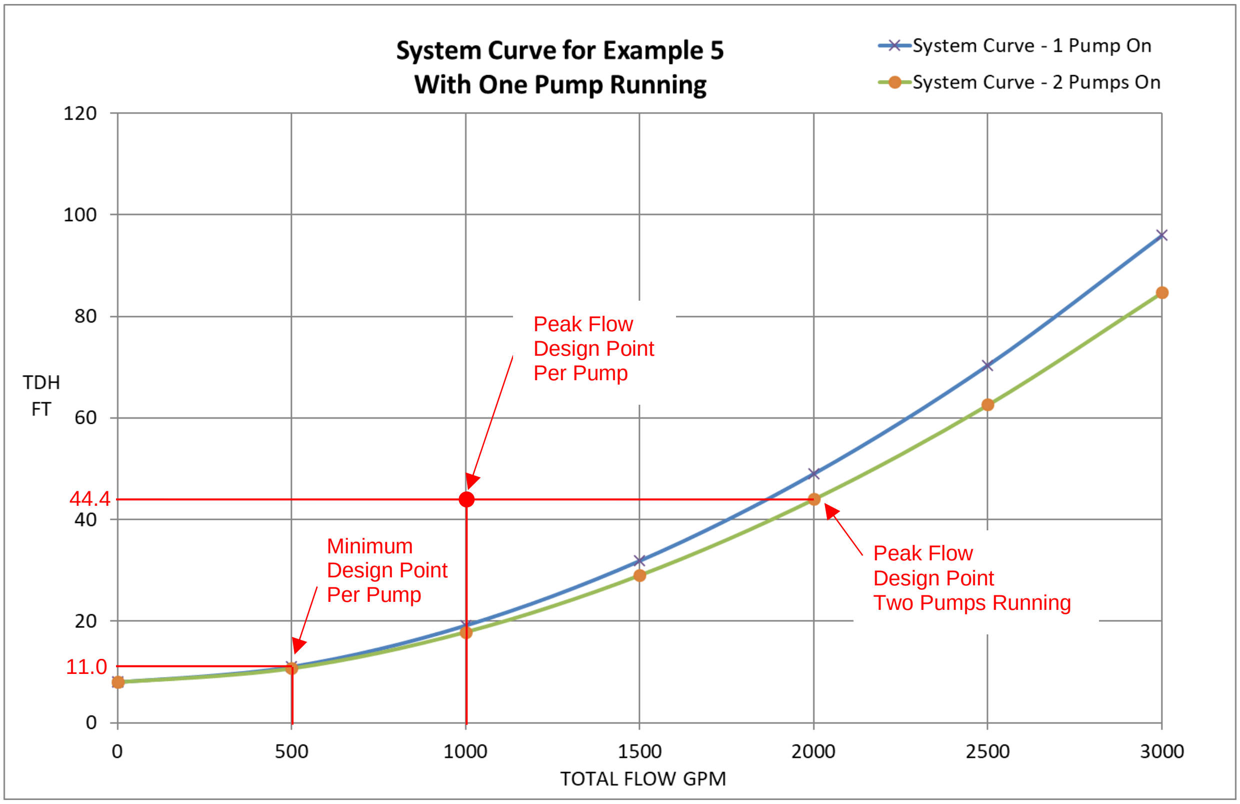


Figure 20: Example 5 system curves with one pump running (blue), and two pumps running (green). Design points are in red.

The hydraulic calculations are provided in a tab called “Example 5” in the Free Software that comes with the course.

Jim creates Table 8 to summarize the pump design criteria, and presents this to Juan for his review.

Table 8: Reclaim Pump Station Design Criteria		
Fluid Type	Reclaimed Water	
Pump Type	End-suction centrifugal	
Number of Pumps	3 (2 duty + 1 standby)	
Drive Type	Variable frequency	
Flow Conditions	Flow Rate (gpm)	TDH (ft)
PHF, Total	2,000	44.4
MDF, Total	500	11.0
Pump Design Point - Peak Flow	1,000	44.4
Pump Design Point - Minimum Flow	500	11.0

If there is a difference in the piping at each branch, or if the pumps are different sizes, then the calculations become more complicated. In such cases, the following options are available:

1. Iterative Calculations: Assume the flow balance between the branches, run the calculations, check the pressures at the discharge connection node, adjust flow rates, run the calculations again, check pressures again, and continue iterations until the pressures are the same at the discharge connection node.
2. Hydraulic Modeling: The piping network is entered into a hydraulic model or similar software to solve for TDH at various flow rates and to create a system curve for each branch.
3. Pump Curves with Branch Losses: With this approach, the pump curve is adjusted to include the branch losses. First, system curves are created without the unique parts of the branches at each pump. Next, the branch losses are calculated at various flow rates. Then, when choosing a pump size, the head losses at the branch are subtracted from the pump curve. This modified "pump+branch curve" is plotted with the system curves to confirm the operating point. See the next Section regarding plotting pump curves with system curves.

Pump Curves and Selection

The importance of making a good pump selection cannot be understated. The main goal is to choose pumps that meet the flow demands while staying within the operating range of the pump. This is known as pump selection or pump sizing. The design engineer is expected to use pump curves, also called performance curves, to choose and confirm the proper pump selection.

The following are typical steps for the pump selection process:

1. Review pump design criteria and system curve(s)
2. Select the type of pump
3. Review pump manufacturer literature including pump curves
4. Make preliminary pump selections using design points
5. Compare and choose a pump
6. Plot pump curve on the system curve
7. Confirm pump capacity at different pump conditions
8. Review net positive suction head (NPSH)
9. Select motor HP
10. Design pump connections, mounting, rails, hatches, etc.
11. Check and adjust wet well dimensions, intake dimensions, and pipe sizes

Examples of pump selection techniques are provided in this section. For further details, consult the Helpful References Section.

Preliminary Selection

Preliminary pump selection is the process of reviewing pump models and choosing one or more good fits for the design conditions. A good start is to review a chart of the capacity ranges of various pump models, as shown in Figure 21. This allows choosing the pump model. Although many pump curves are available in catalogs or websites, it is still a good practice to request the curves from the pump supplier while also confirming the pump is a good fit for the application.

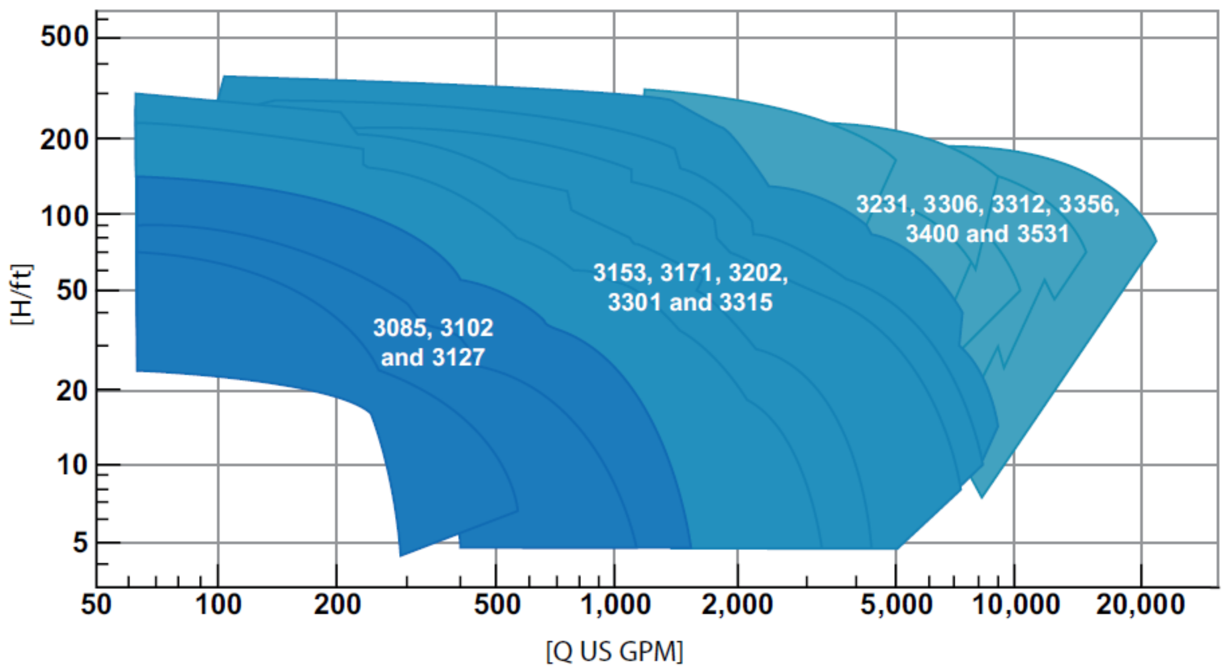


Figure 21: Example chart of capacity ranges for various pump model numbers.
https://www.xylem.com/siteassets/brand/flygt/flygt-resources/flygt-resources/fb002-896876_flygt_n-pump_series.pdf

Detailed Pump Selection

After the pump model is identified, detailed curves should be reviewed for different pump sizes. Note that catalogs typically show curves for full-size impellers or nominal size impellers. To get a pump selection that exactly hits the design point, the impeller often needs to the "trimmed" so the pump curve hits the precise design point. Some pump manufacturers provide software or online tools that allow the creation of trimmed impeller pump curves.

In reviewing pump curves that meet the design point, consider which curve provides the following:

- The best efficiency at the average operating condition, which will be between the high and low head conditions.
- All operating points are in the preferred operating range, which is between 70% to 120% of the flow rate at the best efficiency point.
- The curve is not excessively flat, excessively steep, and is smooth without hills or valleys.
- The flow rate at the intersection of the low head system curve and the pump curve is not excessive (typical of a flat pump curve).

- The lowest maximum shaft power across the full recommended operating range, which is the boldened part of the pump curve.

<u>Affinity Laws</u>
When designing for variable speed pumps, curves are needed for full speed and minimum speed (typically 50%). If the pump manufacturer does not provide curves at varying speeds, the design engineer should calculate them using the following Affinity Laws (see Table 9 for an example):

- Flow rate: The flow rate varies proportionally with the speed change. For example, half the speed results in half the flow: 1/2 speed = 1/2 flow
- TDH: The pump head varies with the square of the speed change. For example, half the speed results in one-fourth the head: $(1/2 \text{ speed})^2 = 1/4$ head
- Power: The power consumption varies by the cube of the speed change. For example, half the speed results in one-eighth the power: $(1/2 \text{ speed})^3 = 1/8$ power.

Table 9: Example Calculation of Pump Curve Points using Affinity Laws			
Full Speed Pump Curve (Given)		1/2 Speed Pump Curve (Calculated with Affinity Laws)	
Flow (gpm)	Head (ft)	Flow (x1/2) (gpm)	Head ($x(1/2)^2$) (ft)
0	50	0	12.5
500	47.5	250	11.9
1000	45	500	11.25

<u>Multiple Pumps</u>
When there are two or more duty pumps in parallel, it is helpful to plot a combined pump curve showing the effect of multiple pumps running at once. The combined pump curve is obtained by adding the pump flow rates at the same head. See the dark red curve in Figure 22 for an example combined pump curve for two pumps in parallel.

Example 6:
Continuing Example 5, Juan asks Jim to plot curves with the system curves for the reclaim pump station design. Operations staff gave a preference for the pump manufacture and model number. Jim is to confirm the pump model will perform at the design conditions.

Solution:
Jim selects a pump curve that meets the design condition of 44.4 ft at 1,000 gpm. He plots the pump curve at 100% speed, from the manufacture, and at 50% speed, using Affinity Laws. At 50% speed, the pump curve is below the minimum flow design point, so the pump can operate at the low design point. See Figure 22 for the result.

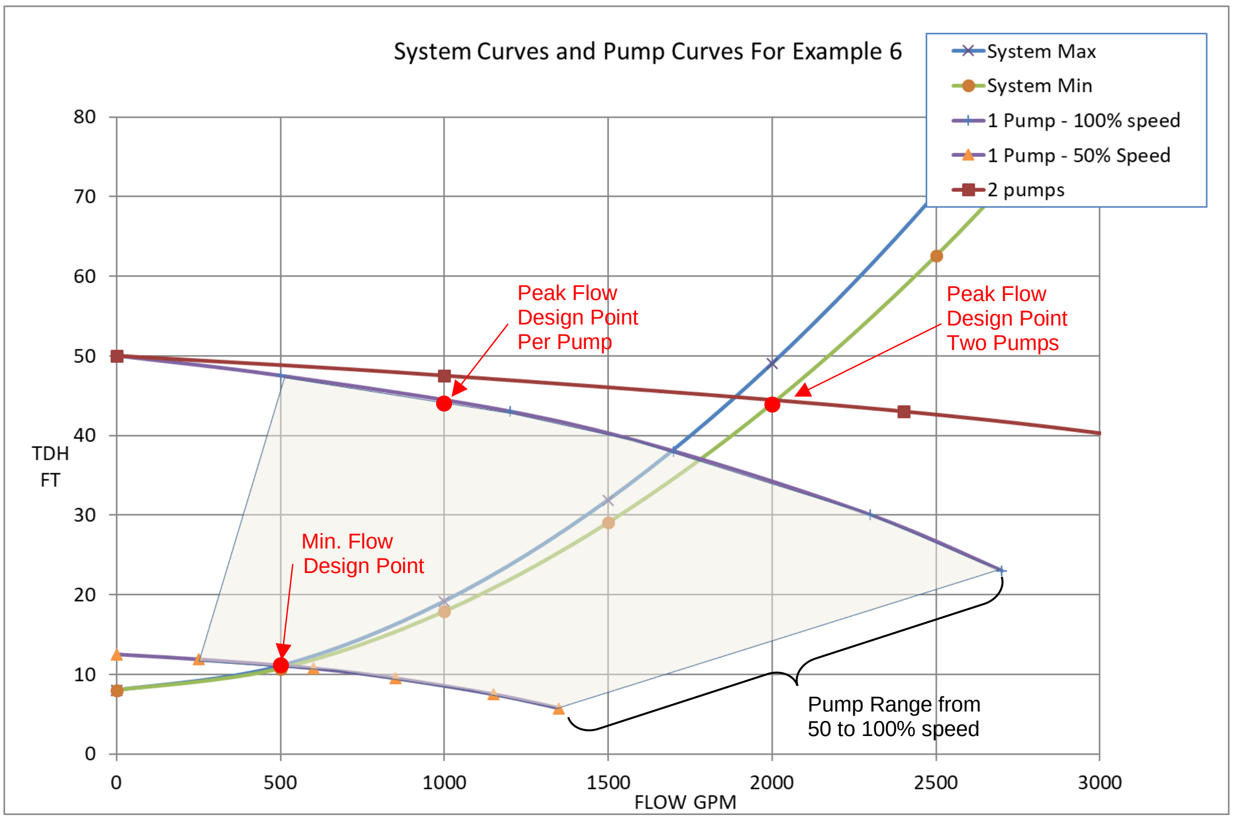


Figure 22: Plot of system curves and pump curves for Example 6.

Note that if the low flow point cannot be achieved, the following options are available:

- The number of duty pumps can be increased,
- Use a combination of large and small pumps,
- Add a modulating valve or pressure sustaining valve to increase the headloss during low flow conditions.

Example 7:
Engineer Beverly is designing a pump station. She has already calculated the low and high head system curves and found a pump that meets the design condition of 500 gpm at 52 ft TDH. Now she needs to plot the pump curve with the system curves, identify the pump operating flow range, confirm the best efficiency point is within the range, and select the motor HP using a service factor of 1.15.

Answer:
Beverly obtained the manufacturer pump curve, as shown in Figure 23. She plotted the the pump curve and the system curves together in Figure 24. Based on the intersections of the curves, the normal operating range would be 500 to 615 gpm, with the best efficiency point (BEP) in that range at 570 gpm. She added notes to the curves showing the range and BEP for clarity.

Beverly reviewed the shaft power (HP) curve in the center chart of Figure 23. She identified the greatest power is 10.9 HP at the far right of the curve. To determine the motor HP, she multiplied this line shaft HP by the service factor, and divided by a typical motor efficiency of around 90%:

$$Minimum\ Motor\ HP = \frac{Line\ shaft\ HP * 1.15}{motor\ eff} = \frac{10.9\ HP * 1.15}{0.90} = 13.9\ HP$$

Beverly rounded up to the next nominal motor size, which is a **15 HP** motor.

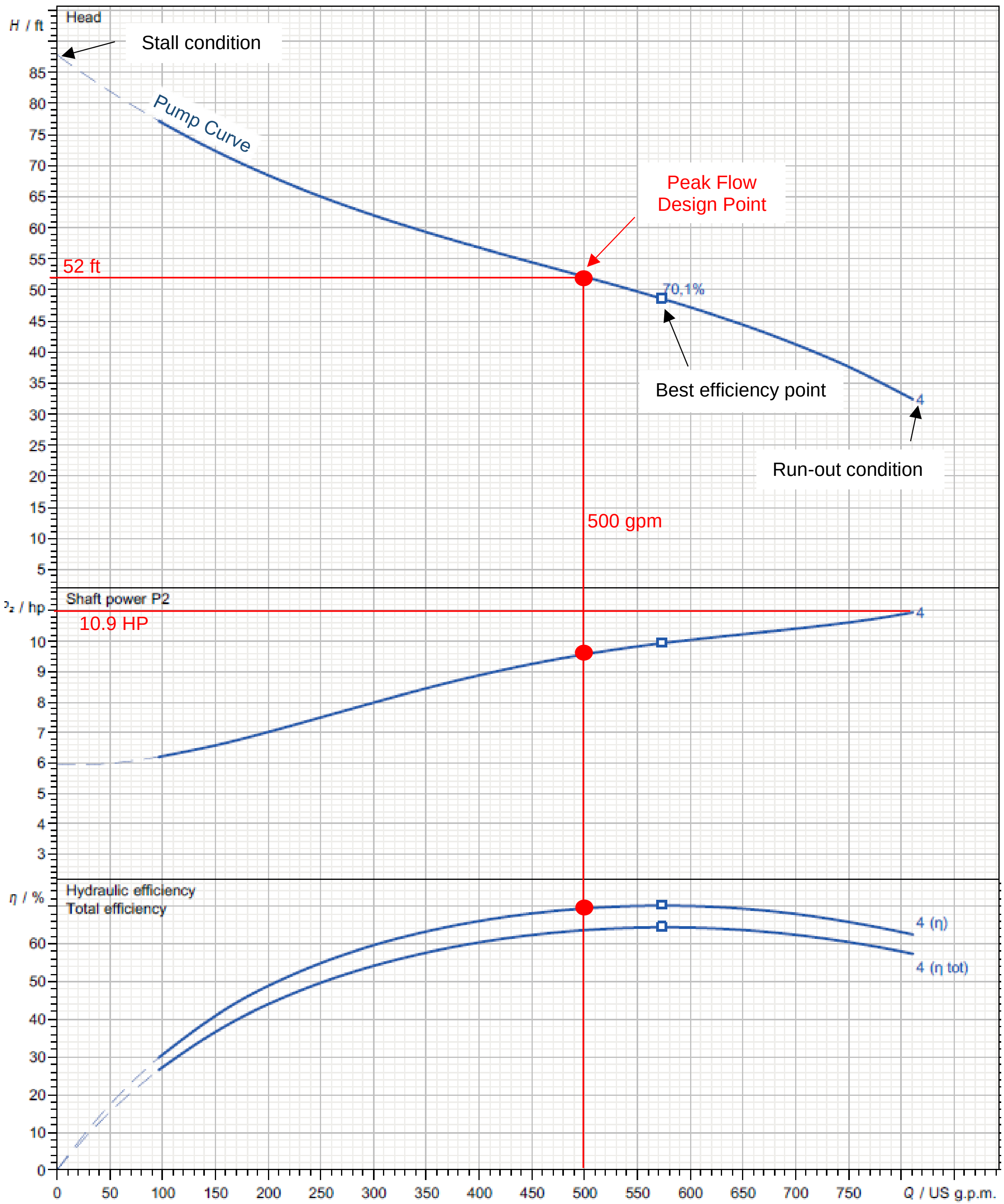


Figure 23: Manufacturer's pump curve for Example 7, with design conditions in red.

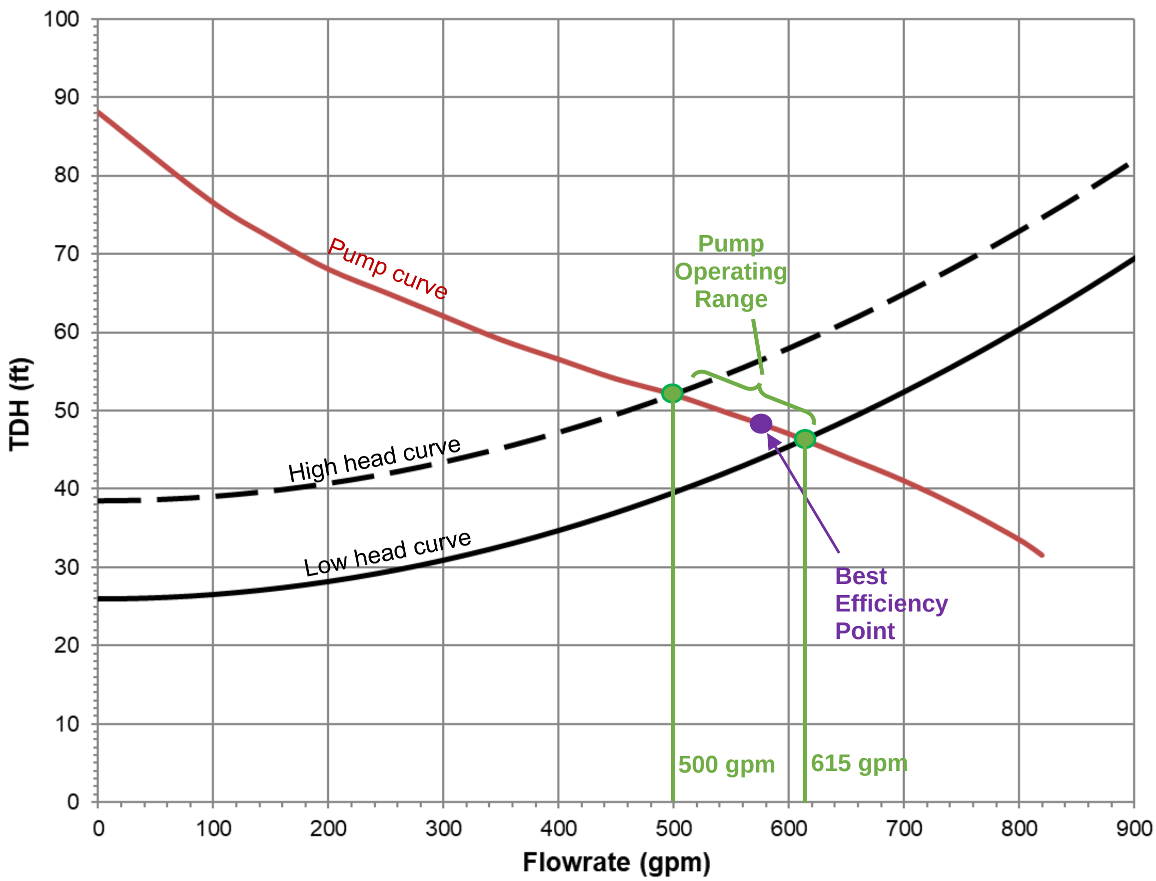


Figure 24: Plot of the pump curve and the system curves for Example 7, showing the best efficiency point within the pump design operating range of 500 to 615 gpm.

Hydraulic Profile

A hydraulic profile is a schematic elevation view of main processes or components with a hydraulic grade line drawn at one or more flow rates. The hydraulic grade line is the fluid elevation plus the pressure head. It does not include the velocity head (like the energy grade line).

The slope of the hydraulic grade represents the head losses in the piping. The vertical rise at the pumps represents the TDH for the pump design. Thus the hydraulic profile is a visual representation of the hydraulic calculations. Mistakes in hydraulic calculations are often discovered when the hydraulic profile is created. See Figure 25 for an example with a hydraulic grade line drawn at the rated flow rate (firm capacity).

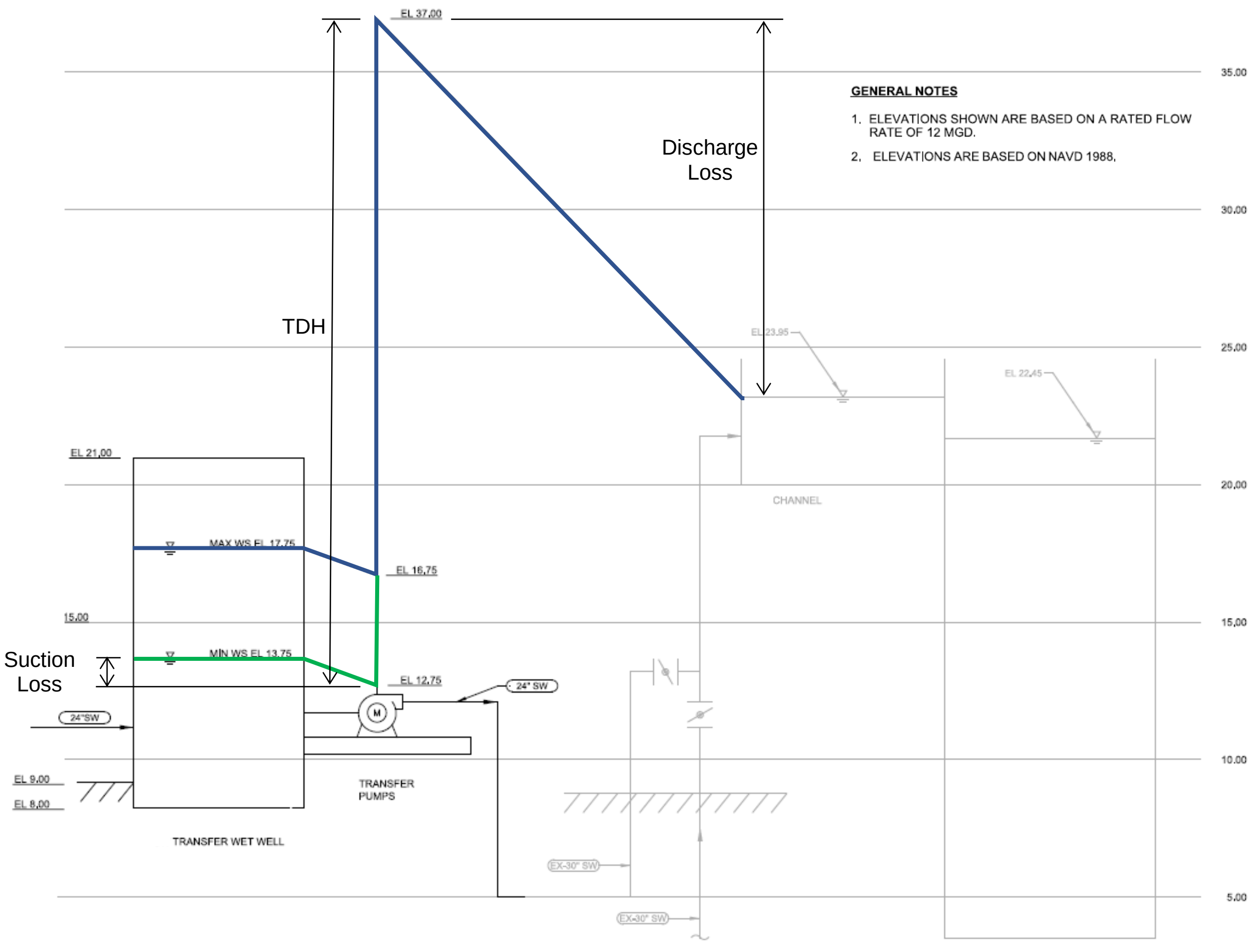


Figure 25: Example hydraulic profile for pumping from a wet well into a channel. The hydraulic grade line is in blue at high water level and green at low water level. The vertical line above the pump represents the TDH in feet: 37.00 – 16.75 = 20.25 at high level and 37.00 – 12.75 = 24.25 at low level.

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