Back to Basics

Basics of Flow with Constant and Variable Speed Pumps

In order to understand the pump operation in the process system, the Process and Control Engineers need to understand the information the pump manufacturer provides and the pump's relationship to the system into which it will be installed.

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Controlling Flow with Constant and Variable Speed Pumps

1. Forward:

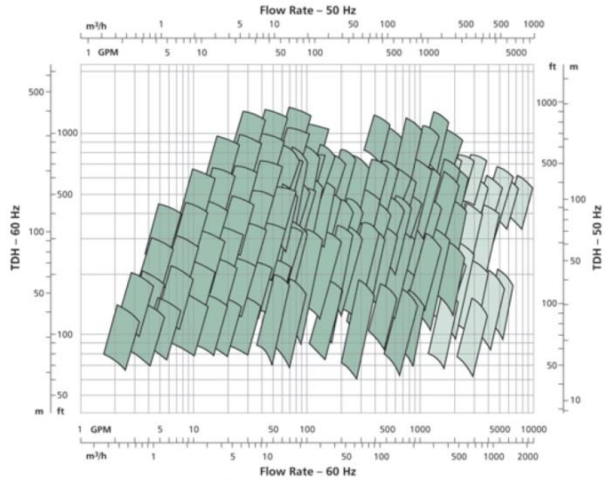
Many systems are designed utilizing centrifugal pumps to move fluid through the system. Currently there is an effort to minimize operational costs by upgrading constant speed pumps with Variable Frequency Drives (VFDs) or providing an initial design using VFDs. Process Engineers and Control Engineers need to understand the proper application, operational requirements and constraints associated with both the constant speed and variable speed designs.

In order to understand the pump operation in the process system, the Process and Control Engineers need to understand the information the pump manufacturer provides and the pump's relationship to the system into which it will be installed.

This whitepaper will review typical pump curves provided by the manufacturer, the relationship of these curves to the system in which the pump is installed and the differences between the constant speed and variable speed operation.

Older pump operating and system resistance curves in the plant files are typically based on constant speed operation of the pump. By using the pump "Affinity Laws" the control and process engineers can generate variable speed curves for that pump. Using these updated curves the engineers can then evaluate and recommend approaches to the system control, pump protection and develop associated costs and benefits.

2. Pump Curves

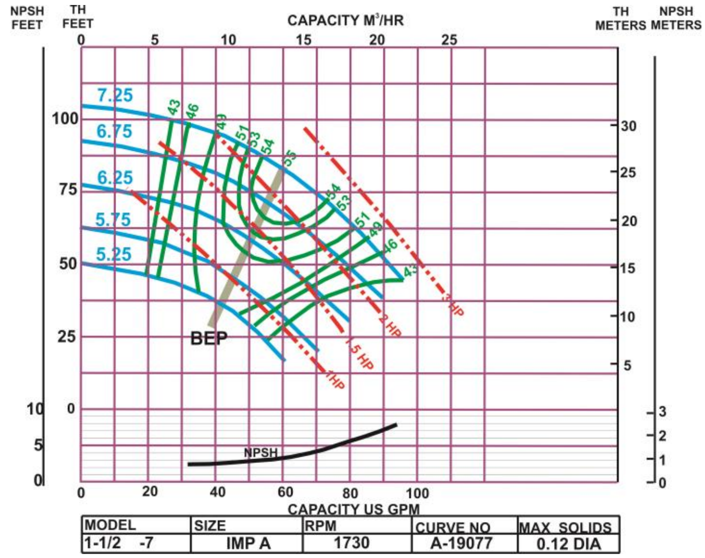


Hydraulic Coverage

Each pump manufacturer reviews the application into which their product will be installed and selects or recommends a pump model/style/size appropriate to the proposed application. Generally the particular pump is selected using a selection chart that aligns the manufacturer's models with respect to developed head and flow. This approach will attempt to find the best casing and impeller size that will meet the original specifications and expected variation and degradation of the flow system. The picture above is a Gould Pump Selection chart. Each of the "tombstones" represents a particular pump model and its flow-head relationship. This chart helps the manufacturer's salesperson narrow the search for a particular pump model/style/size.

Once the pump model/style/size is selected, the manufacturer will provide the appropriate pump curves applicable to that model. Occasionally the customer may request a "bench test" of the pump (typically for critical flow applications) under the expected flow conditions and provide a custom pump curve. In general, however, geometrically similar pumps will not appreciably deviate from the 'standard' pump curves for that pump type.

2.1. Fixed Speed Curves



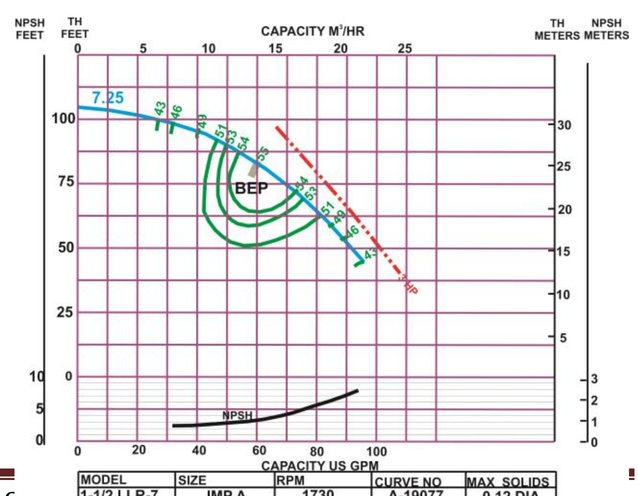
The typical curve provided to the customer contains more information than they expect. Current curves include information in both English and Metric units. The table at the bottom of the chart indicates the particular model (or serial number if a 'bench test' has been performed), the impeller type, the speed at which the chart was generated and any other information common to that pump type.

2.1.1. Diameter (See Chart in Section 2.1)

The first thing that is noted is that there are several curves (blue lines) that relate the pump's flow-head relationship. The pump casing is capable of supporting several different diameter impellers. The manufacturer will initially recommend and provide an impeller with the proper size to match the customer's requirements. For our process examples I will use the 7.25 inch impeller and associated data. Multiple impeller size options are provided to allow the customer to replace the impeller with a more appropriate size after commissioning if a better match to the system is required. If an adjustment is needed the customer may purchase a new impeller of the needed size or the customer my wish to machine the existing larger impeller if a smaller size is needed.

2.1.2. Flow vs Pressure

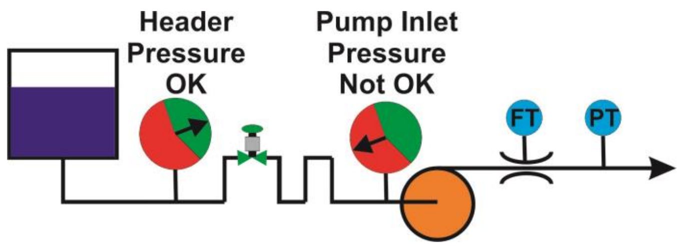
Once the impeller diameter is established, the pump will operate along that blue flow-head curve. As the system flow changes, the pressure developed by the pump will follow the curve appropriate for that impeller diameter. The remaining impeller curves can now be ignored.



2.1.3. NPSHR, NPSHA (See Chart in Section 2.1)

At the bottom of the chart is another curve labeled "NPSH". This curve represents the Net Positive Suction Head required (NPSHR) at the INLET OF THE PUMP. The pressure at the inlet of the pump MUST be greater than or equal to this value above the saturation pressure of the fluid that is being pumped.

As the fluid enters the eye of the impeller the fluid velocity changes (increases) and the local pressure of the fluid changes (decreases). If the pressure decreases too much, the fluid will fractionate forming vapor bubbles. Problems do not necessarily occur at this point. As the fluid passes through the vanes of the impeller the pressure increases causing these vapor bubbles to collapse. The collapsing bubbles release energy and can remove material from the impeller or casing. Over a period of time this damage can result in the failure of the pump casing or impeller.



The reason the "inlet of the pump" is capitalized and underlined in the first paragraph of this section is that using pressure measurements in the piping upstream of the pump suction is not acceptable. Many times the process designers do not account for the piping pressure losses, restrictions or adverse dynamics between the inlet header and the pump inlet. The resulting pressure, Net Positive Suction Head Available (NPSHA), at the inlet falls below the NPSHR.

Many pump manufacturers provide a plugged connection at the pump inlet flange to allow the customer to attach a pressure gage at the correct location. They also provide a similar plugged connection at the discharge of the pump to allow proper measurement of the discharge pressure. These two points are the basis for the developed head for the associated pump curves.

2.1.4. Efficiency (See Chart in Section 2.1)

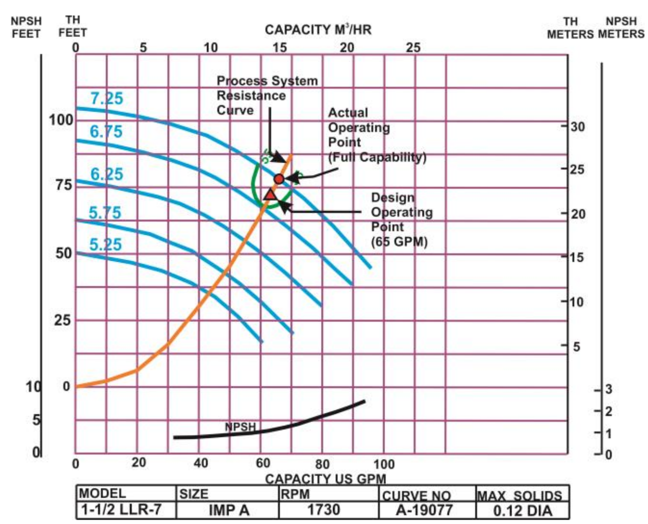
The manufacturers chart will also include efficiencies (green lines) associated with flow and pressure for that particular pump. For each pump model/style/size the manufacturer has identified and charted the mechanical pump efficiency. This information is overlaid on the flow-head chart to provide a visual relationship between the flow and efficiency at a particular operating point.

Mechanically the pump is happiest operating at the best efficiency point. A greater percentage of the power input is delivered to the fluid at the higher efficiencies. At lower efficiencies more power may be wasted in fluid heating, fluid turbulence and additional mechanical stresses on the pump. Ideally the pump should be operated at the best operating point both for efficiency and longevity of the pump.

2.1.5. Horse Power (HP) (See Chart in Section 2.1)

The final element on the chart is the input power (red lines) required to operate the pump under all of the expected operating conditions. This provides the system designer with the information needed to select a motor and its associated equipment.

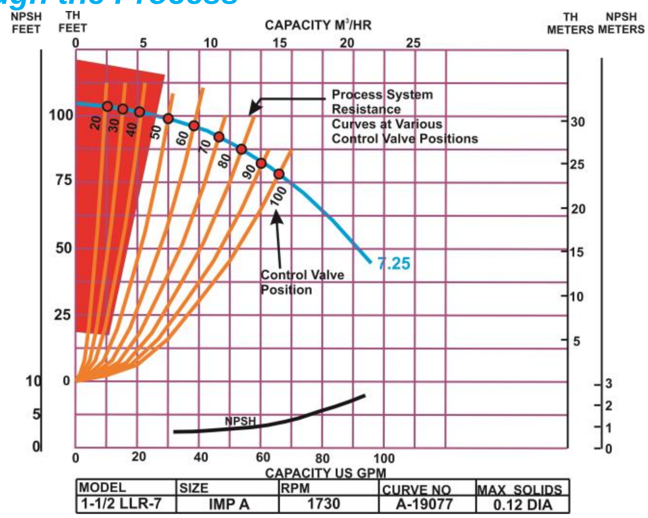
2.1.6. Operating Point



The pump does not operate independent of its associated process system. It is a balance between the capability of the pump and the resistance presented by the system. During the original selection of the pump and associated impeller, a system resistance curve is produced which reflects loss of head with the corresponding flow through the system. The impeller was selected to meet or exceed the requirements of the process system. As noted above, we have selected the 7.25 inch impeller which exceeds the design operating flow.

2.1.6.1. Control of Flow through the Process

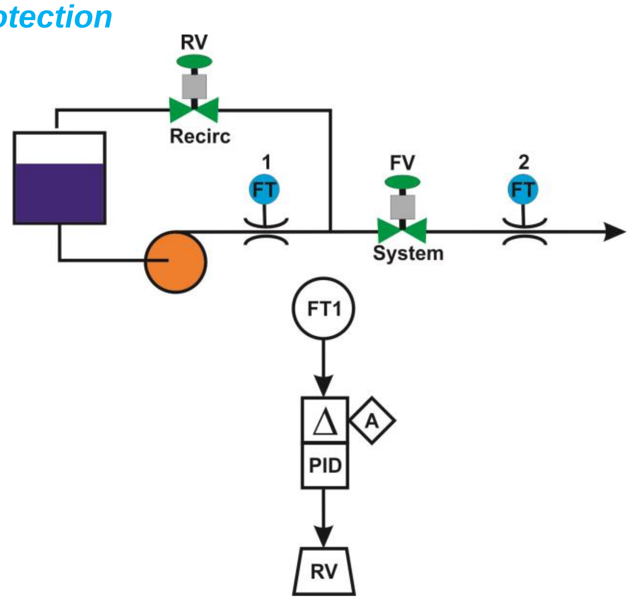
Since the 7.25 inch impeller was selected, the flow will balance at the intersection of the pump pressure-flow curve and the system resistance curve. To provide an adjustable flow, a control valve is included in the process flow discharge piping to adjust the flow to the process. (Increase system head losses).



As the control valve decreases position, the flow-resistance curve changes by increasing the system pressure losses for lower flows. However the pump has some operating constraints. The pump should not be operated below a flow determined by the manufacturer. In this case this region is noted by the red area at the low flow end of the curve. Flows through the pump in this region can cause flow instability in the pump casing, high dynamic stresses on the pump shaft or inadequate flow to carry away the heat in the casing. There have been instances in which a pump with no or very little flow have burnt the paint from the casing and damaged the casing and impeller.

2.1.6.2. Pump Control and Protection

We can see from the chart above that flows corresponding to valve positions lower than 45% result in pump operations in the unstable/undesirable region of the pump curve. Flow through the process system (metered by FT2) is a wild variable controlled by process needs. We need to protect the pump by maintaining a minimum flow through the pump (metered by FT1). There are two approaches to protecting the pump.



The first is to toggle open a recirculation valve (RV) when the process flow control valve (FV) position closes below 45% to assure adequate flow through the pump. The RV would close after the position of the FV opens beyond a predefined dead band (>45% + dead band) allowing adequate flow through the pump. This arrangement protects the pump with a minimum of components (position feedback or switch on FV, a solenoid operated RV and possibly a flow restriction orifice in the recirculation loop). This is an adequate arrangement if there is only occasional operation near this region of the pump curve. One major drawback of this type of system is the step-change to the head available to the process system and may cause some downstream process flow disruptions if the process must operate near this area of the pump curve.

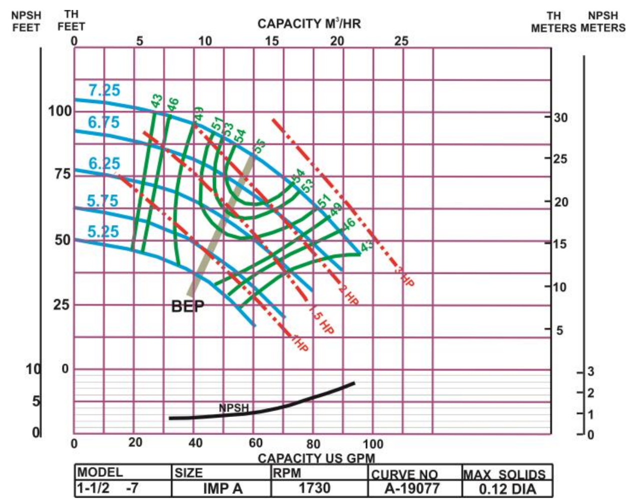
If there is a full range operation of flow to the process, then it may be necessary to add a flowmeter (FT1), controller and a positioner on the recirculation valve (RV) as shown in the figure above. The setpoint for the controller would be the minimum flow of the pump operation (25 GPM in this example). The controller should be set to prevent reset-windup in the closed position. We want to make sure the RV will begin controlling as soon as it is needed. As the flow changes in the process system, the recirculation controller will uniformly and steadily maintain the minimum flow through the pump. This control will minimize the interaction between the pump recirculation and flow to the process. The drawback to this arrangement is the additional cost (flowmeter, controller and positioner on the recirculation valve).

2.2. Variable Speed Pump Curves

Manufacturing companies and engineers are being encouraged to replace constant speed pumps with Variable Frequency Drives (VFD). This encouragement is driven by the potential to save money in operating costs along with potential rebates by the power companies. The natural response is to jump along with the rest of the herd and upgrade or select a new pump-motor combination with a VFD.

The process and control engineers should examine the process and determine the actual savings and appropriate application of these VFDs. They also should have a fundamental understanding of the physics and appropriate application of this equipment. Many times the documentation available to the engineers does not reflect variable speed operation of the pumps.

2.2.1. Flow vs Pressure



The pump curves that are normally available to the engineer are the pump selection curves generated at a fixed speed. This is especially true for upgrades from constant speed to VFD applications. You will notice that the chart includes all of the available impellers but has not been limited the specific impeller selection we are using in these examples (7.25 inch). All of the data is generated at 1730 RPM. (This data represents a 4-pole induction motor with an unloaded speed of 1800 RPM). There is a 70 RPM slip between the line and motor shaft while under load. This chart is not immediately useful to the engineers since we need to view only one impeller size (7.25 inch) at multiple speeds.

We need to create a chart applicable for a variable speed application. For this we apply the Pump Affinity Laws to generate a useful chart.

2.2.1.1. *Pump Affinity Laws*

The pump affinity laws relate the characteristics of flow, pressure and power with pump speed and impeller diameter in geometrically similar pumps. Since we will be dealing with the same pump, it is indeed geometrically similar to itself. For changes in pump speed or impeller diameter the following relations apply:

Affinity Law Relationships

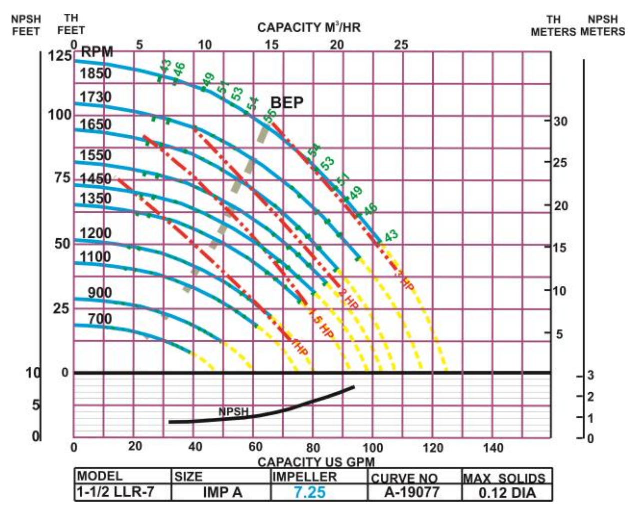
Change of Diameter	Change of Speed
$\frac{Q1}{Q2} = \frac{D1}{D2}$	$\frac{Q1}{Q2} = \frac{N1}{N2}$
$\frac{H1}{H2} = \left(\frac{D1}{D2}\right)^2$	$\frac{H1}{H2} = \left(\frac{N1}{N2}\right)^2$
$\frac{HP1}{HP2} = \left(\frac{D1}{D2}\right)^3$	$\frac{HP1}{HP2} = \left(\frac{N1}{N2}\right)^3$

Where:

Q = Volumetric Flow
D = Impeller Diameter
N = Speed
H = Developed Head
HP = Mechanical Horsepower

Flow through the system is linearly proportional to the impeller diameter ratio or speed ratio. Pressure developed is proportional to the square of the impeller diameter ratio or the speed ratio. Power required is proportional to the cube of the impeller diameter ratio or the speed ratio.

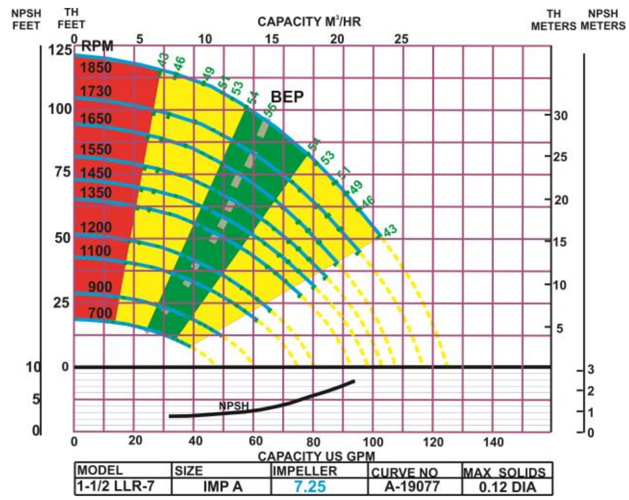
From this set of laws we can generate a new chart that reflects the installed pump capabilities at different motor speeds.



The manufacturer has provided the details of the pump for 1730 RPM. By using this base speed we can take each point of the original curve ($flow_{base}$, $head_{base}$) and replot a new point representing a change in pump speed. This conversion is duplicated over the full flow range. New flow is linear with the speed ratio, new head is a square of the speed ratio. This new point ($flow_{new}$, $head_{new}$) represents one additional point on the new speed curve. This may appear to be a laborious task, but a computer and spreadsheet can make the task more manageable. Each pump speed must be calculated in a similar fashion and plotted to create a new set of pump curves applicable to a VFD application.

2.2.1.2. *Pump Constraints with VFD*

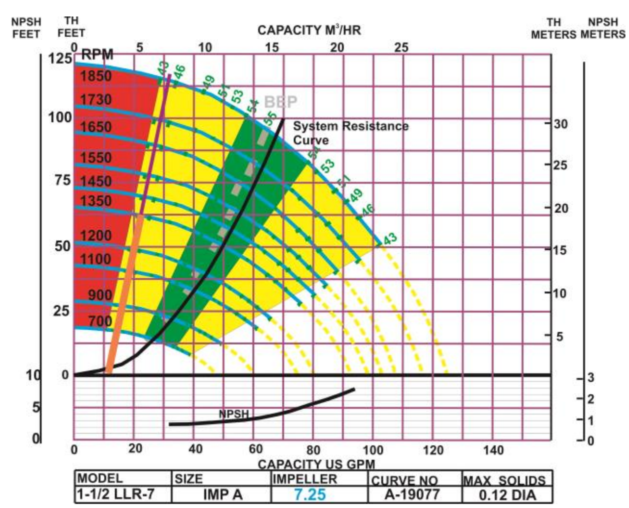
Running a pump with a VFD does not eliminate problems or constraints. These constraints must still be respected but can be modified to reflect the change in impeller speed. There are additional constraints and problems that may occur because of the variable speed.



A new chart has several representative areas that are noted. Ideally it would be nice to operate the pump in the green region near the Best Efficiency Points at each speed. The yellow regions represent the acceptable areas to operate the pump. Operation of the pump increases bearing loading in the yellow areas and the efficiency decreases slightly. The red area still represents a do-not-operate in this region. However you will note that it decreases as function of RPM. We need to respect the NPSH for the pump since the velocities at the inlet of the pump are still related to the flow at the inlet eye of the impeller.

It should also be noted that many VFDs can be operated beyond the normal power line frequency. It is possible to exceed the capability of the motor to deliver the required horsepower to the pump, resulting in a burned-out motor or possibly mechanical damage to the process if the resulting pressure exceeds the downstream equipment capabilities.

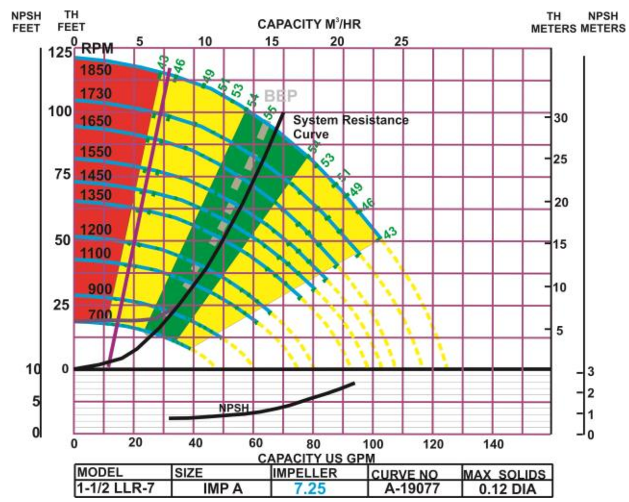
2.2.2. *Operating Point*



In the same fashion that we plotted the fixed speed pump with a system resistance curve, we can place the system resistance curve on the variable speed pump curve. In this example we have an ideal application in which the system resistance and flow requirements beautifully match pump speed with operation in the desired operating area. In this application the VFD will save lots of money since the power savings will be very good.

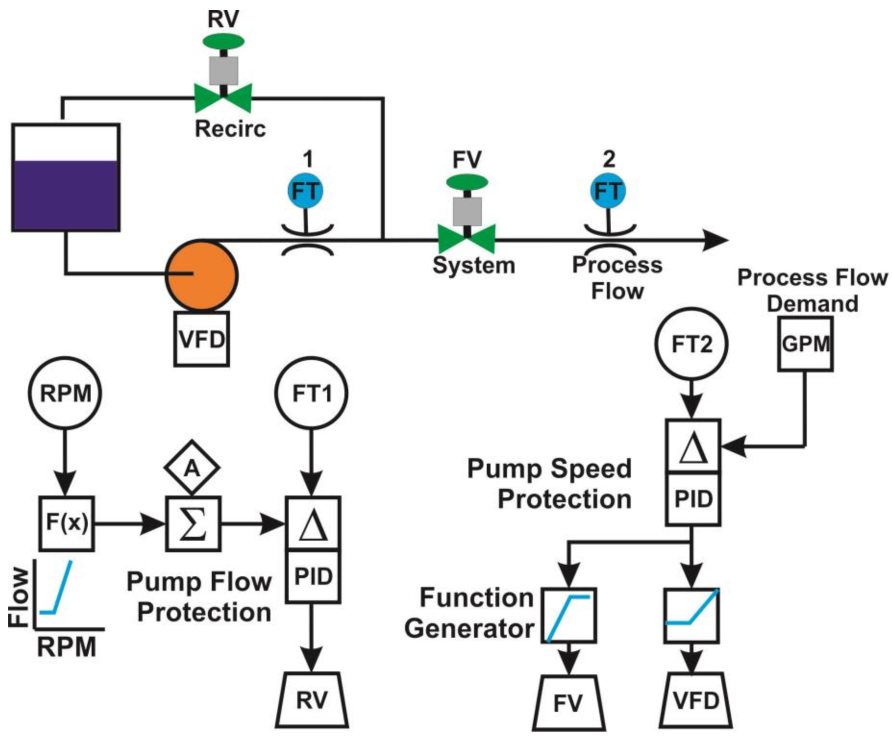
There may be some pump protection requirements at the very low speeds and the pump manufacturer should be consulted. Accordingly there may be a low RPM limit associated with the pump operation and as such recirculation protection must then be provided. It is expected that the low frequency/speed limit will be experienced by the motor or VFD. There may also be pump stability or other mechanical reasons to limit pump turndown.

2.2.3. VFD Pump Protection



When a limit is reached at the low end of the pump operating curve, we must accommodate the flow through the process system and protect the pump. In this case we will have a compound limit for the flow curve. The first corresponds to a turn-down or minimum pump head. The second limit is the minimum flow through the pump. From the speed-head chart we will use 20 feet of head as the minimum pressure across the pump and at that head 15 GPM is the pump flow protection. We can see that the system resistance will need to be modified with a flow valve to compensate for the pump's inability to control below 20 feet of head.

An example control arrangement can be created to provide the two-way protection for the pump. The arrangement suggested below is simplified and would require additional blockware to properly provide operational control.



In this arrangement the minimum flow would follow the RPM (or frequency)/flow chart down until it reached a minimum flow setting. At that setting the recirculation valve (RV) would begin modulating to maintain the minimum flow through the pump (FT1). The operator is provided with a BIAS in the control scheme to provide additional flow margin if necessary.

The second part of the arrangement (Pump Speed Protection) protects the pump or overcomes the minimum speed arrangement by leaving the VFD at a minimum setting and transfers the control to the process control valve. Alternatively the control swap could occur ahead of the PID control and a second controller would be provided to allow independent controllers for each the Flow Control Valve (FV) and VFD.

2.2.4. HP

Each application using VFD control should be examined on the actual or final design information over the expected pump operating range. Only then can the benefits of a VFD application over a Constant Speed be determined.

Motor Horsepower should be plotted over the expected flow ranges for both the VFD and Constant Speed approaches.

The equations to the right are used for water power and required mechanical power needed to drive the associated pumping system. For the centrifugal pumps, the efficiency changes over the flow rates through that pump. Each flow point through the pump must be individually calculated using the flow, head, specific gravity and efficiency. Once all flow data points are established for each, the VFD and Constant Speed application, the expected time and power differences at each flow point needs to be tabulated. Associated costs for the differences can then be used for determining the benefits of using the VFD motor controller.

Horsepower For Pump

Fluid Horsepower

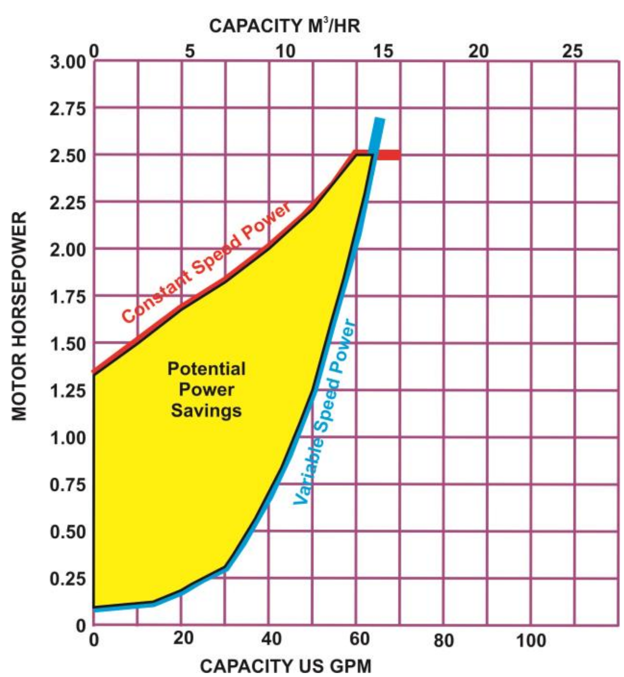
$$HP_{fluid} = \frac{Q\ H\ Sg}{3960}$$

Motor Horsepower

$$HP_{motor} = \frac{Q\ H\ Sg}{3960\ Eff}$$

Where:
Q = Volumetric Flow (GPM)
Sg = Specific Gravity
H = Developed Head (Ft)
Eff = Pump Efficiency
HP_{fluid} = Fluid Horsepower
HP_{motor} = Motor Horsepower

For the examples above, the Constant Speed and Variable Speed arrangements, the power curves have been calculated/estimated. Efficiencies below the stable operating region can only be estimated since operation in these regions is not recommended/allowed.



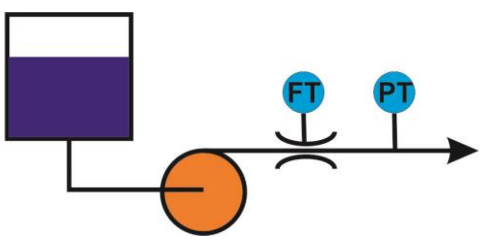
The power curves obtained for each speed arrangement were plotted on the same chart. The constant speed power curve shows higher power usage due to the higher static pressure. The variable speed power usage does indeed show that the system has the opportunity to save significant power. If the system normally runs at the 60+ GPM point, then the savings will be minimal. A detailed analysis of the costs and small benefits should then be carefully weighed. If the system operates for a significant period of time at 30 to 40 GPM, then larger savings can be expected.

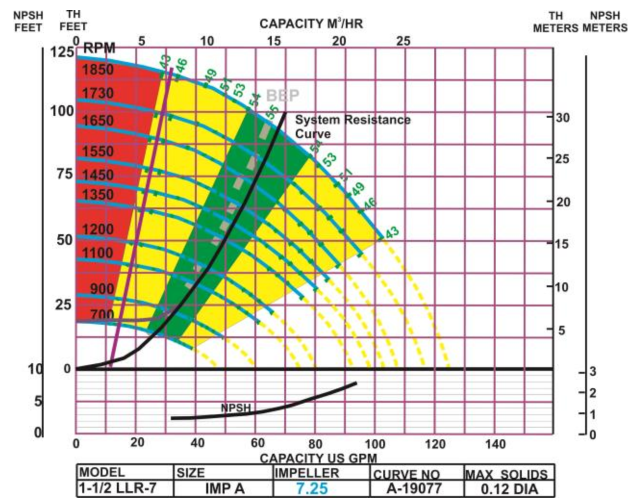
3. System Resistance Curves

3.1. Open Flow (No Static Head)

Many of the pumping applications are relatively simple. The source fluid supply head and destination fluid head are at relatively similar elevations. There is not a head offset to the flow curve that would require an adjustment to the resistance curve.

The free flow system resistance curve passes through the 0 Flow, 0 Head point on the Flow-Head chart. Under these conditions we have the ideal arrangement for a VFD speed controlled pump.



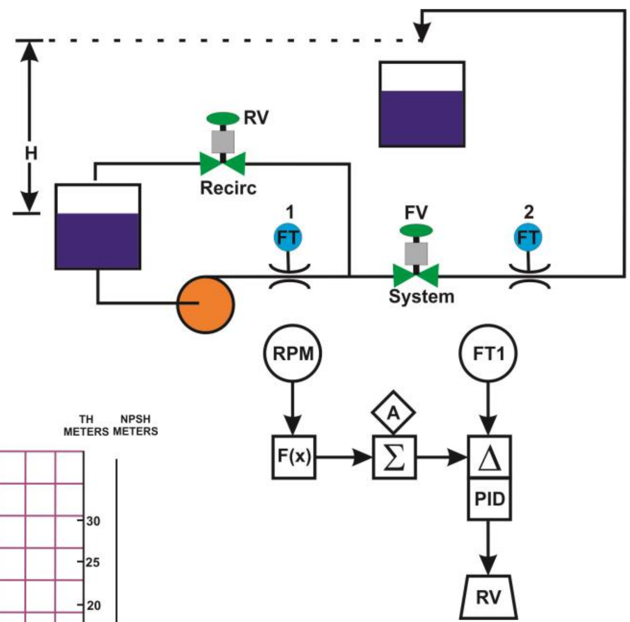


The previous examples up to this point have assumed this type of arrangement. The flow-head system resistance curves in all of the above systems pass through the 0 Flow, 0 Head point.

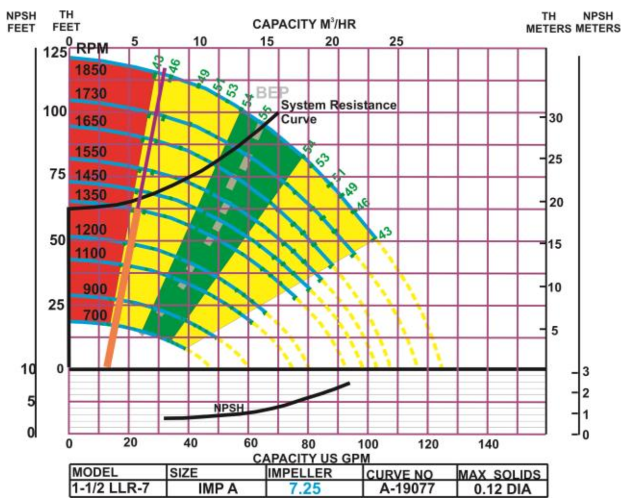
3.2. Header Flow (Existing Static Head)

The majority of applications do not pump from one level to the same level. They need to provide the fluid to a higher head discharge due either an elevation difference or the pump is supplying flow to a header in parallel with other sources. The flow from the pump cannot begin until the pump discharge pressure exceeds that of the destination discharge header. Elevation applications and common header applications can be addressed in a similar fashion recognizing the differences.

3.2.1. Elevated Head



The typical arrangement for an elevated head pumping system is shown on the right. The pump draws its suction from a reservoir at one level and transfers it to another level. In this case the system resistance diagram must be modified to reflect this difference.



The variable speed pumping curve on the left reflects the elevated system resistance curve. The pump is unable to move any water until it achieves approximately 60 feet of discharge head. We also note that several problems are observed.

The pump must be operating at nearly 1350 RPM before the discharge head matches the elevation difference and flow through the pump begins. During this rollup time the pump is operating in a deadheaded condition and it is important that recirculation is enabled and operating properly. The recirculation flow function generator [F(x)] must reflect the minimum required flow rate at each RPM to protect the pump during the pump rollup.

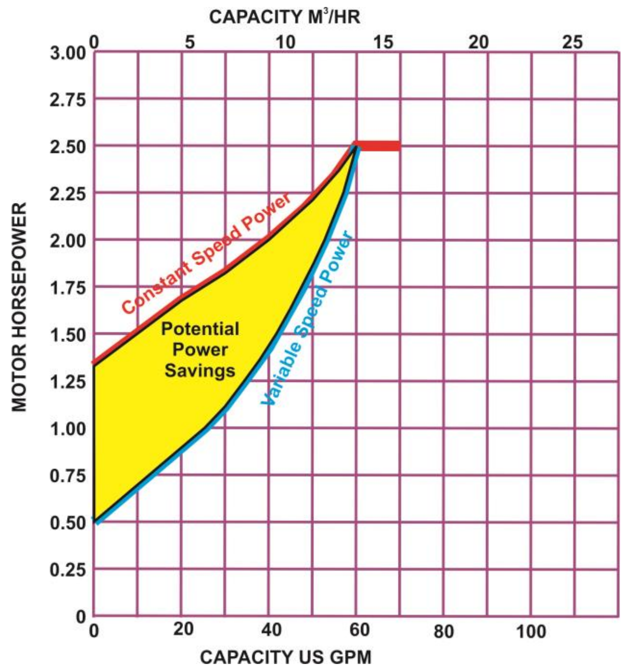
The second problem we should note is that the pump RPM range is now limited with respect to the desired flow range. Instead of flow being variable between 0 and 1850 RPM, in this instance it is only variable between 1350 and 1850 RPM. This may cause a flow resolution issue if the VFD does not have the appropriate resolution.

Alternatively if the destination tank is large enough, as in a water tower supplying a town, it may be more practical to cycle the pump rather than regulate the flow. During periods of low demand for the water, allow the pump to fill the tank until it reaches a full tank point and stop the pump. When the demand draws the tank down to a lower level, the pump restarts and allows the tank to refill. Multiple fill pumps would allow sequencing of the pumps to match the system demand. This eliminates the need for VFD control or pump recirculation with this open-flow, check-valve operation.

3.2.1.1. Horsepower

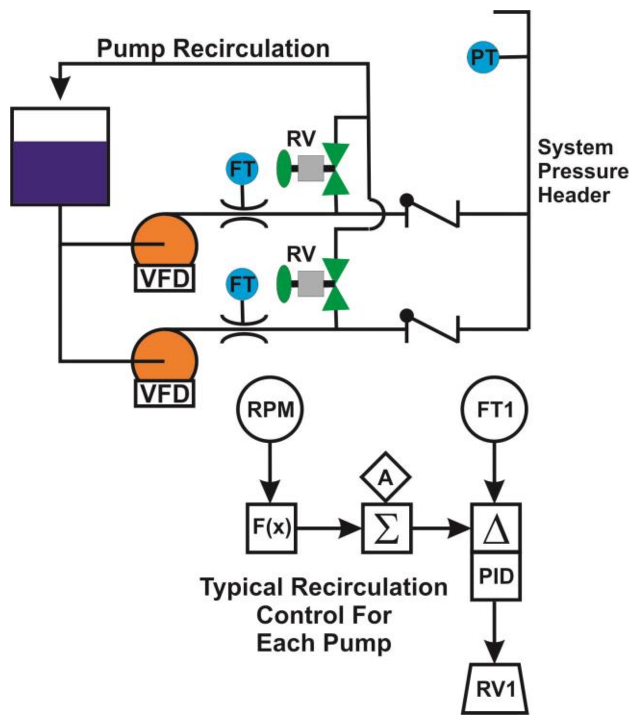
The constant speed power is compared to the variable speed power for this elevated head application. Note that the potential power savings are significantly less than a zero head application as shown in section 2.2.4 above.

Benefits for the application will be determined in the same manner as in section 2.2.4. The time at each flow and associated savings for each flow need to be tabulated and compared with the costs of this application.



3.2.2. Pressurized Header

A pressurized header application is very similar to the elevated application. Generally the only difference is that header applications typically have several pumps supplying the header.



In this case each pump will have its own pump protection. As each pump is started the VFD will spool up until the header pressure is reached. During this rollup time the pump needs recirculation protection.

Also during the operation of parallel pumps, each pump may not carry the same flow load. Small differences in pump speeds may lead to large differences in flow. These imbalances may require no recirculation flow on one pump while the other pump may require significant protection. For this application it may be necessary to provide a flow-balancing arrangement for the pump flow.