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 is 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 analyzed 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 figure 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*:

$$\mathbf{Q \propto N}$$

$$\mathbf{H \propto N^2}$$

$$\mathbf{P \propto N^3}$$

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

$Q_1 / Q_2 = N_1 / N_2$

Example: $100 / Q_2 = 3000/1500$

$Q_2 = 50\ m^3/hr$

Head: Head is proportional to the square of speed

$H_1/H_2 = N_1^2 / N_2^2$

Example: $100 /H_2 = 3000^2 / 1500^2$

$H_2 = 25\ m$

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

$kW_1 / kW_2 = N_1^3 / N_2^3$

Example: $40/kW_2 = 3000^3 / 1500^3$

$kW_2 = 5\ kW$

As can be seen from the above laws, reduction in speed will result in considerable 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.

Points of equal efficiency on the curves for three 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.

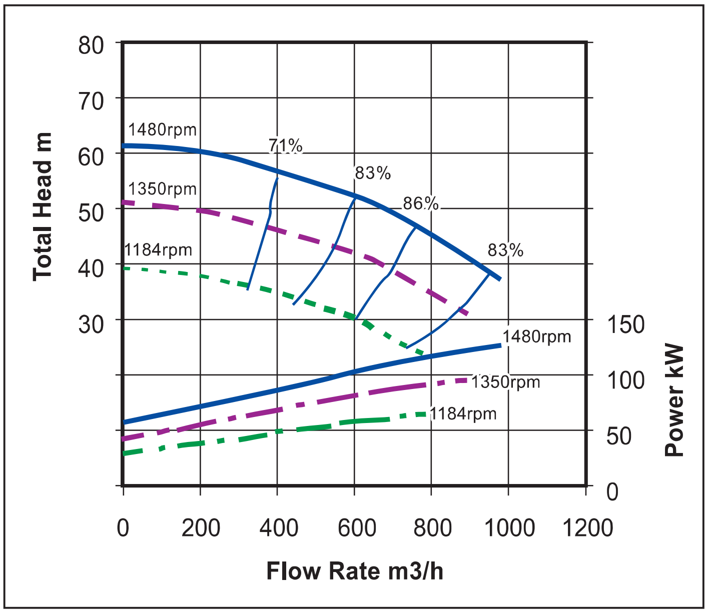

Figure 6.13 Example of Speed Variation Effecting Centrifugal Pump Performance

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.

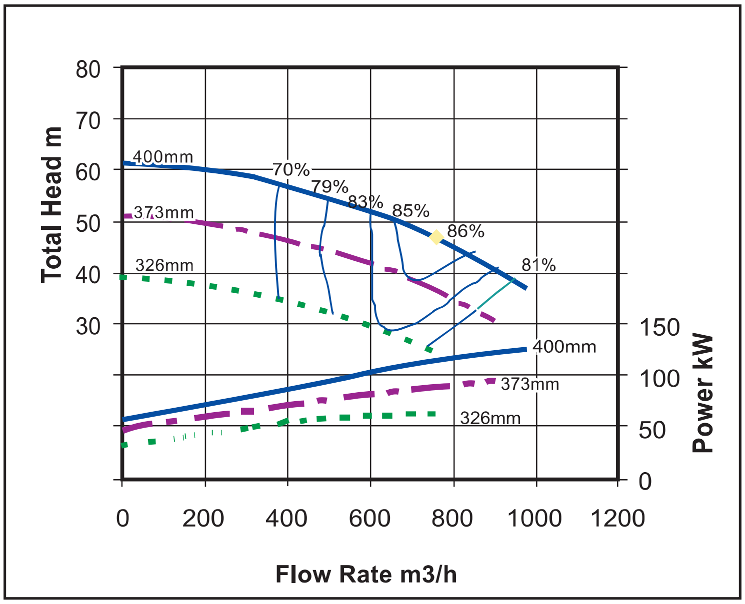

Figure 6.14 Example: Impeller Diameter Reduction on Centrifugal Pump Perfomance

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

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 liquid pressure at the eye of pump 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 cavitation 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 by which 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.

$$\mathbf{Q \propto N}$$

$$\mathbf{NPSHR \propto N^2}$$

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 the speed variation changes the duty point, the pump and system curves are overlaid. 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.

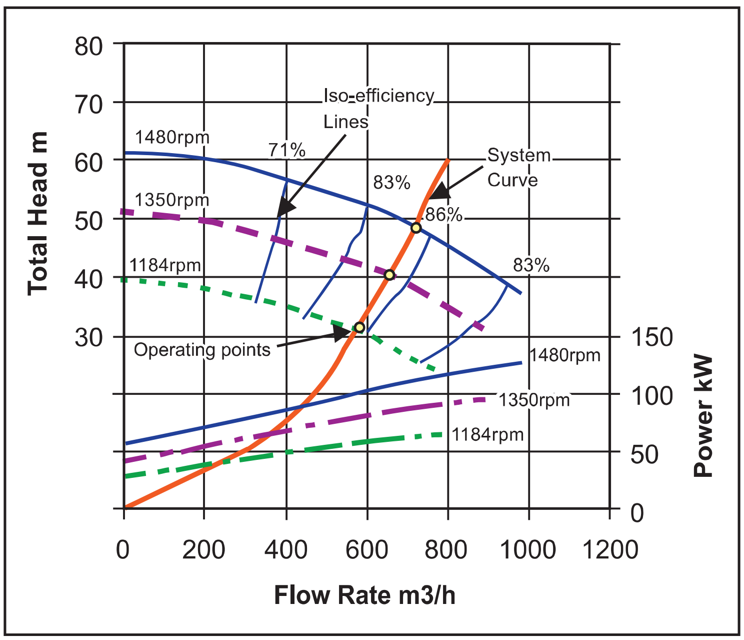

Figure 6.15 Example of the Effect of Pump Speed Change in a System with Only Friction Loss