6. PUMPS AND PUMPING SYSTEM

Syllabus

Pumps and Pumping System: Types, Performance evaluation, Efficient system operation, Flow control strategies and energy conservation opportunities

6.1 Pump Types

Pumps come in a variety of sizes for a wide range of applications. They can be classified according to their basic operating principle as dynamic or displacement pumps. Dynamic pumps can be sub-classified as centrifugal and special effect pumps. Displacement pumps can be sub-classified as rotary or reciprocating pumps.

In principle, any liquid can be handled by any of the pump designs. Where different pump designs could be used, the centrifugal pump is generally the most economical followed by rotary and reciprocating pumps. Although, positive displacement pumps are generally more efficient than centrifugal pumps, the benefit of higher efficiency tends to be offset by increased maintenance costs.

Since, worldwide, centrifugal pumps account for the majority of electricity used by pumps, the focus of this chapter is on centrifugal pump.

Centrifugal Pumps

A centrifugal pump is of a very simple design. The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving part, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, polycarbonate, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and captures and directs the water off the impeller.

Water enters the center (eye) of the impeller and exits the impeller with the help of centrifugal force. As water leaves the eye of the impeller a low-pressure area is created, causing more water to flow into the eye. Atmospheric pressure and centrifugal force cause this to happen. Velocity is developed as the water flows through the impeller spinning at high speed. The water velocity is collected by the diffuser and converted to pressure by specially designed passageways that direct the flow to the discharge of the pump, or to the next impeller should the pump have a multi-stage configuration.

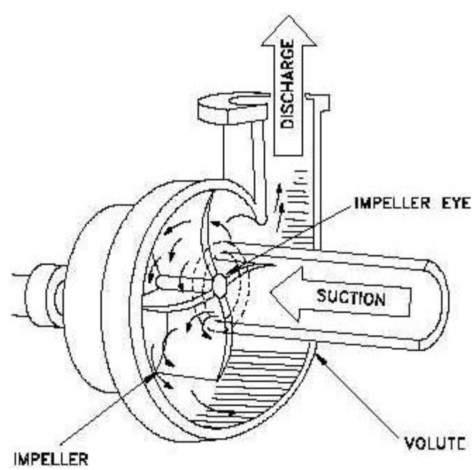
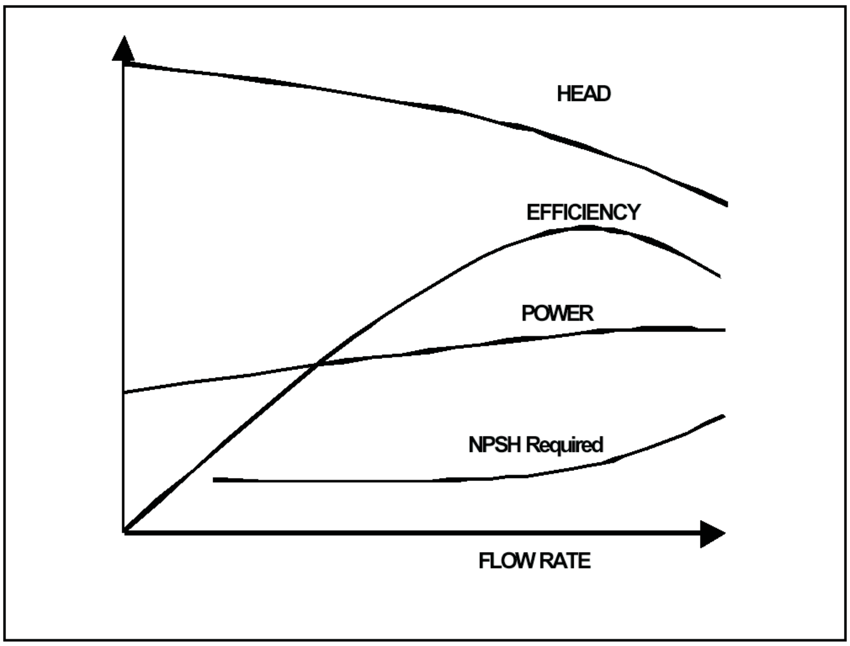


Figure 6.1 Centrifugal pump

The pressure (head) that a pump will develop is in direct relationship to the impeller diameter, the number of impellers, the size of impeller eye, and shaft speed. Capacity is determined by the exit width of the impeller. The head and capacity are the main factors, which affect the horsepower size of the motor to be used. The more the quantity of water to be pumped, the more energy is required.

A centrifugal pump is not positive acting; it will not pump the same volume always. The greater the depth of the water, the lesser is the flow from the pump. Also, when it pumps against increasing pressure, the less it will pump. For these reasons it is important to select a centrifugal pump that is designed to do a particular job.

Since the pump is a dynamic device, it is convenient to consider the pressure in terms of head i.e. meters of liquid column. The pump generates the same head of liquid whatever the density of the liquid being pumped. The actual contours of the hydraulic passages of the impeller and the casing are extremely important, in order to attain the highest efficiency possible. The standard convention for centrifugal pump is to draw the pump performance curves showing Flow on the horizontal axis and Head generated on the vertical axis. Efficiency, Power & NPSH Required (described later), are conventionally shown on the vertical axis, plotted against Flow, as illustrated in Figure 6.2.



System

Figure 6.2 Pump Performance Curve

Given the significant amount of electricity attributed to pumping systems, even small improvements in pumping efficiency could yield very significant savings of electricity. The pump is among the most inefficient of the components that comprise a pumping system, including the motor, transmission drive, piping and valves.

Hydraulic power, pump shaft power and electrical input power

Hydraulic power P_h = Q (m^3/s) x Total head, h_d - h_s (m) x ρ (kg/m^3) x g (m/s^2) / 1000

Where h_d – discharge head, h_s – suction head, ρ – density of the fluid, g – acceleration due to gravity

Pump shaft power P_s = Hydraulic power, P_h / pump efficiency, η_{Pump}

Electrical input power = $\dfrac{\text{Pump shaft power } P_s}{\eta_{Motor}}$

6.2 System Characteristics

In a pumping system, the objective, in most cases, is either to transfer a liquid from a source to a required destination, e.g. filling a high level reservoir, or to circulate liquid around a system, e.g. as a means of heat transfer in heat exchanger.

A pressure is needed to make the liquid flow at the required rate and this must overcome head 'losses' in the system. Losses are of two types: static and friction head.

Static head is simply the difference in height of the supply and destination reservoirs, as in Figure 6.3. In this illustration, flow velocity in the pipe is assumed to be very small. Another example of a system with only static head is pumping into a pressurised vessel with short pipe runs. Static head is independent of flow and graphically would be shown as in Figure 6.4.

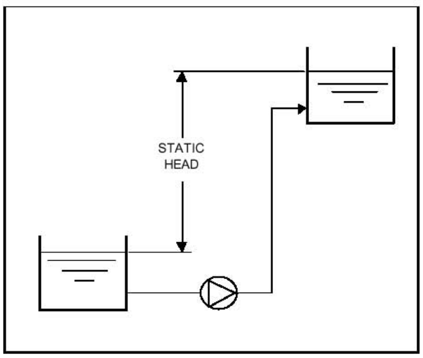


Figure 6.3 Static Head

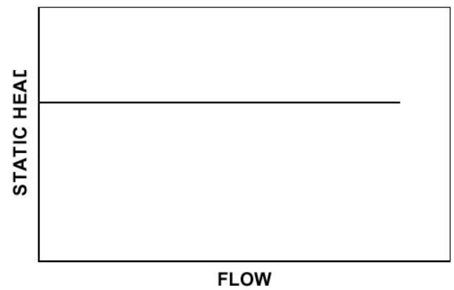


Figure 6.4 Static Head vs. Flow

Friction head (sometimes called dynamic head loss) is the friction loss, on the liquid being moved, in pipes, valves and equipment in the system. Friction tables are universally available for various pipe fittings and valves. These tables show friction loss per 100 feet (or metres) of a specific pipe size at various flow rates. In case of fittings, friction is stated as an equivalent length of pipe of the same size. The friction losses are proportional to the square of the flow rate. A closed loop circulating system without a surface open to atmospheric pressure, would exhibit only friction losses and would have a system friction head loss vs. flow curve as Figure 6.5.

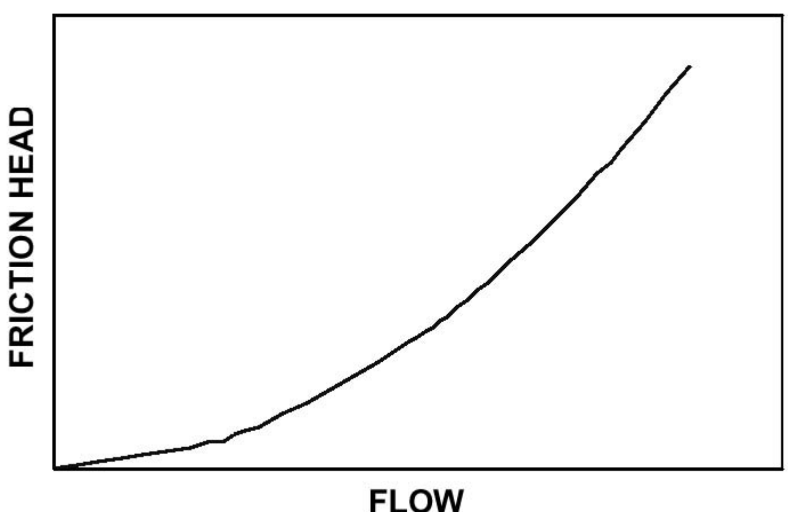


Figure 6.5 Friction Head vs. Flow

Most systems have a combination of static and friction head and the system curves for two cases are shown in Figures 6.6 and 6.7. The ratio of static to friction head over the operating range influences the benefits achievable from variable speed drives which shall be discussed later.

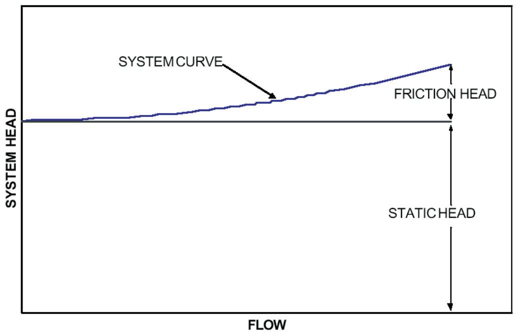


Figure 6.6 System with High Static Head

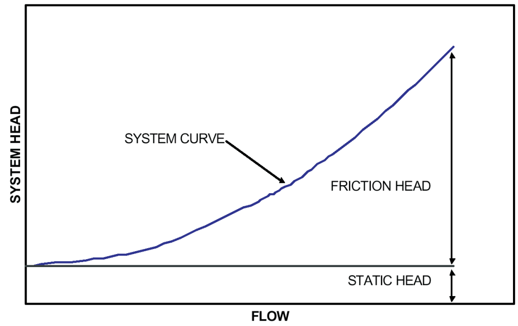


Figure 6.7 System with Low Static Head

Static head is a characteristic of the specific installation and reducing this head where this is possible, generally helps both the cost of the installation and the cost of pumping the liquid. Friction head losses must be minimised to reduce pumping cost, but after eliminating unnecessary pipe fittings and length, further reduction in friction head will require larger diameter pipe, which adds to installation cost.

6.3 Pump Curves

The performance of a pump can be expressed graphically as head against flow rate. The centrifugal pump has a curve where the head falls gradually with increasing flow. This is called the pump characteristic curve (Head - Flow curve) -see Figure 6.8.

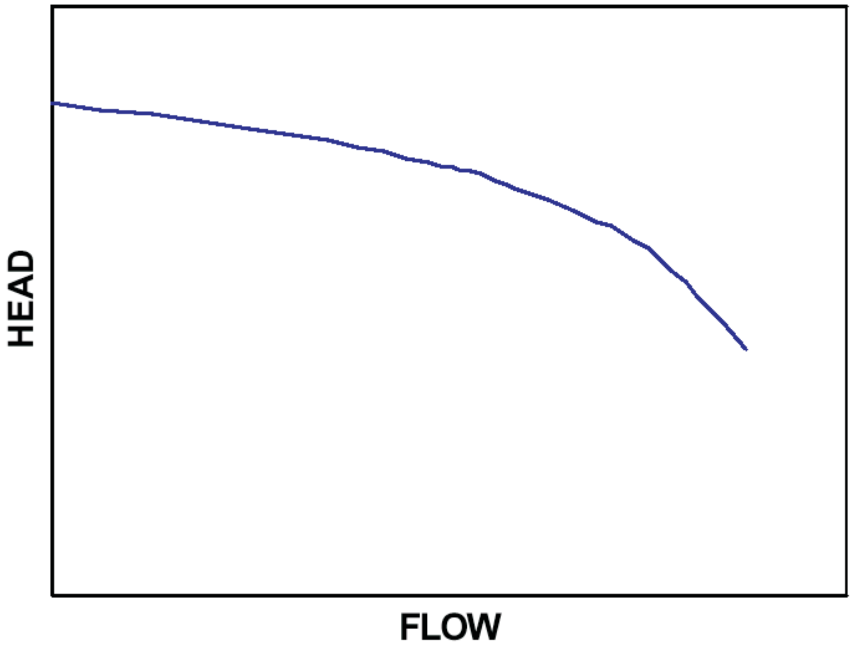


Figure 6.8 Head- Flow Curve

Pump operating point

When a pump is installed in a system the effect can be illustrated graphically by superimposing pump and system curves. The operating point will always be where the two curves intersect. Figure 6.9.

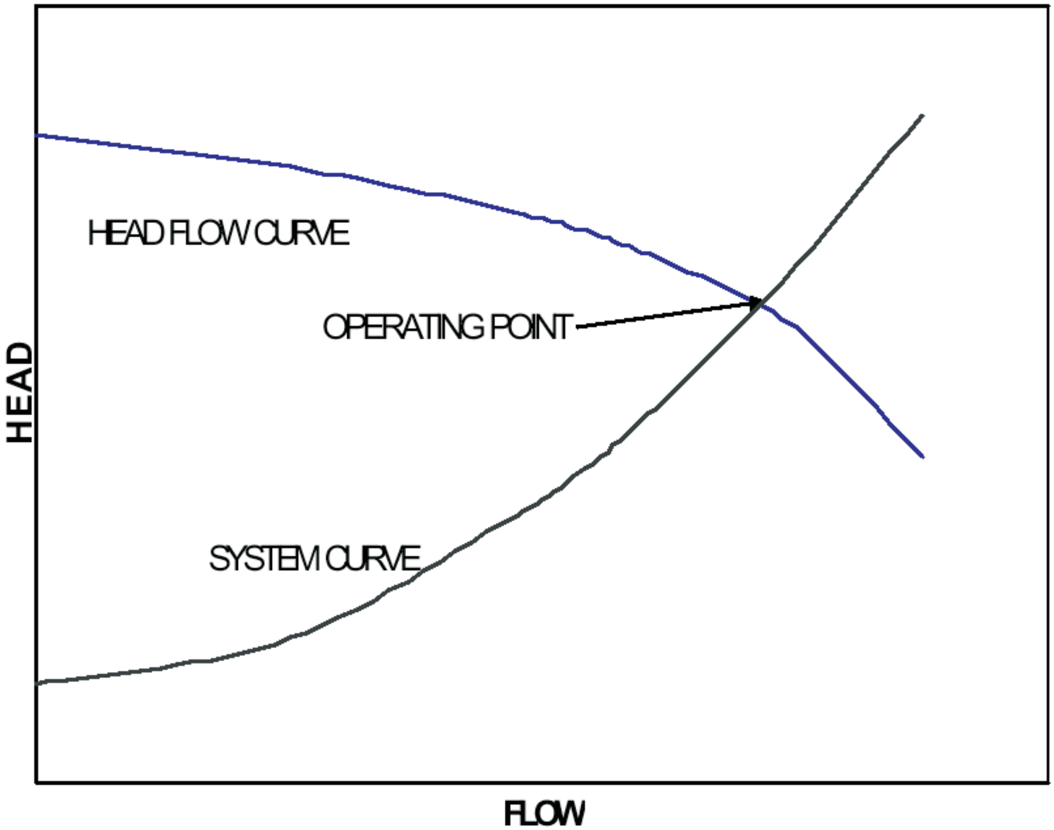


Figure 6.9 Pump Operating Point

If the actual system curve is different in reality to that calculated, the pump will operate at a flow and head different to that expected.

For a centrifugal pump, an increasing system resistance will reduce the flow, eventually to zero, but the maximum head is limited as shown. Even so, this condition is only acceptable for a short period without causing problems. An error in the system curve calculation is also likely to lead to a centrifugal pump selection, which is less than optimal for the actual system head losses. Adding safety margins to the calculated system curve to ensure that a sufficiently large pump is selected will generally result in installing an oversized pump, which will operate at an excessive flow rate or in a throttled condition, which increases energy usage and reduces pump life.

6.4 Factors Affecting Pump Performance

Matching Pump and System Head-flow Characteristics

Centrifugal pumps are characterized by the relationship between the flow rate (Q) they produce and the pressure (H) at which the flow is delivered. Pump efficiency varies with flow and pressure, and it is highest at one particular flow rate.

The Figure 6.10 below shows a typical vendor-supplied head-flow curve for a centrifugal pump. Pump head-flow curves are typically given for clear water. The choice of pump for a given application depends largely on how the pump head-flow characteristics match the requirement of the system downstream of the pump.

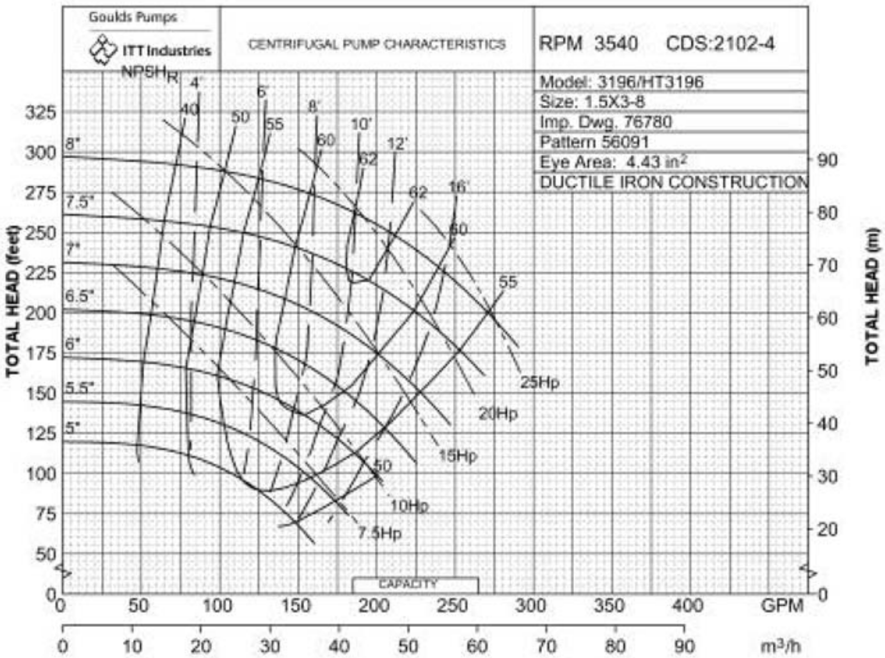


Figure 6.10 Typical Centrifugal Pump Performance Curve

Effect of over sizing the pump

As mentioned earlier, pressure losses to be overcome by the pumps are function of flow – the system characteristics – are also quantified in the form of *head-flow curves*. The system curve is basically a plot of system resistance i.e. head to be overcome by the pump versus various flow rates. The system curves change with the physical configuration of the system; for example, the system curves depends upon height or elevation, diameter and length of piping, number and type of fittings and pressure drops across various equipment - say a heat exchanger.

A pump is selected based on how well the pump curve and system head-flow curves match. The pump operating point is identified as the point, where the system curve crosses the pump curve when they are superimposed on each other.

The Figure 6.11 shows the effect on system curve with throttling.

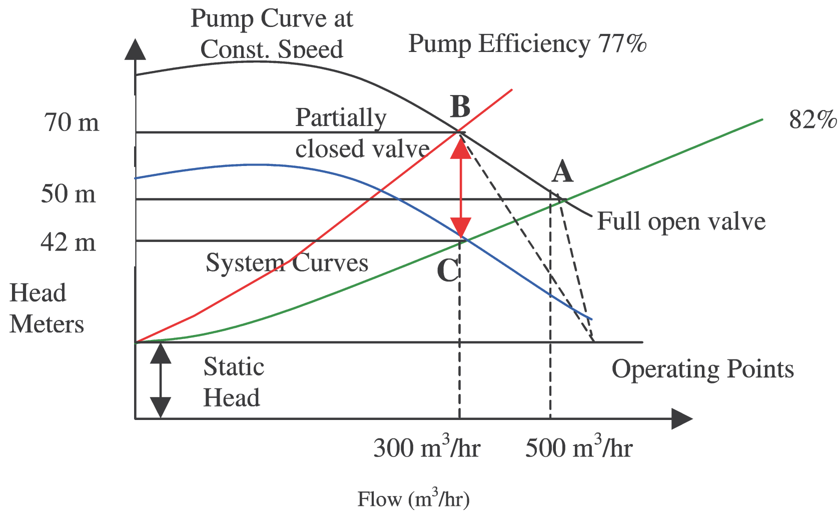


Figure 6.11 Effect on System Curve with Throttling

In the system under consideration, water has to be first lifted to a height – this represents the static head.

Then, we make a system curve, considering the friction and pressure drops in the system- this is shown as the green curve.

Suppose, we have estimated our operating conditions as 500 m^3/hr flow and 50 m head, we will chose a pump curve which intersects the system curve (Point A) at the pump's *best efficiency point* (BEP).

But, in actual operation, we find that 300 m^3/hr is sufficient. The reduction in flow rate has to be effected by a throttle valve. In other words, we are introducing an artificial resistance in the system.

Due to this additional resistance, the frictional part of the system curve increases and thus the new system curve will shift to the left -this is shown as the red curve.

So the pump has to overcome additional pressure in order to deliver the reduced flow. Now, the new system curve will intersect the pump curve at point B. The revised parameters are 300 m^3/hr at 70 m head. The red double arrow line shows the additional pressure drop due to throttling.

You may note that the best efficiency point has shifted from 82% to 77% efficiency.

So what we want is to actually operate at point C which is 300 m^3/hr on the original system curve. The head required at this point is only 42 meters.

What we now need is a new pump which will operate with its best efficiency point at C. But there are other simpler options rather than replacing the pump. The speed of the pump can be reduced or the existing impeller can be trimmed (or new lower size impeller). The blue pump curve represents either of these options.

Energy loss in throttling

Consider a case (see Figure 6.12) where we need to pump 68 m^3/hr of water at 47 m head. The pump characteristic curves (A…E) for a range of pumps are given in the Figure 6.12.

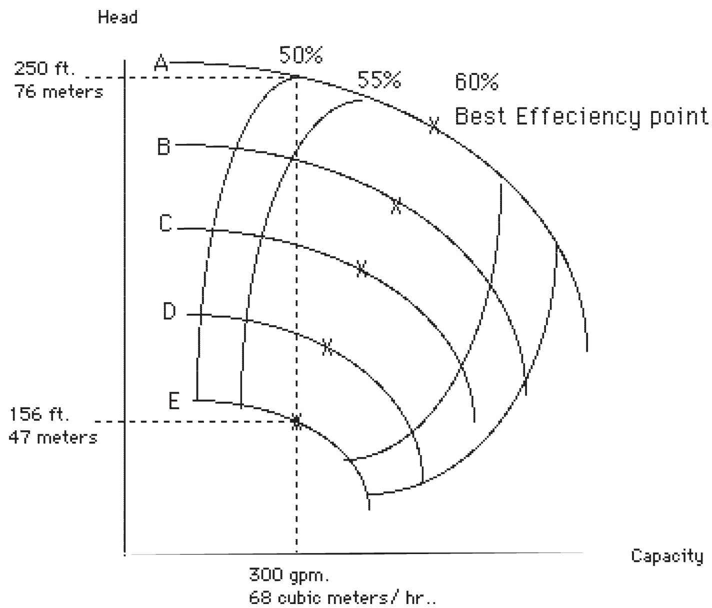


Figure 6.12 Pump Characteristic Curves

If we select E, then the pump efficiency is 60%

Hydraulic Power = Q (m^3/s) x Total head, $h_d - h_s$ (m) x ρ (kg/m^3) x g (m/s^2) / 1000

$$= \frac{(68/3600) \times 47 \times 1000 \times 9.81}{1000}$$

= 8.7 kW

Shaft Power - 8.7 / 0.60 = 14.5 Kw
Motor Power - 14.5 / 0.9 = 16.1Kw (considering a motor efficiency of 90%)

If we select A, then the pump efficiency is 50% (drop from earlier 60%)

Obviously, this is an oversize pump. Hence, the pump has to be throttled to achieve the desired flow. Throttling increases the head to be overcome by the pump. In this case, head is 76 metres.

Hydraulic Power = Q (m^3/s) x Total head, $h_d - h_s$ (m) x ρ (kg/m^3) x g (m /s^2) / 1000

$$= \frac{(68/3600) \times 76 \times 1000 \times 9.81}{1000}$$

= 14 kW

Shaft Power - 14 / 0.50 = 28 Kw
Motor Power - 28 / 0.9 = 31 Kw (considering a motor efficiency of 90%)

Hence, additional power drawn by A over E is 31 –16.1 = 14.9 kW.

Extra energy used = 8760 hrs/yr x 14.9 = 1,30,524 kwh/annum
= Rs. 5,22,096/annum

In this example, the extra cost of the electricity is more than the cost of purchasing a new pump.

6.5 Efficient Pumping System Operation

To understand a pumping system, one must realize that all of its components are interdependent. When examining or designing a pump system, the process demands must first be established and most energy efficiency solution introduced. For example, does the flow rate have to be regulated continuously or in steps? Can on-off batch pumping be used? What are the flow rates needed and how are they distributed in time?

The first step to achieve energy efficiency in pumping system is to target the end-use. A plant water balance would establish usage pattern and highlight areas where water consumption can be reduced or optimized. Good water conservation measures, alone, may eliminate the need for some pumps.

Once flow requirements are optimized, then the pumping system can be analysed for energy conservation opportunities. Basically this means matching the pump to requirements by adopting proper flow control strategies. Common symptoms that indicate opportunities for energy efficiency in pumps are given in the Table 6.1.

TABLE 6.1 SYMPTOMS THAT INDICATE POTENTIAL OPPORTUNITY FOR ENERGY SAVINGS

Symptom	Likely Reason	Best Solutions
Throttle valve-controlled systems	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower rpm
Bypass line (partially or completely) open	Oversized pump	Trim impeller, smaller impeller, variable speed drive, two speed drive, lower rpm
Multiple parallel pump system with the same number of pumps always operating	Pump use not monitored or controlled	Install controls
Constant pump operation in a batch environment	Wrong system design	On-off controls
High maintenance cost (seals, bearings)	Pump operated far away from BEP	Match pump capacity with system requirement

Effect of speed variation

As stated above, a centrifugal pump is a dynamic device with the head generated from a rotating impeller. There is therefore a relationship between impeller peripheral velocity and generated head. Peripheral velocity is directly related to shaft rotational speed, for a fixed impeller diameter and so varying the rotational speed has a direct effect on the performance of the pump. All the parameters shown in fig 6.2 will change if the speed is varied and it is important to have an appreciation of how these parameters vary in order to safely control a pump at different speeds. The equations relating rotodynamic pump performance parameters of flow, head and power absorbed, to speed are known as the *Affinity Laws*:

$$Q \propto N$$
$$H \propto N^2$$
$$P \propto N^3$$

Where:
Q = Flow rate
H = Head
P = Power absorbed
N = Rotating speed
Efficiency is essentially independent of speed

Flow: Flow is proportional to the speed

$$\mathbf{Q_1 / Q_2 = N_1 / N_2}$$
Example: $\mathbf{100 / Q_2 = 1750/3500}$
$\mathbf{Q_2 = 200\ m^3/hr}$

Head: Head is proportional to the square of speed

$$H_1/H_2 = (N_1^2) / (N_2^2)$$
Example: $100 / H_2 = 1750^2 / 3500^2$
$H_2 = 400$ m

Power(kW): Power is proportional to the cube of speed

$$\mathbf{kW_1 / kW_2 = (N_1^3) / (N_2^3)}$$
Example: $\mathbf{5/kW_2 = 1750^3 / 3500^3}$
$\mathbf{kW_2 = 40}$

As can be seen from the above laws, doubling the speed of the centrifugal pump will increase the power consumption by 8 times. Conversely a small reduction in speed will result in drastic reduction in power consumption. This forms the basis for energy conservation in centrifugal pumps with varying flow requirements. The implication of this can be better understood as shown in an example of a centrifugal pump in Figure 6.13 below.

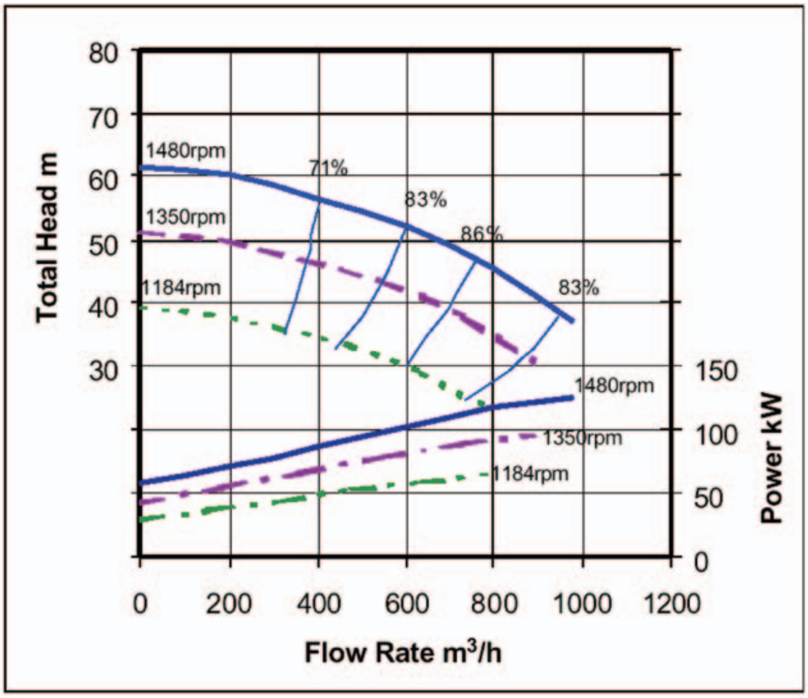


Figure 6.13 Example of Speed Variation Effecting Centrifugal Pump Performance

Points of equal efficiency on the curves for the 3 different speeds are joined to make the iso-efficiency lines, showing that efficiency remains constant over small changes of speed providing the pump continues to operate at the same position related to its best efficiency point (BEP).

The affinity laws give a good approximation of how pump performance curves change with speed but in order to obtain the actual performance of the pump in a system, the system curve also has to be taken into account.

Effects of impeller diameter change

Changing the impeller diameter gives a proportional change in peripheral velocity, so it follows that there are equations, similar to the affinity laws, for the variation of performance with impeller diameter D:

$$Q \propto D$$

$$H \propto D^2$$

$$P \propto D^3$$

Efficiency varies when the diameter is changed within a particular casing. Note the difference in iso-efficiency lines in Figure 6.14 compared with Figure 6.13. The relationships shown here apply to the case for changing only the diameter of an impeller within a fixed casing geometry, which is a common practice for making small permanent adjustments to the performance of a centrifugal pump. Diameter changes are generally limited to reducing the diameter to about 75% of the maximum, i.e. a head reduction to about 50%. Beyond this, efficiency and NPSH are badly affected. However speed change can be used over a wider range without seriously reducing efficiency. For example reducing the speed by 50% typically results in a reduction of efficiency by 1 or 2 percentage points. The reason for the small loss of efficiency with the lower speed is that mechanical losses in seals and bearings, which generally represent <5% of total power, are proportional to speed, rather than speed cubed. It should be noted that if the change in diameter is more than about 5%, the accuracy of the squared and cubic relationships can fall off and for precise calculations, the pump manufacturer's performance curves should be referred to.

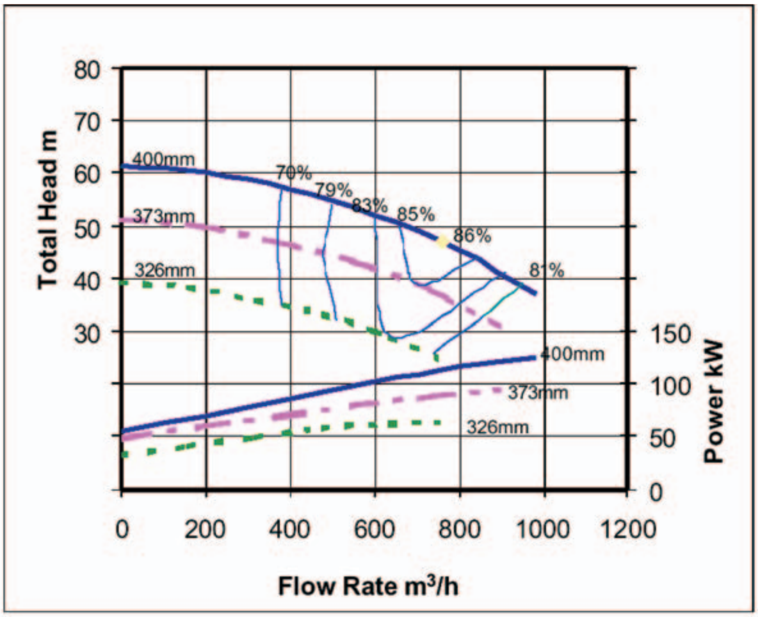


Figure 6.14 Example: Impeller Diameter Reduction on Centrifugal Pump Performance

The illustrated curves are typical of most centrifugal pump types. Certain high flow, low head pumps have performance curve shapes somewhat different and have a reduced operating region of flows. This requires additional care in matching the pump to the system, when changing speed and diameter.

Pump suction performance (NPSH)

Liquid entering the impeller eye turns and is split into separate streams by the leading edges of the impeller vanes, an action which locally drops the pressure below that in the inlet pipe to the pump.

If the incoming liquid is at a pressure with insufficient margin above its vapour pressure, then vapour cavities or bubbles appear along the impeller vanes just behind the inlet edges. This phenomenon is known as cavitation and has three undesirable effects:

1) The collapsing cavitation bubbles can erode the vane surface, especially when pumping water-based liquids.
2) Noise and vibration are increased, with possible shortened seal and bearing life.
3) The cavity areas will initially partially choke the impeller passages and reduce the pump performance. In extreme cases, total loss of pump developed head occurs.

The value, by which the pressure in the pump suction exceeds the liquid vapour pressure, is expressed as a head of liquid and referred to as Net Positive Suction Head Available – (NPSHA). This is a characteristic of the system design. The value of NPSH needed at the pump suction to prevent the pump from cavitating is known as NPSH Required – (NPSHR). This is a characteristic of the pump design.

The three undesirable effects of cavitation described above begin at different values of NPSHA and generally there will be cavitation erosion before there is a noticeable loss of pump head. However for a consistent approach, manufacturers and industry standards, usually define the onset of cavitation as the value of NPSHR when there is a head drop of 3% compared with the head with cavitation free performance. At this point cavitation is present and prolonged operation at this point will usually lead to damage. It is usual therefore to apply a margin bywhich NPSHA should exceed NPSHR.

As would be expected, the NPSHR increases as the flow through the pump increases, see fig 6.2. In addition, as flow increases in the suction pipework, friction losses also increase, giving a lower NPSHA at the pump suction, both of which give a greater chance that cavitation will occur. NPSHR also varies approximately with the square of speed in the same way as pump head and conversion of NPSHR from one speed to another can be made using the following equations.

$$Q \propto N$$

$$NPSHR \propto N^2$$

It should be noted however that at very low speeds there is a minimum NPSHR plateau, NPSHR does not tend to zero at zero speed It is therefore essential to carefully consider NPSH in variable speed pumping.

6.6 Flow Control Strategies

Pump control by varying speed

To understand how speed variation changes the duty point, the pump and system curves are over-laid. Two systems are considered, one with only friction loss and another where static head is high in relation to friction head. It will be seen that the benefits are different. In Figure 6.15,

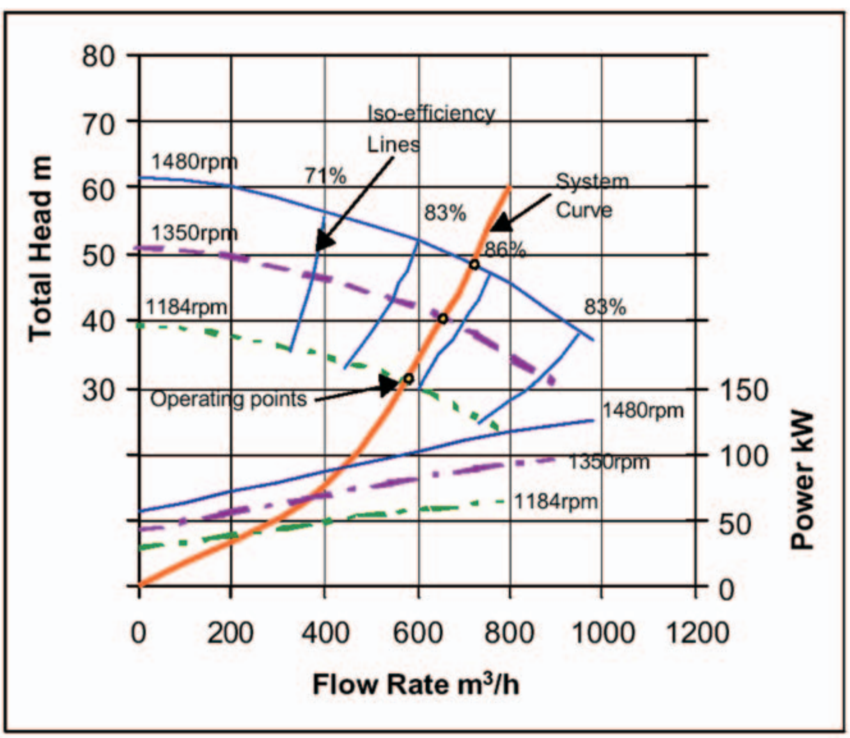


Figure 6.15 Example of the Effect of Pump Speed Change in a System With Only Friction Loss

reducing speed in the friction loss system moves the intersection point on the system curve along a line of constant efficiency. The operating point of the pump, relative to its best efficiency point, remains constant and the pump continues to operate in its ideal region. The affinity laws are obeyed which means that there is a substantial reduction in power absorbed accompanying the reduction in flow and head, making variable speed the ideal control method for systems with friction loss.

In a system where static head is high, as illustrated in Figure 6.16, the operating point for the pump moves relative to the lines of constant pump efficiency when the speed is changed. The reduction in flow is no longer proportional to speed. A small turn down in speed could give a big reduction in flow rate and pump efficiency, which could result in the pump operating in a region where it could be damaged if it ran for an extended period of time even at the lower speed. At the lowest speed illustrated, (1184 rpm), the pump does not generate sufficient head to pump any liquid into the system, i.e. pump efficiency and flow rate are zero and with energy still being input to the liquid, the pump becomes a water heater and damaging temperatures can quickly be reached.

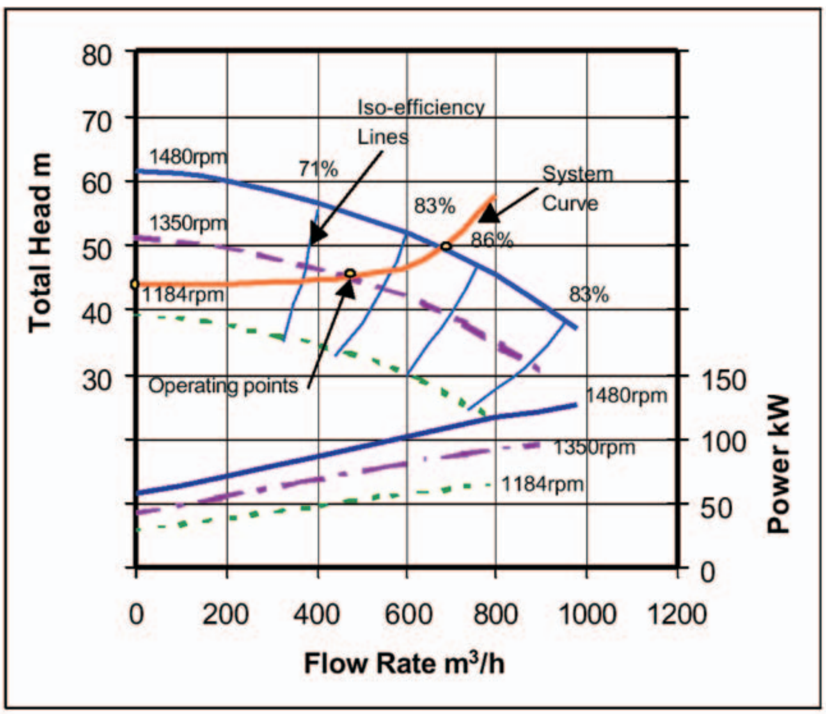


Figure 6.16 Example for the Effect of Pump Speed Change with a System with High Static Head.

The drop in pump efficiency during speed reduction in a system with static head, reduces the economic benefits of variable speed control. There may still be overall benefits but economics should be examined on a case-by-case basis. Usually it is advantageous to select the pump such that the system curve intersects the full speed pump curve to the right of best efficiency, in order that the efficiency will first increase as the speed is reduced and then decrease. This can extend the useful range of variable speed operation in a system with static head. The pump manufacturer should be consulted on the safe operating range of the pump.

It is relevant to note that flow control by speed regulation is always more efficient than by control valve. In addition to energy savings there could be other benefits of lower speed. The hydraulic forces on the impeller, created by the pressure profile inside the pump casing, reduce approximately with the square of speed. These forces are carried by the pump bearings and so reducing speed increases bearing life. It can be shown that for a centrifugal pump, bearing life is inversely proportional to the 7^{th} power of speed. In addition, vibration and noise are reduced and seal life is increased providing the duty point remains within the allowable operating range.

The corollary to this is that small increases in the speed of a pump significantly increase power absorbed, shaft stress and bearing loads. It should be remembered that the pump and motor must be sized for the maximum speed at which the pump set will operate. At higher speed the noise and vibration from both pump and motor will increase, although for small increases the change will be small. If the liquid contains abrasive particles, increasing speed will give a corresponding increase in surface wear in the pump and pipework.

The effect on the mechanical seal of the change in seal chamber pressure, should be reviewed with the pump or seal manufacturer, if the speed increase is large. Conventional mechanical seals operate satisfactorily at very low speeds and generally there is no requirement for a minimum speed to be specified, however due to their method of operation, gas seals require a minimum peripheral speed of 5 m/s.

Pumps in parallel switched to meet demand

Another energy efficient method of flow control, particularly for systems where static head is a high proportion of the total, is to install two or more pumps to operate in parallel. Variation of flow rate is achieved by switching on and off additional pumps to meet demand. The combined pump curve is obtained by adding the flow rates at a specific head. The head/flow rate curves for two and three pumps are shown in Figure 6.17.

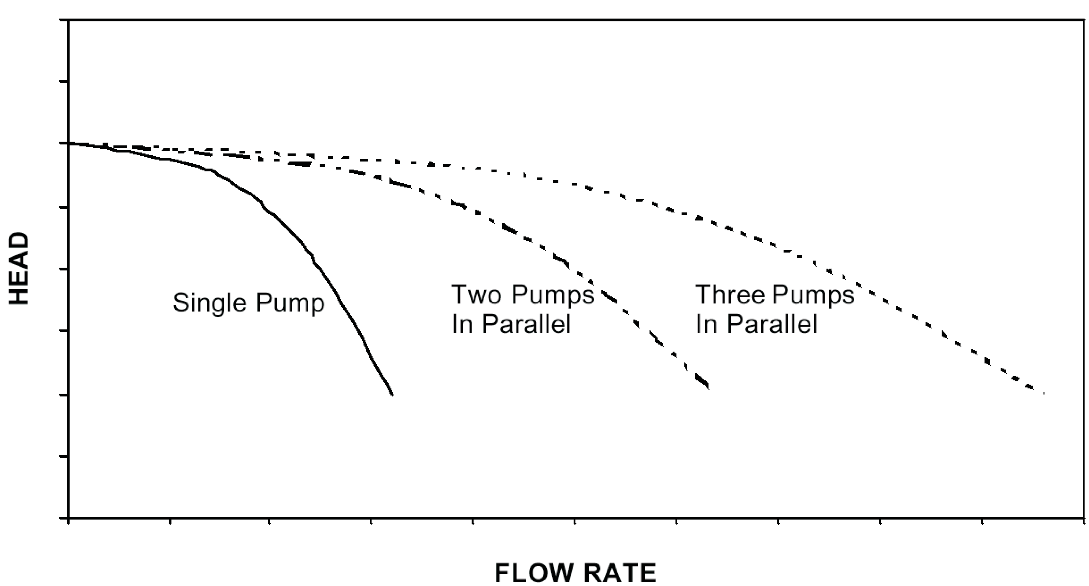


Figure 6.17 Typical Head-Flow Curves for Pumps in Parallel

The system curve is usually not affected by the number of pumps that are running. For a system with a combination of static and friction head loss, it can be seen, in Figure 6.18, that the operating point of the pumps on their performance curves moves to a higher head and hence lower flow rate per pump, as more pumps are started. It is also apparent that the flow rate with two pumps running is not double that of a single pump. If the system head were only static, then flow rate would be proportional to the number of pumps operating.

It is possible to run pumps of different sizes in parallel provided their closed valve heads are similar. By arranging different combinations of pumps running together, a larger number of different flow rates can be provided into the system.

Care must be taken when running pumps in parallel to ensure that the operating point of the pump is controlled within the region deemed as acceptable by the manufacturer. It can be seen from Figure 6.18 that if 1 or 2 pumps were stopped then the remaining pump(s) would operate well out along the curve where NPSH is higher and vibration level increased, giving an increased risk of operating problems.

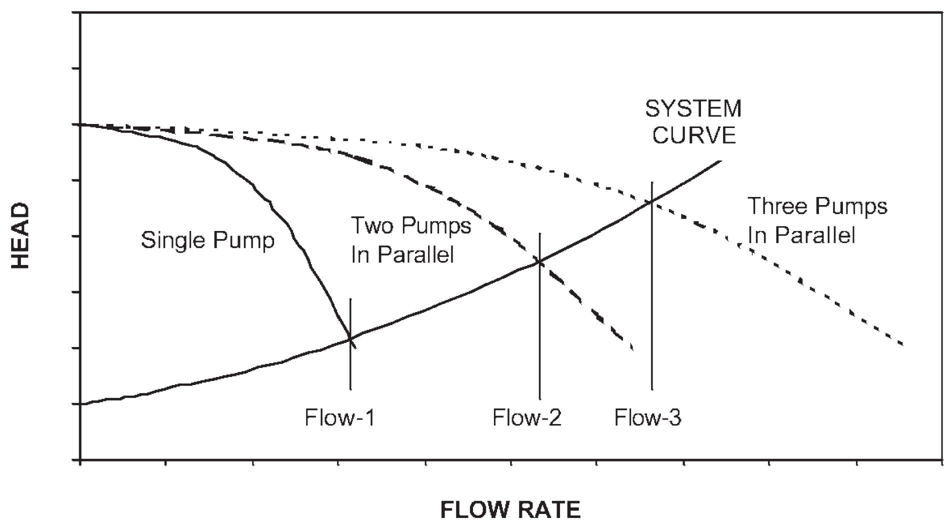


Figure 6.18 Typical Head-Flow Curves for Pumps in Parallel, With System Curve Illustrated.

Stop/start control

In this control method, the flow is controlled by switching pumps on or off. It is necessary to have a storage capacity in the system e.g. a wet well, an elevated tank or an accumulator type pressure vessel. The storage can provide a steady flow to the system with an intermittent operating pump. When the pump runs, it does so at the chosen (presumably optimum) duty point and when it is off, there is no energy consumption. If intermittent flow, stop/start operation and the storage facility are acceptable, this is an effective approach to minimise energy consumption.

The stop/start operation causes additional loads on the power transmission components and increased heating in the motor. The frequency of the stop/start cycle should be within the motor design criteria and checked with the pump manufacturer.

It may also be used to benefit from "off peak" energy tariffs by arranging the run times during the low tariff periods.

To minimise energy consumption with stop start control it is better to pump at as low flow rate as the process permits. This minimises friction losses in the pipe and an appropriately small pump can be installed. For example, pumping at half the flow rate for twice as long can reduce energy consumption to a quarter.

Flow control valve

With this control method, the pump runs continuously and a valve in the pump discharge line is opened or closed to adjust the flow to the required value.

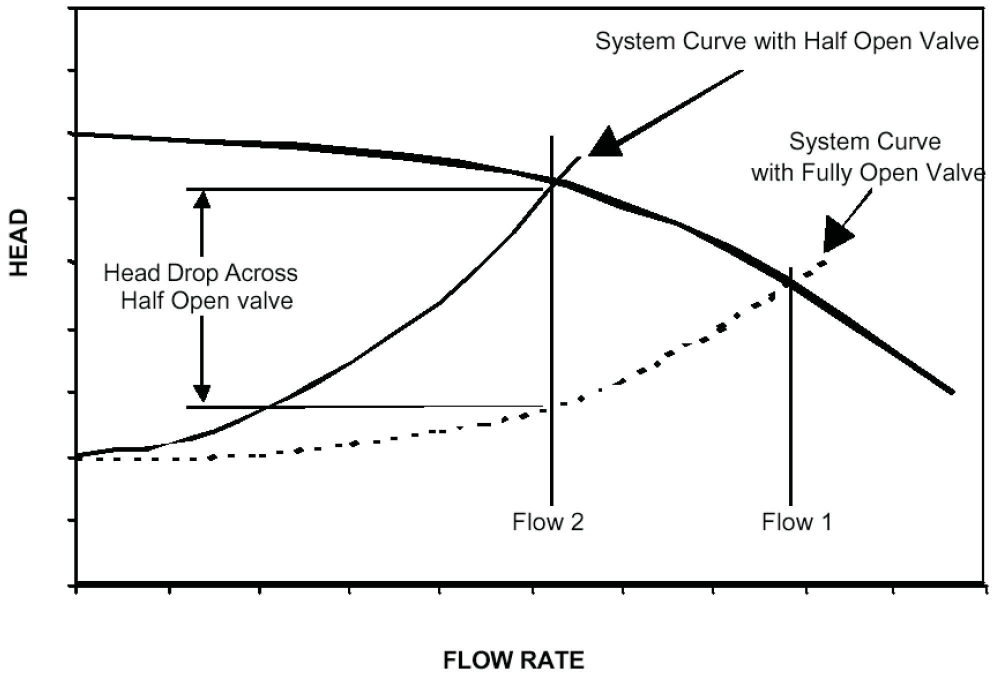


Figure 6.19 Control of Pump Flow by Changing System Resistance Using a Valve.

To understand how the flow rate is controlled, see Figure 6.19. With the valve fully open, the pump operates at "Flow 1". When the valve is partially closed it introduces an additional friction loss in the system, which is proportional to flow squared. The new system curve cuts the pump curve at "Flow 2", which is the new operating point. The head difference between the two curves is the pressure drop across the valve.

It is usual practice with valve control to have the valve 10% shut even at maximum flow. Energy is therefore wasted overcoming the resistance through the valve at all flow conditions. There is some reduction in pump power absorbed at the lower flow rate (see Figure 6.19), but the flow multiplied by the head drop across the valve, is wasted energy. It should also be noted that, while the pump will accommodate changes in its operating point as far as it is able within its performance range, it can be forced to operate high on the curve, where its efficiency is low, and its reliability is affected.

Maintenance cost of control valves can be high, particularly on corrosive and solids-containing liquids. Therefore, the lifetime cost could be unnecessarily high.

By-pass control

With this control approach, the pump runs continuously at the maximum process demand duty, with a permanent by-pass line attached to the outlet. When a lower flow is required the surplus liquid is bypassed and returned to the supply source.

An alternative configuration may have a tank supplying a varying process demand, which is kept full by a fixed duty pump running at the peak flow rate. Most of the time the tank over-flows and recycles back to the pump suction. This is even less energy efficient than a control valve because there is no reduction in power consumption with reduced process demand.

The small by-pass line sometimes installed to prevent a pump running at zero flow is not a means of flow control, but required for the safe operation of the pump.

Fixed Flow reduction

Impeller trimming

Impeller trimming refers to the process of machining the diameter of an impeller to reduce the energy added to the system fluid.

Impeller trimming offers a useful correction to pumps that, through overly conservative design practices or changes in system loads are oversized for their application.

Trimming an impeller provides a level of correction below buying a smaller impeller from the pump manufacturer. But in many cases, the next smaller size impeller is too small for the pump load. Also, smaller impellers may not be available for the pump size in question and impeller trimming is the only practical alternative short of replacing the entire pump/motor assembly. (see Figures 6.20 & 6.21 for before and after impeller trimming).

Impeller trimming reduces tip speed, which in turn directly lowers the amount of energy imparted to the system fluid and lowers both the flow and pressure generated by the pump.

The Affinity Laws, which describe centrifugal pump performance, provide a theoretical relationship between impeller size and pump output (assuming constant pump speed):

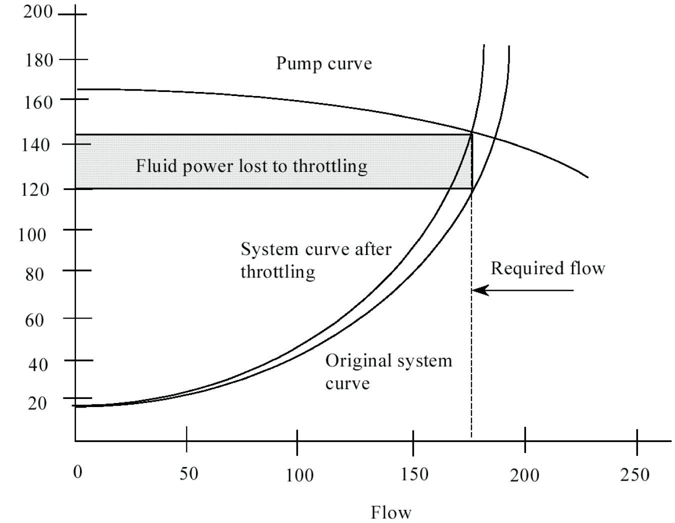


Figure 6.20 Before Impeller trimming

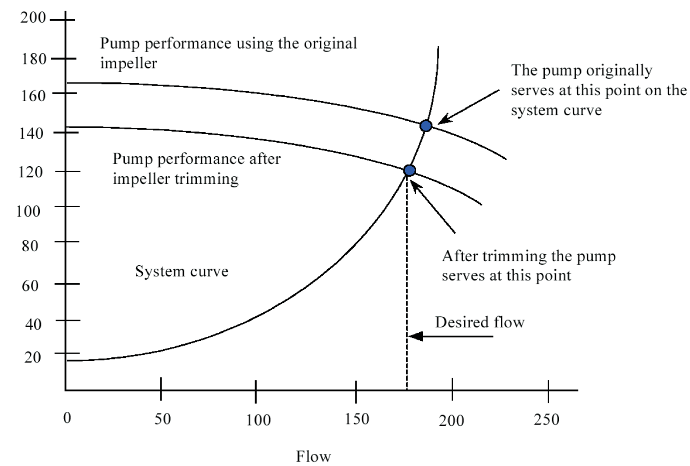


Figure 6.21 After Impeller Trimming

Where:

Q = flow

H = head

BHP = brake horsepower of the pump motor

Subscript 1 = original pump,

Subscript 2 = pump after impeller trimming

D = Diameter

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

$$H_2 = \left[\frac{D_2}{D_1}\right]^2 H_1$$

$$BHP_2 = \left[\frac{D_2}{D_1}\right]^3 BHP_1$$

Trimming an impeller changes its operating efficiency, and the non-linearities of the Affinity Laws with respect to impeller machining complicate the prediction of pump performance. Consequently, impeller diameters are rarely reduced below 70 percent of their original size.

Meeting variable flow reduction

Variable Speed Drives (VSDs)

In contrast, pump speed adjustments provide the most efficient means of controlling pump flow. By reducing pump speed, less energy is imparted to the fluid and less energy needs to be throttled or bypassed. There are two primary methods of reducing pump speed: multiple-speed pump motors and variable speed drives (VSDs).

Although both directly control pump output, multiple-speed motors and VSDs serve entirely separate applications. Multiple-speed motors contain a different set of windings for each motor speed; consequently, they are more expensive and less efficient than single speed motors. Multiple speed motors also lack subtle speed changing capabilities within discrete speeds.

VSDs allow pump speed adjustments over a continuous range, avoiding the need to jump from speed to speed as with multiple-speed pumps. VSDs control pump speeds using several different types of mechanical and electrical systems. Mechanical VSDs include hydraulic clutches, fluid couplings, and adjustable belts and pulleys. Electrical VSDs include eddy current clutches, wound-rotor motor controllers, and variable frequency drives (VFDs). VFDs adjust the electrical frequency of the power supplied to a motor to change the motor's rotational speed. VFDs are by far the most popular type of VSD.

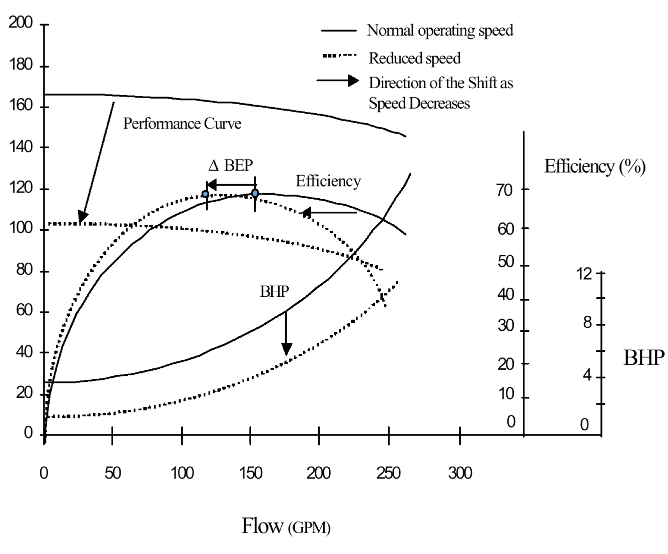


Figure 6.22 Effect of VFD

However, pump speed adjustment is not appropriate for all systems. In applications with high static head, slowing a pump risks inducing vibrations and creating performance problems that are similar to those found when a pump operates against its shutoff head. For systems in which the static head represents a large portion of the total head, caution should be used in deciding whether to use VFDs. Operators should review the performance of VFDs in similar applications and consult VFD manufacturers to avoid the damage that can result when a pump operates too slowly against high static head.

For many systems, VFDs offer a means to improve pump operating efficiency despite changes in operating conditions. The effect of slowing pump speed on pump operation is illustrated by the three curves in Figure 6.22. When a VFD slows a pump, its head/flow and brake horsepower (BHP) curves drop down and to the left and its efficiency curve shifts to the left. This efficiency response provides an essential cost advantage; by keeping the operating efficiency as high as possible across variations in the system's flow demand, the energy and maintenance costs of the pump can be significantly reduced.

VFDs may offer operating cost reductions by allowing higher pump operating efficiency, but the principal savings derive from the reduction in frictional or bypass flow losses. Using a system perspective to identify areas in which fluid energy is dissipated in non-useful work often reveals opportunities for operating cost reductions.

For example, in many systems, increasing flow through bypass lines does not noticeably impact the backpressure on a pump. Consequently, in these applications pump efficiency does not necessarily decline during periods of low flow demand. By analyzing the entire system, however, the energy lost in pushing fluid through bypass lines and across throttle valves can be identified.

Another system benefit of VFDs is a soft start capability. During startup, most motors experience in-rush currents that are 5 – 6 times higher than normal operating currents. This high current fades when the motor spins up to normal speed. VFDs allow the motor to be started with a lower startup current (usually only about 1.5 times the normal operating current). This reduces wear on the motor and its controller.

6.7 Energy Conservation Opportunities in Pumping Systems

- Ensure adequate NPSH at site of installation
- Ensure availability of basic instruments at pumps like pressure gauges, flow meters.
- Operate pumps near best efficiency point.
- Modify pumping system and pumps losses to minimize throttling.
- Adapt to wide load variation with variable speed drives or sequenced control of multiple units.
- Stop running multiple pumps - add an auto-start for an on-line spare or add a booster pump in the problem area.
- Use booster pumps for small loads requiring higher pressures.
- Increase fluid temperature differentials to reduce pumping rates in case of heat exchangers.
- Repair seals and packing to minimize water loss by dripping.
- Balance the system to minimize flows and reduce pump power requirements.
- Avoid pumping head with a free-fall return (gravity); Use siphon effect to advantage:
- Conduct water balance to minimise water consumption
- Avoid cooling water re-circulation in DG sets, air compressors, refrigeration systems, cooling towers feed water pumps, condenser pumps and process pumps.

- In multiple pump operations, carefully combine the operation of pumps to avoid throttling
- Provide booster pump for few areas of higher head
- Replace old pumps by energy efficient pumps
- In the case of over designed pump, provide variable speed drive, or downsize / replace impeller or replace with correct sized pump for efficient operation.
- Optimise number of stages in multi-stage pump in case of head margins
- Reduce system resistance by pressure drop assessment and pipe size optimisation

QUESTIONS	
1.	What is NPSH of a pump and effects of inadequate NPSH?
2.	State the affinity laws as applicable to centrifugal pumps?
3.	Explain what do you understand by static head and friction head?
4.	What are the various methods of pump capacity control normally adopted?
5.	Briefly explain with a diagram the energy loss due to throttling in a centrifugal pump.
6.	Briefly explain with a sketch the concept of pump head flow characteristics and system resistance.
7.	What are the effects of over sizing a pump?
8.	If the speed of the pump is doubled, power goes up by a) 2 times b) 6 times c) 8 times d) 4 times
9.	How does the pump performance vary with impeller diameter?
10.	State the relationship between liquid kW, flow and pressure in a pumping application.
11.	Draw a pump curve for parallel operation of pumps (2 nos).
12.	Draw a pump curve for series operation of pumps (2 nos).
13.	List down few energy conservation opportunities in pumping system.

REFERENCES

1. British Pump Manufacturers' Association
2. BEE (EMC) Inputs
3. PCRA Literature