SUNDYNE

Centrifugal

Pump

Control Bulletin

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CENTRIFUGAL PUMP CONTROL BULLETIN

CONTENTS

Glossary

- By-pass or recycle flow - is the amount of process fluid taken from the pump discharge pipe that is returned to the suction source instead of being piped into the normal process stream. The bypass flow is figured into the sizing of the pump but is not delivered to the normal process flow. This is one method used to prevent a pump from operating below the specified minimum flow. Bypass flow returning to the suction vessel should be a small percentage of total volume to prevent temperature rise of process fluid.

- Control valve - The control valve is defined as the main forward flow control valve, not the minimum flow system recycle valve. A valve, typically pneumatically actuated, that varies resistance by throttling to achieve a desired flow rate.

- Head - In a pump it is the pressure rise through the pump in feet of liquid. In pipe elevation change it is the vertical distance between one location and another. Expressed in terms of ft (m).

- Hydraulic efficiency - is the measure of the hydraulic perfection of the pump. It is the ratio of the energy output to the energy input applied to the pump shaft.

- Minimum flow (Stable and Thermal)

  Minimum Continuous Stable Flow - lowest flow at which the pump can operate without exceeding vibration limits.

  Minimum Continuous Thermal Flow - lowest flow at which the pump can operate without its operation being impaired by the temperature rise of the pumped liquid.

- Pump curve or pump head flow curve - also known as the performance curve or characteristic curve, it is the amount of head a pump produces at any given flow rate. The curve shape is determined by the design of the hydraulics.

- Pump design flow - the pump flow rate at the pump's best efficiency point (BEP).

- Pump surging - sudden variations in pressure within a pump. System changes can cause damaging pressure surges in pumps.

- System curve - is the resistance of the system to passing fluid through it. The system is comprised of the piping, fittings, valves, vessels and instrumentation that transports and lies in the path of the pumped liquid. It is the sum of the static head losses and friction head losses (in feet (m) of liquid). The friction losses increase as the flow rate is increased, making the resistance head curve rise.

Scope

This bulletin is published to provide helpful suggestions for controlling centrifugal pumps between the minimum and maximum recommended flow points. Several process characteristics and control practices are discussed. However, these discussions should not be considered exhaustive since the means of controlling centrifugal pumps is at least as varied as system complexity, flexibility and cost. Many factors such as unknown pressure or temperature variations, chemical environment, or other changes could affect the recommendations shown here. The ultimate responsibility for specifying and operating the system lies with the purchaser.

1. Centrifugal Pump Control Theory

Operating capacity of a centrifugal pump is controlled by the intersection of the system head curve and the pump characteristic (performance) curve. Each system has its own unique head curve, expressed in feet of liquid being pumped, and comprised of static head and friction head. A centrifugal pump has its own particular head-flow performance curve as supplied by the pump manufacturer for clear, cold water. Correction of this curve is necessary only when fluid viscosity differs significantly from water.

Total system static head consists of elevation changes between suction and discharge liquid levels and includes any differences in pressure on the liquid surfaces. See Figure 1.

System friction head involves the head loss in the system caused by fluid flow through piping, valves, fittings, etc. These losses vary approximately as the square of the capacity for turbulent flow and directly with capacity for laminar flow. Frictional head loss depends on the size, length, type, surface condition, age of pipe and fittings and the particular fluid being pumped.

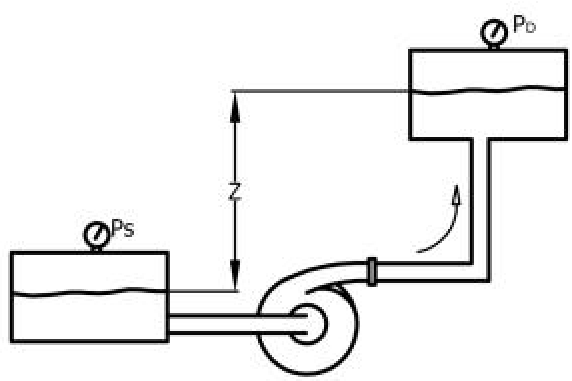

Figure 1. Total System Static Head. Z is Measured in Feet

Z = Elevation change from discharge to suction in feet
γ = Specific Gravity
P_D = Pressure in Discharge tank, PSI
P_S = Pressure in Suction tank, PSI

(US Units)

$$Total\_Static\_Head = Z + \frac{(P_D - P_S)\times 2.31}{g}$$

(Metric)

$$Total\_Static\_Head = Z + \frac{(P_D - P_S)\times .102}{g}$$

Combining the system static head and friction head produces the system head curve as shown in Figure 2. Systems in which there is a very high discharge pressure at low flow rates (such as boiler systems) are called reactive systems. Systems in which the pressure downstream of the pump goes to zero at zero flow rate (such as spray system) are termed resistive systems. Many are a combination of these two types.

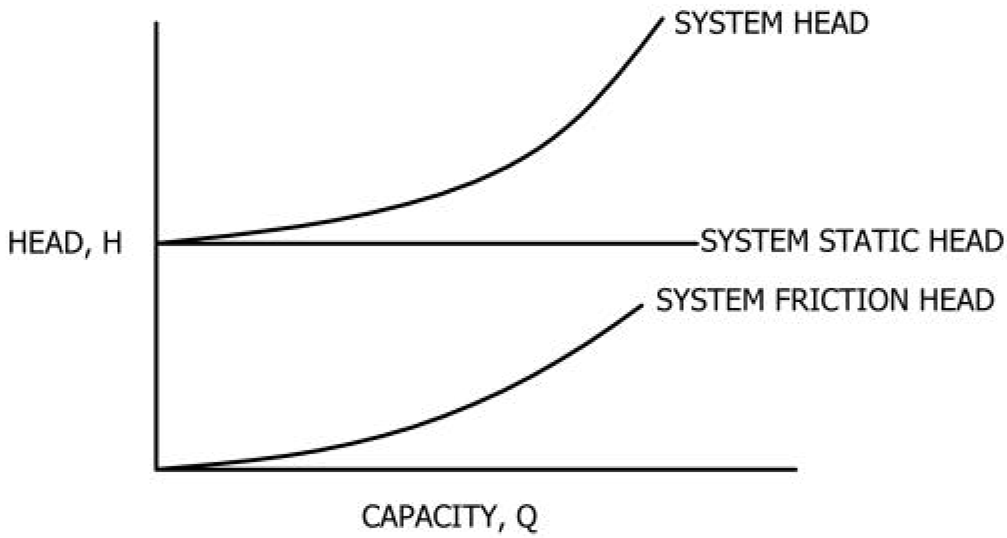

Figure 2. Total System Head = System Static Head + System Friction Head

The pump characteristic curve at constant speed is then superimposed on this system head curve (figure 3). Intersection of the two curves determines the flow rate delivered to the system by that particular pump. Q1 is the operating capacity in the example below.

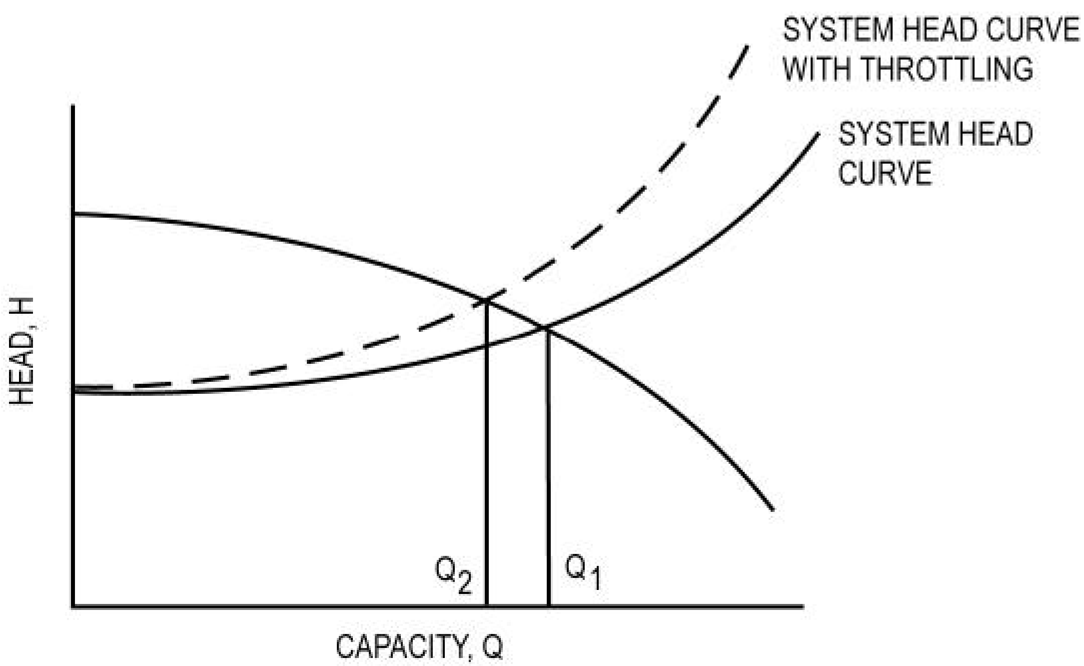

Figure 3. Pump Head Curve

Q1 : Operating capacity without throttling
Q2 : Operating capacity with throttling

In order to change the operating capacity, one or both of the curves must change shape.

A change of speed, an impeller trim, or total replacement of the impeller/diffuser combination can shift the pump head curve. The system head curve can be altered by modifications to the piping system, or by installation of a throttle valve downstream of the pump.

A throttle control valve is the most practical solution for varying flow requirements, and its effect is shown in Figure 3. The throttle valve has the effect of moving the system curve. If the throttle valve is wide open, the system curve and pump curve would intersect at flow point Q1: if the valve is closed slightly, the system curve and pump curve intersection point will move to flow Q2. A new system head curve and operating capacity are obtained at each valve setting, but it is important to realize that the head lost across the throttle valve is wasted energy.

Total head developed by a centrifugal pump typically increases with each successive decrease in flow (control valve setting) until shutoff is reached. *However, a centrifugal pump operating at shutoff will overheat and may experience severe damage.*

There can be times when the system and pump curves do not intersect (Figure 4). In this instance, the pump is likely to be operating below its minimum flow point and may experience bearing damage or shaft breakage. A bypass control system is required to keep the pump operating on its curve. The amount of flow that must be bypassed is system dependent.

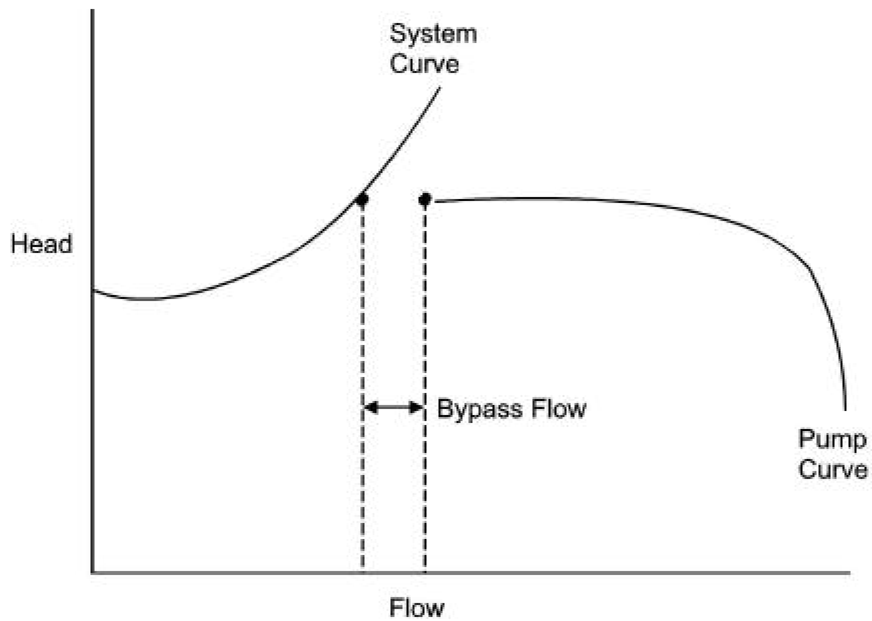

Figure 4. The Effect of a Low Flow System with the Pump Operating Below its Minimum Flow Requirement

An adequate margin should be provided between the system and the pump curves so that a steep angle of intersection exists at any operating point (Figure 5, Pt.. B).

It is important that a flat angle of Intersection between the system and pump curves be avoided (point A). A steep angle of Intersection (point B) will prevent flow "hunting".

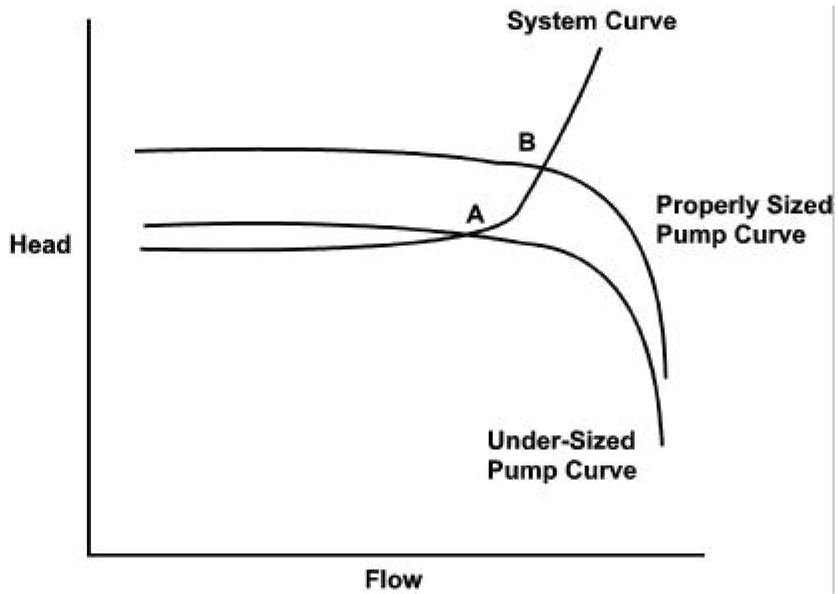

Figure 5. Intersection of Two Pump Curves with the System Curve. Operating Point B with the Steep Intersection between Pump and System Curves will Provide Stable Operation

Both the system head and pump characteristic curve shift gradually over a period of time. The system head curve, partially comprised of friction head, changes because the pipe deteriorates and frictional losses increase. For example, friction losses in a pipe increase roughly sixty percent in fifteen years. Centrifugal pump performance deteriorates with time because internal running clearances between stationary and rotating parts increase due to wear. The rate of wear depends on the type of pump, severity of service, process fluid, materials, differential head produced by the pump and various other factors. Wear is typically not a factor in Sundyne pumps due to their open running clearances. The overall effect of pipe deterioration and pump wear (when applicable) is a reduction in the operating capacity.

To account for these factors, pump users specify design conditions with a safety margin in capacity and total head. Consequently, the pump is oversized for the true operating conditions, and the system must be throttled back to the desired capacity. Throttling represents wasted energy; so considerable power savings can be realized by proper sizing of the pump and system.

Another critical design parameter usually specified with a safety margin is the Net Positive Suction Head (NPSH) available. In the majority of pump applications, overall pump efficiency must be sacrificed in order to meet conservatively low NPSH requirements. At this lower efficiency, driver size can be affected in certain borderline situations. The net result is increased power consumption and possibly higher initial cost for the pump. Again, pump users can realize power saving benefits by specifying design conditions without excessive margins.

2. Minimum Flow

Historically, pump manufacturers have been urged to oversimplify their published minimum flow limits. Often a simple percentage of the design flow of a unit has been used, irrespective of the system in which it is installed. This conservatism limits the operation to a flow that is well above the potential of the pump in certain systems.

There are two minimum flow phenomena of concern. One occurs at start up when the pump is filling the pipeline, while the other occurs at low flow as the flow control valve is closed or system resistance is increased.

Start Up

When the volume of pipe from the pump discharge to the main control valve is filled, a shock wave can develop which transmits energy back upstream into the pump. The energy of the wave is a function of the pump pressure rise (head), volume of liquid, and specific gravity. As the wave enters the discharge of the pump, the shaft is "shocked" causing potential damage to it and the bearings. Impeller blade failure can occur as well.

To minimize or eliminate "water hammer", a recycle valve should be opened during start up to minimize damage to the pump in the event that the main flow control valve fails shut or is closed too quickly during normal operation.

Low Flow

Per API, there are two definitions related to the pump minimum flow.

a. Minimum Continuous Stable Flow (lowest flow at which the pump can operate without exceeding vibration limits).

b. Minimum Continuous Thermal Flow (lowest flow at which the pump can operate without its operation being impaired by the temperature rise of the pumped liquid).

Pump operations below these points can cause shaft vibration and reduce the mechanical seal life. These problems can be aggravated with high-energy pumps.

Factors affecting minimum flow:

The following pump and system design factors affect minimum flow and will be covered in detail:
2.1 Temperature rise within the pump
2.2 Hydraulic design of the pump
2.3 System surge
2.4 Control valve location
2.5 System stiffness

2.1. Temperature Rise within a Pump

When operated away from its design flow or best efficiency point (BEP), inefficient operation results in heat being added to the fluid as it travels through the pump. If it becomes heated to the point of boiling (which is related to suction pressure) a vapor pocket will form, causing it to "surge" or "vapor lock". The temperature rise through a pump can be approximated by the equation:

(US Units)

$$\Delta T(F) = \frac{HEAD}{778 \times Sp\_Heat} \times \left(\frac{1}{h} - 1 \right)$$

(Metric)

$$\Delta T(C) = \frac{HEAD}{10.4 \times Sp\_Heat} \times \left(\frac{1}{h} - 1 \right)$$

Where head ft (m) and efficiency are at the flow rate being examined.
Values of the Specific Heat of common fluids are provided in Table 1.

Acedic Acid	0.51 (2.14)	Gasoline	0.50 (2.09)
Acetone	0.54 (2.26)	Kerosene	0.50 (2.09)
Alcohol (absolute)	0.58 (2.43)	Machine Oil	0.40 (1.67)
Butane	0.56 (2.34)	Naptha	0.31 (1.30)
Ethylene Glycol	0.60 (2.51)	Propane	0.59 (2.47)
Fuel Oil	0.56 (2.34)	Water	1.0 (4.19)
Freon 12	0.24 (1.00)		

Table 1. Mean Specific Heats BTU/lb°F (kJ/kg°K) of Common Liquids between 32 and 212 °F (0 and 100°C)

Hydraulic Efficiency Calculation:

a. When determining the temperature rise at a low flow rate where there are no published efficiency lines, the efficiency may be scaled down in direct proportion with the flow reading at the last published efficiency line.

   For example: If the efficiency at 40 GPM is 50%, the approximate efficiency at 20 GPM would be 20/40x.50=25%.

b. For integral gear driven machines the mechanical losses in the gearbox should be subtracted from the overall horsepower (BHP) to determine hydraulic horsepower.

   1. Check manufacturer's data for mechanical horsepower loss in the gearbox.

   2. Subtract the mechanical horsepower loss from the overall horsepower BHP (BKW) from the field selection procedure to determine the hydraulic horsepower HP (kW).

   3. Calculate the hydraulic efficiency:

(US Units)

$$h = \frac{Q \times H \times g}{3960 \times HP}$$

(Metric)

$$h = \frac{Q \times H \times g}{367 \times KW}$$

?	=	Hydraulic Efficiency
Q	=	Flow, GPM (m^3/hr)
H	=	Head, Feet (m)
γ	=	Specific Gravity
HP	=	Hydraulic Horsepower
kW	=	Hydraulic kW

2.2. Hydraulic Design of the Pump

Hydraulic design of the pump is a major consideration in determining pump minimum flow. The pump manufacturer's data sheets should provide pump minimum flow requirements. Minimum flow requirements for Sundyne and Sunflo pump models are included in sections 10 and 11 of this document.

2.3. System Surge

Fluctuations in head capacity (flow surging) may occur at low flow. This phenomenon can easily be visualized using a simple spring-mass system. The fluid in the system represents the mass that must be capable of moving freely. This condition exists whenever the suction and discharge tanks have free surfaces, as on boiler feed service. Any energy storage device within the system provides the necessary spring. An example would be long piping or a tank with a vapor or air cushion. Finally, a periodic excitation force is required to start and maintain oscillation of the fluid mass. These hydraulic forces at distinct frequencies usually originate within a centrifugal pump during low flow operation, or as a reaction by the pump to the system itself.

One method used to avoid flow surging problems is proper location of the discharge throttle valve. Placement of the valve close to the pump discharge flange minimizes the amplitude, and thus the effects of the flow oscillations. The mass of the oscillating fluid is reduced in volume, and the turbulent flow through the valve destroys the frequency of the excitation force.

If the throttle valve is remotely located, flow surging will be of low frequency and high amplitude since the fluid mass is large. This situation must be avoided, as it may induce violent mechanical vibrations in the piping system. Consequently, the preferred throttle valve location will always be close to the pump discharge flange in order to minimize the potential and the effects of flow surging. Minimum flow is a complex function of valve location; pump horsepower and user's piping system dynamics.

Flow surging problems can also be resolved by installing a bypass line. Bypassing a portion of the pump capacity back to suction maintains pump operation closer to its design flow where the amplitude of the hydraulic excitation forces is small. The bypass line also protects the pump from overheating and damage if system flow rate is reduced below minimum flow.

2.4. Control Valve Location

The system component that is the most influential in causing low flow pump surging is the location of the pump control valve. When a centrifugal pump is operated at a very low flow rate, recirculation occurs within the impeller, and it surges at the natural frequency of the system. As the control valve is moved away from the pump, there is a decrease in the frequency and an increase in the amplitude of the pressure waves (Figure 6 A & B).

The energy imparted to the system by the pump is similar to the strumming of a guitar string. The frequency is a function of the length of the string, and is analogous to the distance from the pump to its control valve. The greater the distance between the valve and pump, the lower the frequency of this oscillation (and the greater the magnitude of the pressure pulsations). This can eventually lead to bearing failure and/or shaft breakage.

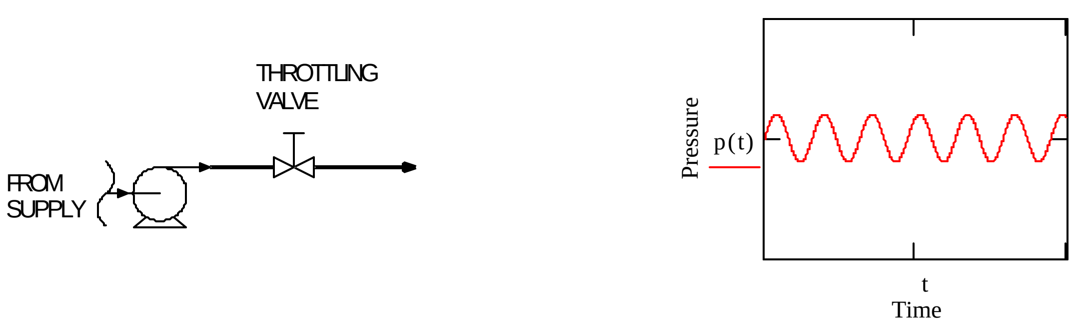

Figure 6A. Pump Surging (Lowest Amplitude) with a Control Valve Close to the Discharge

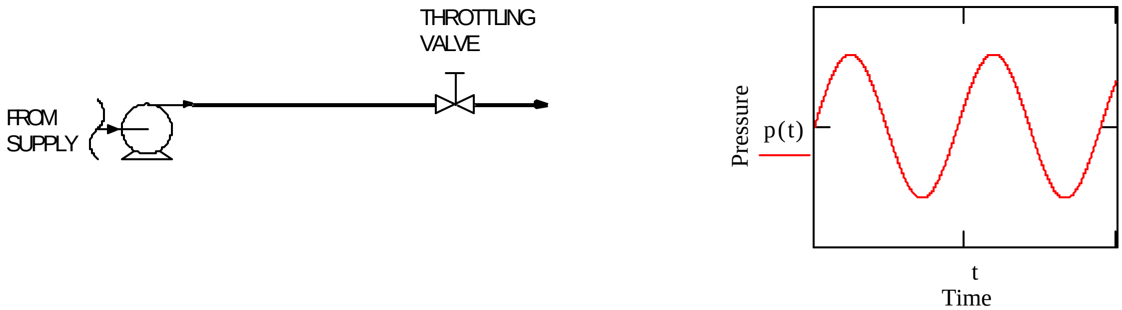

Figure 6B. Pump Surging (Higher Amplitude) with a Control Valve Far from the Pump Discharge

2.5. System Stiffness

The stiffness of the pumping system affects both the frequency and amplitude of pump surging. A “soft” system can result in low frequency high amplitude surging that is detrimental to pump performance and life. A "soft" system can be caused by a number of factors: There can be air or gas entrained in the pumping fluid, air can be trapped in an elbow, or a system can have open tanks or an accumulator. A closed loop boiler with a remotely located control valve that takes only a minimal pressure drop at low flow rates, or a system in which there were flexible lines attached to the suction and discharge of the pump, would be typical of a soft system.

The following rules should be applied to minimize the effects of soft systems:

a. Place the control valve at or within five feet of the discharge of the pump.

b. Use flexible discharge lines only downstream of the control valve and use a minimum of 10 pipe diameters of straight piping into the pump suction.

c. Size pump control valves so that 5-8% of the differential pump head is taken across the valve.

d. Install a low flow bypass system if operation below minimum flow is anticipated (Figures 5 A, B, & C).

e. Eliminate entrained air and gasses or anticipate a higher minimum flow limit if they cannot be avoided.

f. Avoid air traps in pumping systems.

g. Consult factory if operation is anticipated below 40% of the BEP for any sustained period and the NPSH is less than 30 feet.

In hydraulically soft processes where a capacity control valve cannot be located near the pump, and maximum turndown is needed, a discharge orifice is often recommended. This isolates the pump from potential system resonances, and permits stable operation at flow rates down to 15-20% of BEP (the pump's best efficiency point). The orifice is normally sized to drop 5% of the pump head at BEP.

3. Minimum Flow Control

Typical low flow bypass systems are illustrated below, starting with the most simple.

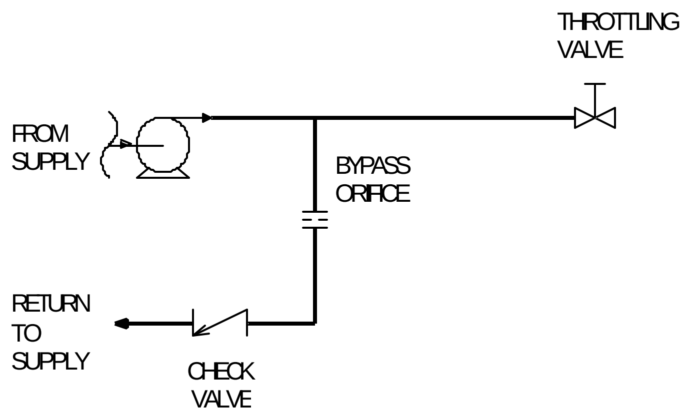

Figure 7A. Continuous Minimum Flow Bypass Loop

Systems that operate at or below the minimum flow point should have a continuous minimum flow bypass loop. This is the least expensive control system, but the pump must be slightly oversized and will draw additional power due to the continuous bypass flow.

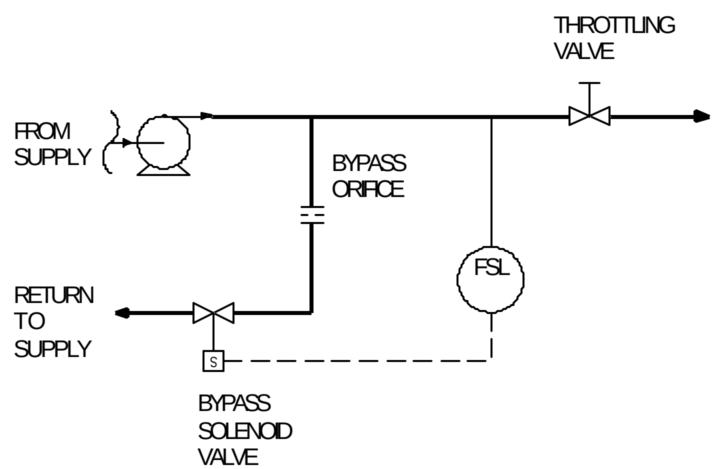

Figure 7B. Flow Switch Minimum Flow Control Loop

Minimum flow may be controlled by using a low flow switch that activates a solenoid valve. Initial cost of this system is slightly more expensive than a constant bypass flow control, but the power consumption is reduced because the pump does not need to be oversized to accommodate the bypass flow.

4. Maximum Flow

Centrifugal pumps can be damaged if operated at excessively high flow rates, even though this is not as harmful as operation below the minimum flow point. Pump operation should be limited so that the discharge pressure is never more than 10% below the design point. Operation at too high a flow rate can result in cavitation damage in the discharge of the pump, seal chattering, and leakage.

5. Parallel Pump Theory and Operation

5.1 Parallel Pump Theory

A typical parallel pump schematic is shown in Figure 8, in which two pumps share a common suction and pump into a commo n discharge system.

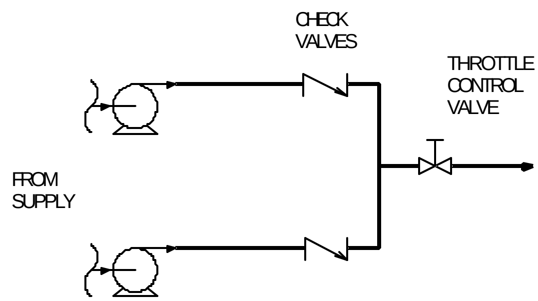

Figure 8. Two Pumps Connected for Parallel Operation

The first discussion will consider pumps with identical head-flow curves operating in parallel and each pump sharing the load equally.

The overall characteristic curv e for pumps (similar or dissimilar) shown in figure 9 in parallel configuration is obtained by adding the capacity of each pump at a constant head. For example, at H = H1, pump A and B each deliver flow rate Q1.

In parallel operation, delivered flow will be Q2 = Q1 + Q1.

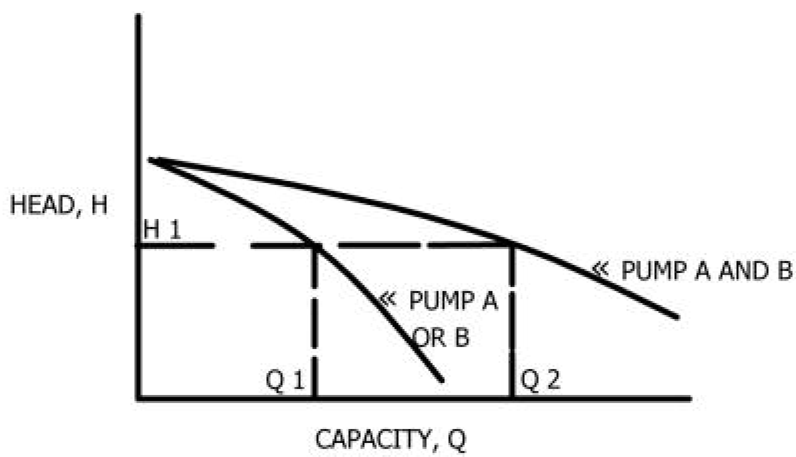

Figure 9. Pumps A and B in Parallel at a Constant Head

Operating capacity for the two pumps in parallel is again determined by the intersection of the system head curve and the combined pump characteristic curve (see Figure 10). Note that using two pumps in parallel doesn't necessarily double the capacity delivered to the system. It depends on the shape of the system head curve.

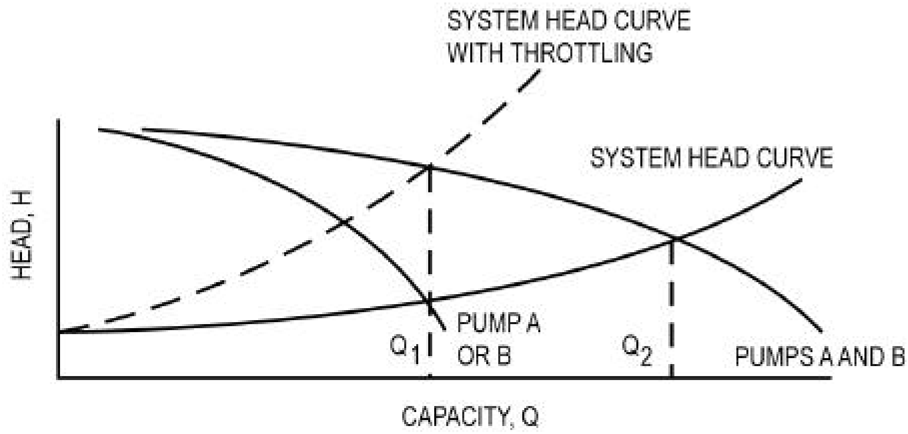

Figure 10. Two Pumps Connected in Parallel with Changing System Head Curves

If system demand decreases from Q2 to Q1 or lower, a decision must be made whether to operate just one pump or both pumps at low flow via throttling. Operation of the single pump has several advantages including higher efficiency and considerably longer life. Single pump operation near its best efficiency consumes less input power than running both pumps at low flow. In addition, time is available for maintenance and repair of the pump out of service.

The ability to operate just one pump in a parallel system requires extra precautions during the selection process. Note that the single pump will usually operate at flow rates higher than its design value. Therefore, driver size and NPSH requirements must be accounted for at this operating condition.

The following discussion considers pumps with different head-flow curves.
Load-sharing problems can arise during low flow operation. To illustrate this problem, individual H-Q curves of two dissimilar pumps and their combined curve are shown below for a parallel system.

Operating capacity for the system head curve shown is Q1 in figure 11. To determine flow rate through each pump, a constant head line is projected back to the individual pump curves. In this example, the capacities delivered by Pump A and B is Q2 and Q3, respectively.

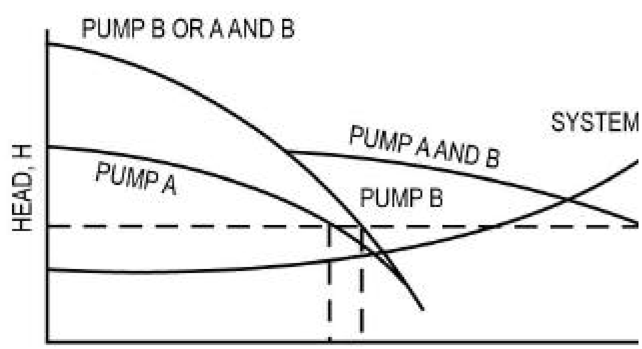

Figure 11. Flow through Parallel Pumps with Unequal Head Curves

As flow demand in this system progressively decreases, discharge throttling produces new system head curves. Pump B gradually supplies a larger percentage of the overall system flow and the load division between the two pumps becomes seriously unbalanced. The system head curve shown in figure 12 represents an extreme low flow condition for these two pumps.

Operating capacity for this new condition will be Q2, delivered entirely by Pump B. Pump A has been backed off the system and is operating below its minimum flow since system head is greater than the head at minimum flow of Pump A. This condition is very dangerous to the pump because, as mentioned earlier, the pump will overheat and suffer internal damage.

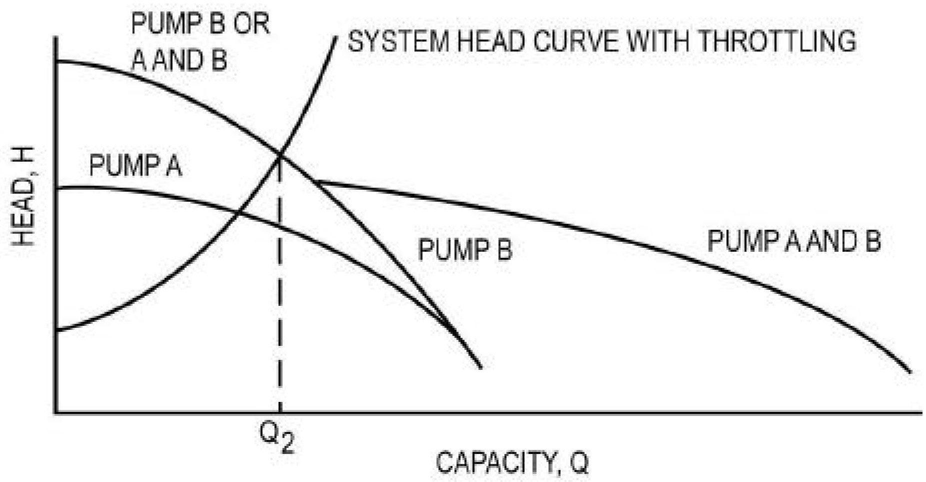

Figure 12. Extreme Low Flow Condition for Two Dissimilar Pumps in Parallel

In summary, for pumps to run successfully in parallel:

a. Pumps should have closely matched head flow curves within the expected operating range. Some deviation is acceptable and, as a general guideline, the maximum head difference between pumps should be roughly 2% at any flow condition.

b. The system head must never exceed head at minimum flow of any pump throughout the operating range.

c. The preferred piping arrangement for parallel systems is uniform between the various legs. This includes both the suction and discharge lines of each pump.

5.2. Parallel Pump Operation

Two pumps operating in parallel will never have exactly the same performance curves (Figure 13). At the design point with the valve open, both pumps are operating at the same flow and head. However, as the system curve changes with the valve positioning, pump A can produce a higher head than the pump B. In this case pump B will stop flowing altogether, causing it to operate in an unstable minimum flow condition.

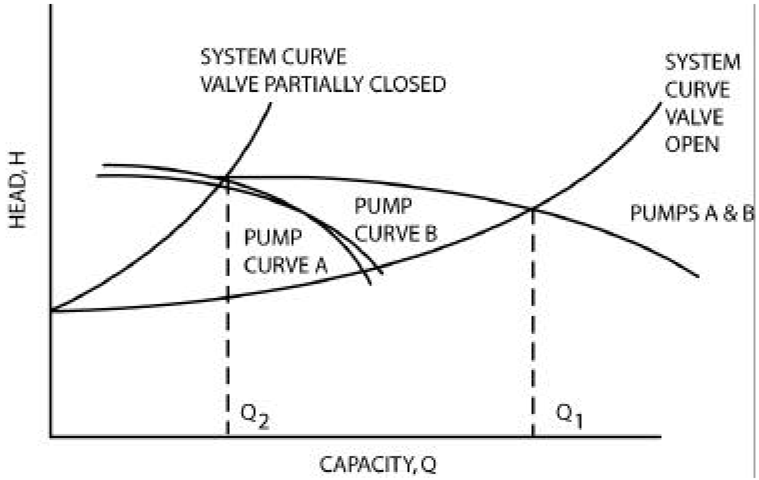

Figure 13. Pumps Operating Well in Parallel together with the System Valve in Open Position. At a Lower Flow Pump B is Unstable Due to Mismatched Curves.

Caution must be taken to avoid operating at a low enough flow rate that pump A can "overpressure" pump B, dead-heading it.

The following steps can be taken to prevent this from occurring:

a. Install trim valves on the discharge of each pump. Set them during operation to ensure that there is an equal power draw between the pumps at the normal operating flow rates.

b. Install a bypass line, if needed, to prevent operation below the minimum recommended flow. Figure 14 shows a simple fixed orifice constant bypass line on each pump and a manually operated throttle control valve.

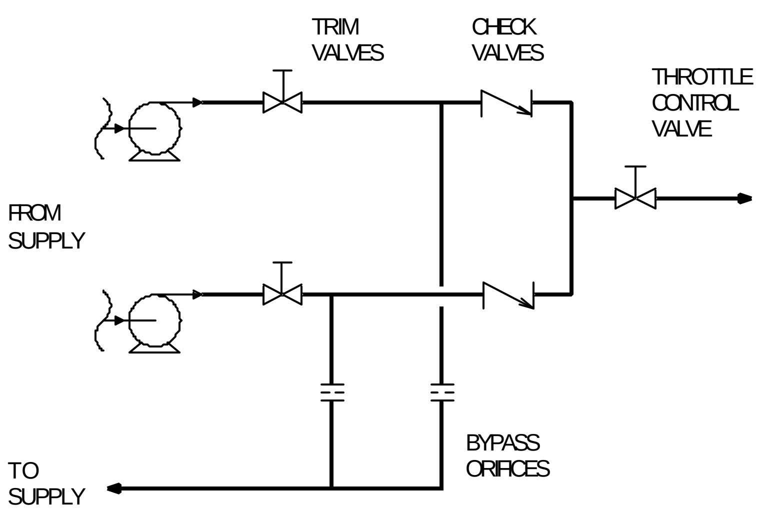

Figure 14. The Most Simple and Economical System Control Incorporating a Throttle Control Valve and Constant Minimum Flow Bypass Lines

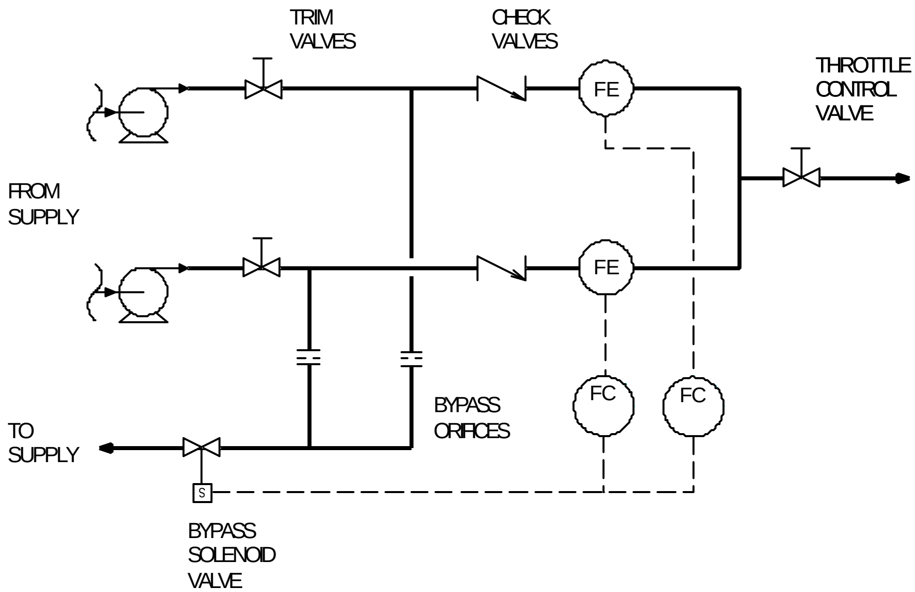

Figure 15. Flow Switches that Control the Solenoid Bypass Loop can be used to Automate Bypass Flow

Figure 15 shows the addition of a flow sensing element (FE) in each pump discharge line controlling a solenoid operated bypass control valve. Either flow switches or line current relays are used to actuate the bypass valve. A single flow element can be used if the two pumps are properly "balanced" using the individual trim valves. The motor current draw is used to balance the pumps during normal operation. The trim valves are then locked so they are tamper-proof. As the flow rate reaches the minimum flow required, the solenoid valve is opened to return some fluid to the supply reservoir.

6. Series Pump Operation

Series operation of pumps is often used when required total head exceeds that which can be supplied by a single pump. A typical arrangement for two pumps in series is shown in figure 16.

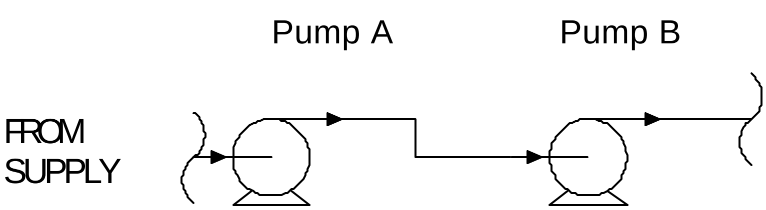

Figure 16. Pumps Arranged in Series

The overall performance curve for pumps (similar or dissimilar) in series is obtained by adding the head developed by each pump at the same capacity (figure 17). For example, at Q = Q1, pumps A and B each develop a total head of H1. In series operation, total developed head will be H2 = H1 +H1.

The intersection point of the system head curve and the combined characteristic pump curve determines the capacity delivered to the system. In figure 18 the operating capacity will be Q1 at a head of H1.

Note in this series pump example that single pump operation is not acceptable because the system head curve never intersects the characteristic curve of the single pump. If attempted, capacity would be zero and the pump in operation overheats.

When applying pumps in series, flange ratings, seal cavity environment, thrust bearing capacity and allowable case working pressure must be carefully reviewed for all operating conditions. These items are especially critical for the second and follow-on pumps in a series installation, since the suction pressure of each is dependent on the discharge of the preceding pump.

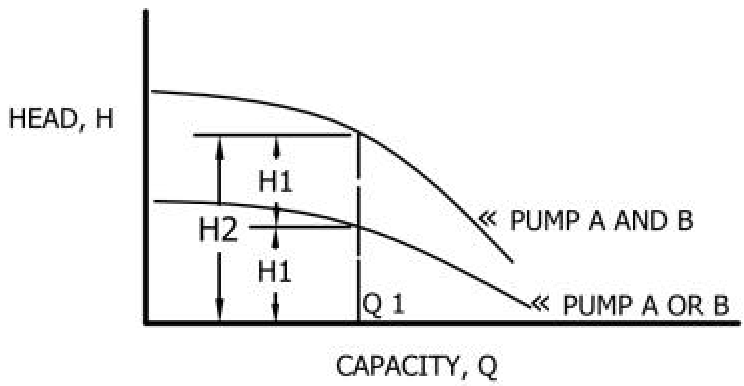

Figure 17. Pumps in Series at Constant Flow

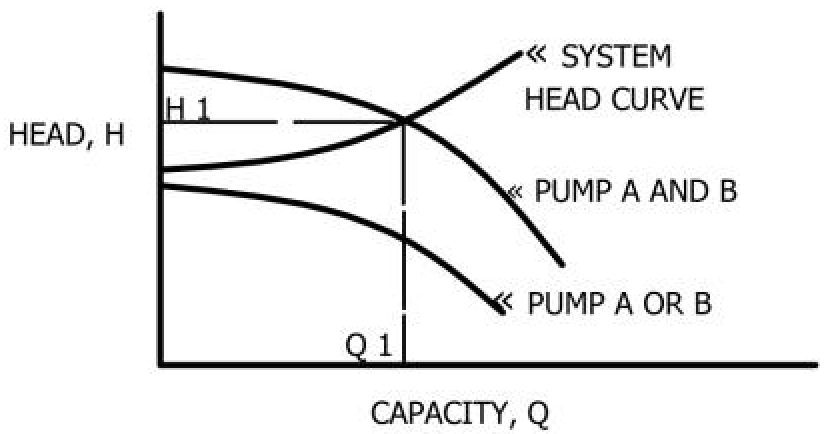

Figure 18. Pumps in Series Superimposed on the System Head Curve

7. Pumps with Non-Rising Head Curves

On certain low specific speed pumps, the characteristic head-flow curve may not rise continuously to shutoff. In other words, peak head is greater than shutoff head, as shown in figure 19.

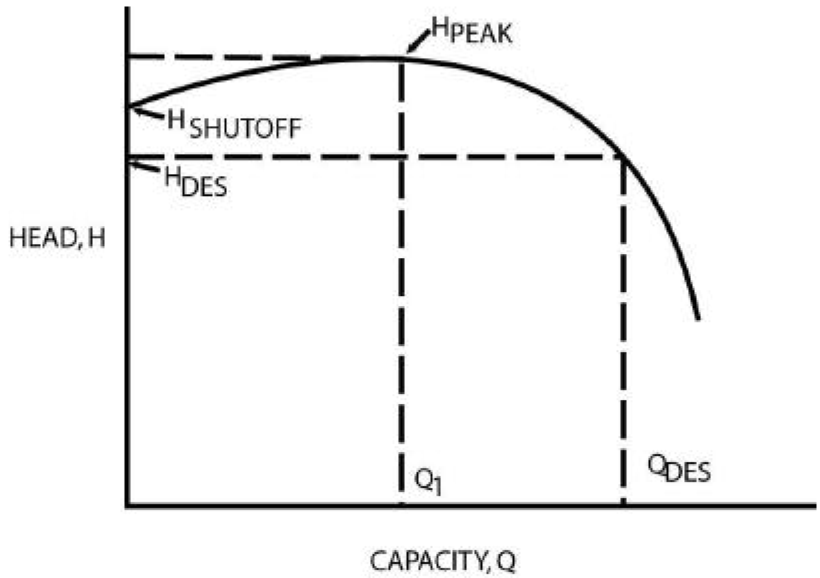

Figure 19. Non-Rising Pump Curve

NOTE: Q1, flow rate at the peak of the curve, as a percentage of the design flow rate (Q des), varies depending on several pump design parameters.

In most installations, this type of pump usually presents no operating problems. Consider the system head curves in Figure 20 with each curve representing a different throttle valve position.

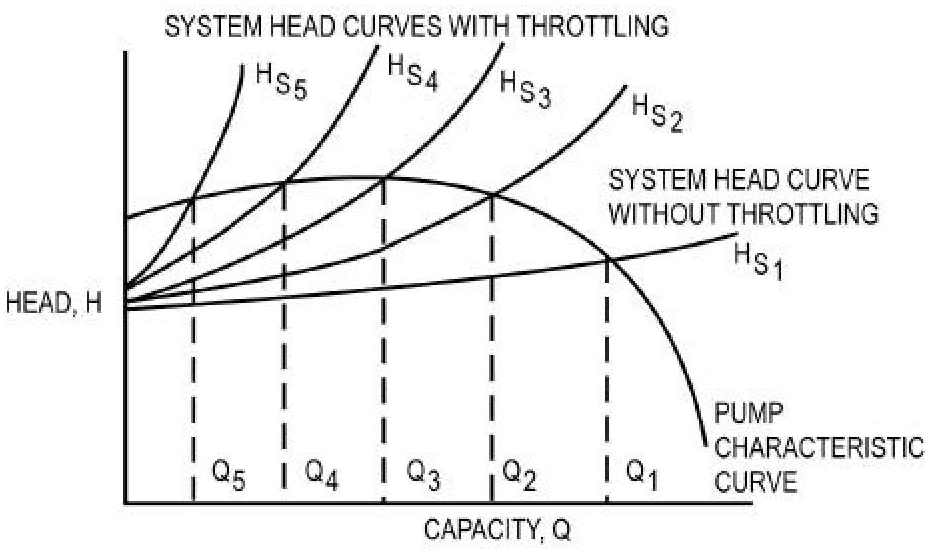

Figure 20. Pump Control at 5 Throttle Valve Settings

Notice that the system operating capacity is clearly defined for each throttle position (i.e. the intersection point of the system head curve and the pump H-Q curve). Control of this pump system will be smooth and predictable in most installations.

In some limited applications, the system head curve may not intersect the pump head-flow curve at a discrete point during low flow operation. In this case, operating capacity may drift even at a constant control valve setting, as illustrated in figure 21. This situation is rare and usually indicates an undersized pump for the installation, because the pump shutoff head is less than system static head

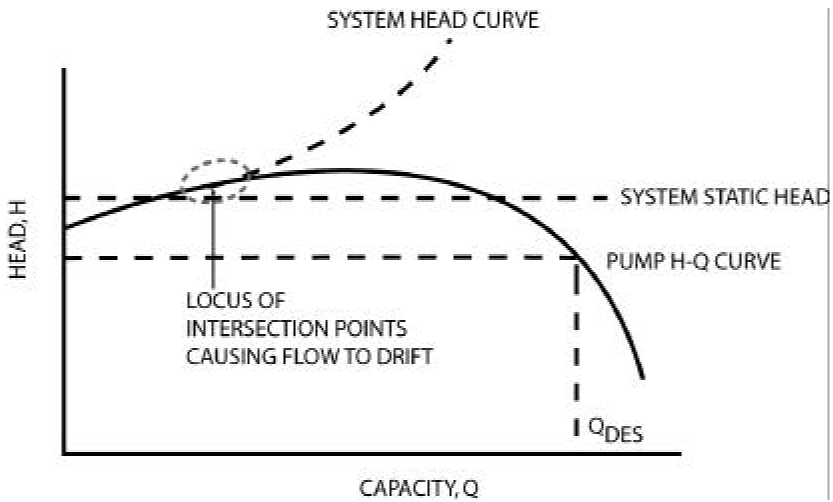

Figure 21. Low Flow Instability

Parallel operation of pumps with non-rising head curves may also present load-sharing problems, as previously discussed, even if they have identical characteristic curves. Consider the two similar pumps in parallel in the following example (figure 22):

If the system operating capacity is Q1 or higher, the load will be divided equally between the two pumps. If system demand decreases to Q2, the load could be equally divided, or one pump may overpower the other. For instance, Pump A could deliver Q_A and Pump B could deliver Q_B (the sum total for the two pumps being the required capacity, Q2). Further reduction in flow demand may eventually back one unit off the system entirely, forcing this unit to run below its minimum flow. This example demonstrates the importance of maintaining system head below peak head of any pump in a parallel system.

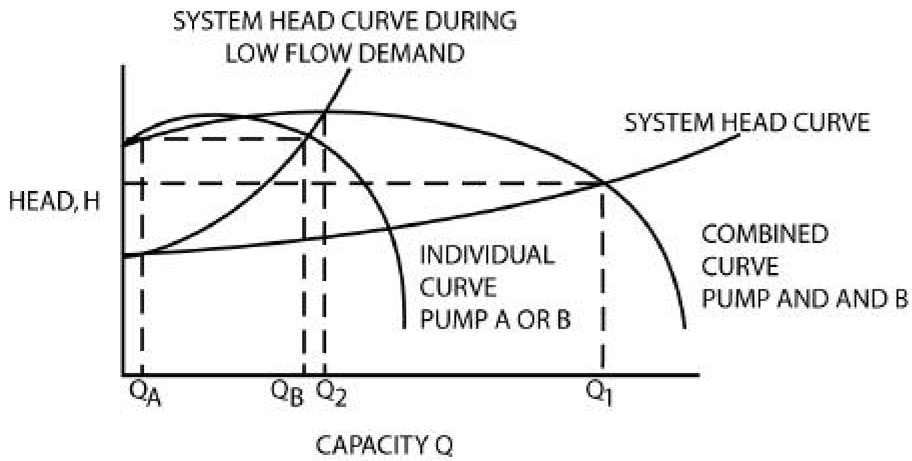

Figure 22. Pumps in Parallel Load Sharing

Load sharing problems on this type of pump can be presented during low flow operation. A flow control system will eliminate the possibility of one unit taking the entire load and shutting off the other. Alternatively, a bypass line installed on each pump will keep it from running at shutoff. The bypass should be a modulating system using flow control, rather than pressure control (figure 23). Either method is strongly recommended if pumps with non-rising head curves are used in a parallel system.

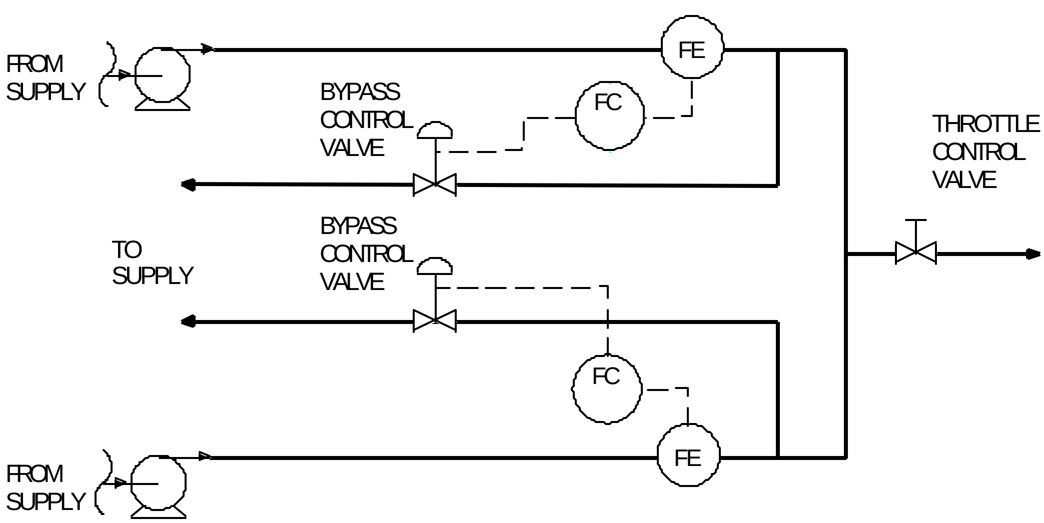

Figure 23. Flow Control Modulating Bypass System

8. Variable Speed Operation

Variable speed operation is being more frequently used to allow for a wider range of pump control. The pump is typically sized with an operating characteristic curve and speed to place the best efficiency point (BEP) near the system rated flow point.

A variable frequency drive (VFD) can be used to control the pump speed. As the speed changes, the pump BEP moves in a parabolic relationship. The flow changes proportionally with the speed while the head changes with the square of the speed. In figure 24 the pump characteristic curve at various speeds is shown intersecting with the system curve.

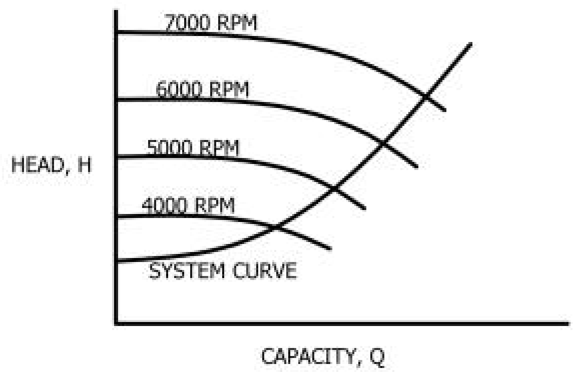

Figure 24. Pump Characteristics at Variable Speeds

As the system flow demand changes, the VFD output frequency and motor speed is either increased or decreased to move up or down the system curve. Since the system curve will rarely match the parabolic movement of the pump curve, it may be necessary to combine throttling with the speed change for optimum performance.

As with fixed speed pumps, the speed of a VFD controlled pump should never be reduced such that the pump characteristic curve is below the system curve. Also, with gear driven pumps with integral shaft drive lube oil pumps it is necessary to maintain adequate speed to maintain sufficient lube oil pressure and circulation unless a continuously running external lube pump is utilized.

If the system curve is relatively flat, as shown in figure 25 there is little benefit in using a variable speed centrifugal pump.

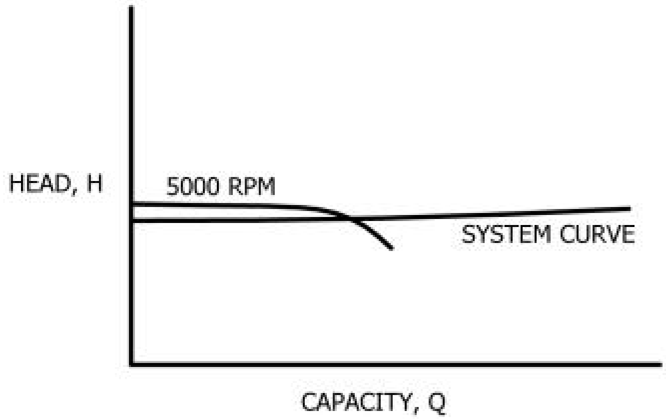

Figure 25. Flat System Curve

9. Pump Protection

Every centrifugal pump has defined minimum and maximum flow limits, and operating a pump outside of these limits reduces the pump life expectancy. The key to protecting a pump is to minimize the amount of time that a pump spends outside its flow limits, so a pump-monitoring device is crucial. One of the most cost effective monitoring methods is to use the pump motor itself. When a pump operates outside its flow limits, the pump motor power consumption also changes:

Flow Condition	Power Consumption	Possible reasons for flow change
Low Flow	Decreases	Dry running, clogged filter, closed valve/dead head, Decoupling (Magnetic Drive Pumps)
High Flow	Increases	Severe cavitation, ruptured discharge line, end of curve operation

A motor load-sensing device can be installed on a pump system to signal an operator or an automatic controller when a pump is operating outside the designed limits.

Types of Motor Load Sensing:

Two methods are available to determine AC motor load:

- Motor current (in Amps)
- Motor input power (in kilowatt or horsepower)

Power Monitoring has distinct advantages over common current monitoring, also known as amperage monitoring. Current is almost constant up to 50% of the motor load range. Because of this, it is very difficult to detect changes below 50%. In contrast, because the input power varies linearly across the entire motor load range, it is an extremely reliable and accurate detector of system changes. As such, most pumping condition changes will be seen in input power fluctuations (See Figure 26).

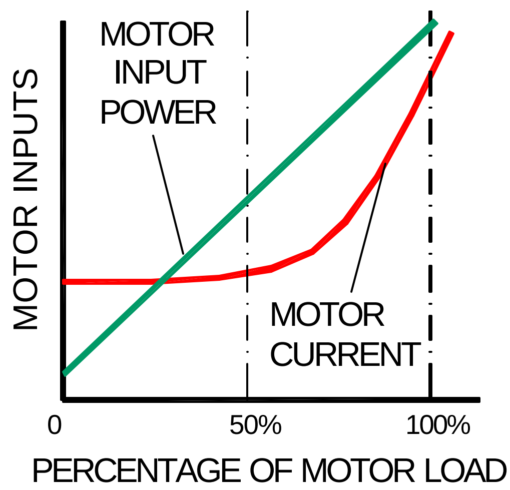

Figure 26. Motor Current and Input Power Curves

Motor Power relationship to Pump

The electrical input power of an AC induction motor is directly related to the motor output power by the motor efficiency:

Output Power = Input Power x Motor Efficiency

Motor output power or Brake Horsepower (BHP) is the amount of motor power required by a pump to maintain a specific flow and head at a constant motor speed. A basic centrifugal pump curve (Figure 27) demonstrates this concept.

To Predict a Pump Flow Rate:

a. Determine motor output power by multiplying the motor efficiency times the motor input power.

b. Locate the measured output power (P_A) on the pump performance BHP curve (assumes the actual specific gravity matches that of the referenced curve).

c. An imaginary vertical line is drawn through that point and intersects the x-axis at the predicted flow rate (Q_A).

d. This relationship is accurate as long as the motor speed is relatively constant.

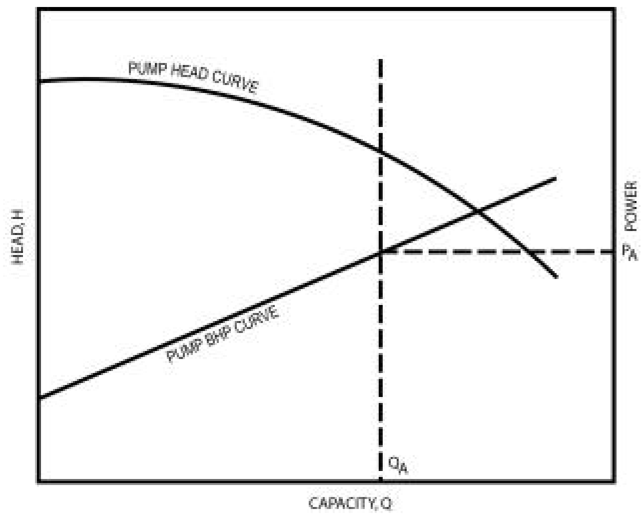

Figure 27. Determining Pump Flow Rate from Motor Power

Using a Power Monitor with Fixed Pump Speeds

A power monitor consists of a one-element wattmeter (electrical power measuring device connected to motor supply) and a relay (electromechanical switch). The monitor operates as a very simple flow indicator that associates a digital (input or calculated output power) value to a specific pump flow rate. The decision-making capability inside the power monitor permits the user to setup a power window. This window has a programmable low limit (value just above minimum flow) and a programmable high limit (a value equal to the point just prior to end of curve/cavitation). If the pump motor operates outside of this window longer than a defined time period, then the monitor opens a relay contact causing a controller to remove power from the pump motor, stopping the pump before major damage can occur.

 Basic Power Monitoring Unit (Model PM-1000 Shows)	 Optional Display Panel with Keypad (Model PM-2000 Shown)

Figure 28.

Measuring Power at Multiple Pump Speeds

A power monitor is configured to operate at a constant pump speed and at fixed trip limits. To measure input power of a pump motor operating at multiple speeds, a power transducer (three-element precision wattmeter with an analog output signal, 4-20 mA) certified for Variable Frequency Drive (VFD) duty is used.

With multiple speeds, a speed sensor is required along with a calculating and decision making device (such as a PLC-Programmable Logic Controller or PC-Personal Computer) to determine the low and high limits of the power window at the different motor speeds. A simple ratio equation for BHP related to initial impeller (motor) speed to other speeds could be derived from the basic pump Affinity Laws:

$$BHP_{new} = BHP_{initial} * (\text{speed value}_{new} / \text{speed value}_{initial})^3$$

The calculating and decision making device must:

- Look at the present power reading
- Compare reading to recalculated power window limits for that motor speed
- Make a decision to trigger an alarm or to stop the pump motor

10. Sundyne LMV Pumps - Hydraulics And Control

A major advantage of most Sundyne pumps is the lack of any close internal clearances. Therefore, the pump head-flow curve remains consistent through time and doesn't require over sizing to account for future performance degradation. An added feature of Sundyne pumps is the relative ease of replacing the hydraulic hardware to shift the head-flow curve. This feature can save power when systems continually operate with excessive throttling. In all cases, a properly sized pump provides substantial savings throughout the life of a unit.

Minimum Flow

When considering the minimum flow during operation, either the minimum continuous stable flow or the minimum continuous thermal flow may be the controlling factor for safe operation.
Pump operations below these points can cause high speed shaft vibration and reduce the mechanical seal life. These problems can be aggravated with high-energy pumps.

Potential Solutions for Addressing Minimum Flow Issues

- Increase minimum flow, i.e. valve set point.

- A discharge orifice located immediately downstream of the pump provides the same affect as locating the control valve close to the pump. The orifice plate isolates any water hammer affects between the pump and piping system.

- Converting the pump from concentric bowl hardware to Z-series hardware may offer the opportunity to reduce the pump minimum flow.

Before any of these solutions are tried, contact either your Representative or one of the factories located in Arvada, Colorado, or Dijon, France. Also, do not forget the minimum continuous thermal considerations when evaluating the reduction in minimum flow.

LMV Pump Model	Throat Style	Minimum Continuous Stable Flow (1)	Maximum Control Valve Distance From Pump (2)
801, 802, 806, 311, 322, 331, 341	Concentric Bowl (single throat)	Figure 29	Figure 29
801Z, 802Z, 311Z, 322Z	Z-Series (single throat)	35% of Rated	“Any” distance
331Z, 341Z	Z-Series (single throat)	35% of rated	10 feet (3 meters) (3)
311, 331, 341	ETA (single throat)	Figure 29	Figure 29
311, 331, 341	Twin throat (two throats)	55% of rated	10 feet (3 meters) (3)
313, 333, 343	Full Emission (Dual Throat)	20% of rated	“Any” distance

Table 2. Sundyne Pump Minimum Flow Requirements and Control Valve Location from Pump

a. Minimum continuous stable flow required for various Sundyne pump models.

b. The control valve is defined as the main forward flow control valve, not the minimum flow recycle valve.

d. This guideline cannot address all pump system variations, bypass arrangements, etc. Per API guidelines, an acceptable means to determine minimum stable flow in the field is to reduce total pump flow while monitoring vibration. Maximum allowable unfiltered vibration determines minimum stable flow.

Control Valve Location

Figure 29 below plots 'Distance to Control Valve' versus 'Percent of Rated Flow' as a function of power. The lower the pumps power the lower the allowable minimum flow and the farther away the control valve may be installed.

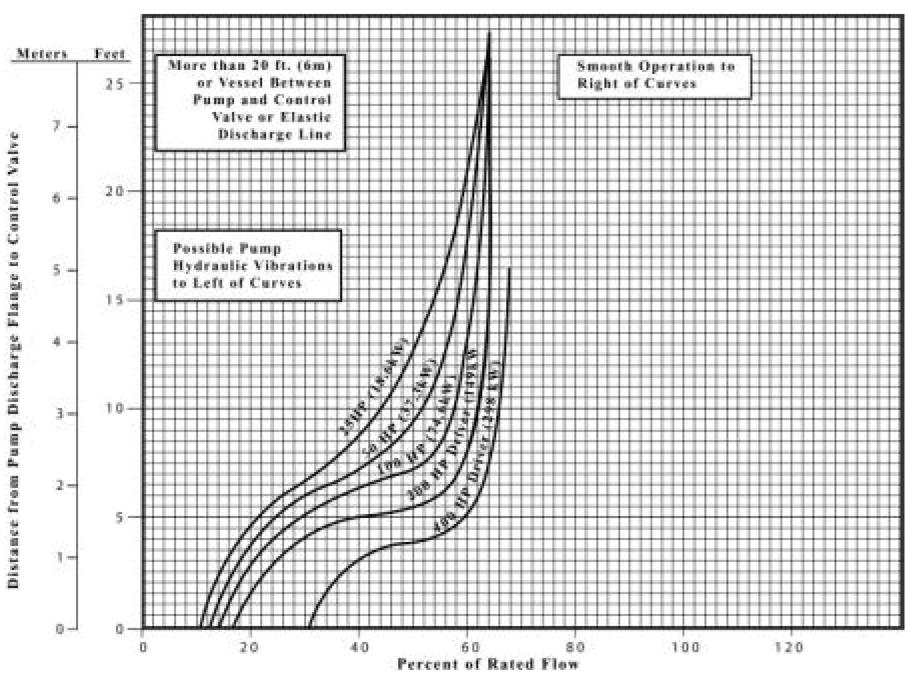

Figure 29. Recommended Flow for Smooth Operation of Concentric Bowl and ETA, Single Throat, Sundyne Pumps

11. Sunflo Hydraulics and Control

Hydraulic Design of Pump

SUNFLO pumps are not as sensitive to recirculation, turbulence, cavitation and high radial loads at low flow rates. Also the lack of any close internal clearances allows the pump head-flow curve to remain consistent through time and doesn't require over sizing to account for future performance degradation.

Each SUNFLO pump model has a defined minimum flow rate that is based on a resistive system with the control valve installed 5 feet from the discharge (see Table 3). The minimum flow is dependent on the hydraulic design of the impeller, inducer, and pump casing. This takes into account the recirculation, turbulence, surging, and cavitation inside the pump as well as the radial loads imposed on the impeller.

When a centrifugal pump is operated below its design flow, the passageways between the impeller blades are actually too large for the flow that is going through them. This results in a recirculation pattern from the tip of the impeller back to the suction, which is superimposed on the flow passing through the pump. As the flow rate is further decreased, the recirculation increases resulting in cavitation damage, erosion and pump surging (Figure 30). It would be possible to alter pump design to improve its low flow performance, but this generally results in reduced efficiency.

In volute pumps the hydraulic forces on the impeller increase at low flow rates. This can result in excessive load on the impeller, and resultant shaft deflection and bearing loads (Figures 31 & 32).

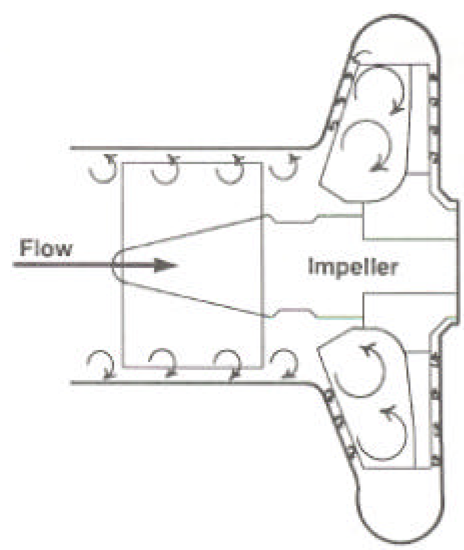

Figure 30. Recirculation at Low Flow Rates

Model	Minimum flow rate (1), (2)
P-1000 / P-1500	20% of BEP
P-2000 (8 bladed impeller)	25 – 50% of BEP
P-2000 (24 bladed impeller)	20% of BEP
P-3000 Diffuser/Open Impeller	20% of BEP

NOTE:
1. Minimum flow rate is based on a cold resistive system with a control valve installed at a maximum of 5 feet from the discharge.
2. Minimum flow may be limited by temperature rise across the pump or by the other system limitations.
3. BEP = Best Efficiency Point

Table 3. Minimum Flow Rates for Sunflo Pumps

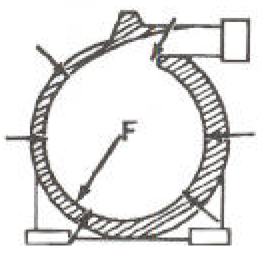

Figure 31A. Pump operating below design flow. As flow decreases the impeller load (F) increases, and is away from the discharge.

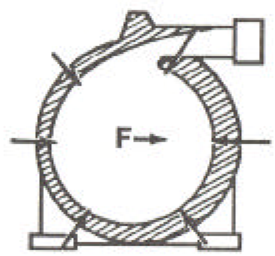

Figure 31B. Pump operating at design flow. The resultant hydraulic impeller load (F) is low in magnitude.

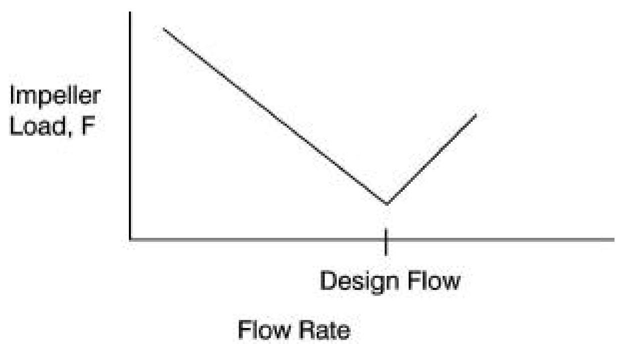

Figure 32. Impeller Load Versus Flow Rate for a Single Volute Pump

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