

**Nuclear Training Course 23001  
(NEIT 230.1)**

# **Mechanical Equipment**

**September 1994**

# MECHANICAL EQUIPMENT

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- 1 ..... Centrifugal Pumps and Systems
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## ABSTRACT OF CURRENT REVISION

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R-3. Module 3 was deleted from the course. To avoid duplication of materials, the topic of boiler safety valves is now covered fully in the 23004 Turbine & Auxiliaries course.

**Note:**

The R-2 version of the course was approved for use, but not issued. The issue of R-2 was withheld to avoid the expense of a course issue when R-3 and the 23004 course revisions were imminent.

*Sept 1994*

## ABSTRACT OF PREVIOUS REVISIONS

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R-2. Minor revision to improve clarity, and minor technical errors. Highlight of major changes are as follows:

- Module 1, Net positive Suction Head, clarification to explanation of NPSH<sub>A</sub>.
- Module 1, Series Mounted Pumps in a Closed Loop System, clarification to explanation.
- Module 1, Pumps in Series and Parallel, clarification to explanation.
- Module 1, Added additional labels to Figure 1.28.
- Module 2, Added statement about the impact of machine wear.
- Module 3, Clarified paragraph 2, boiler safety valve capacity.

*Feb 1994*

Revised by: N. Ritter

## NOTES &amp; REFERENCES

*July 1993*

R-1. Complete revision to update course, reflecting operator job family requirements. Added abstract of revisions.

- Previous module 230.10-1, Centrifugal Pumps, completely rewritten.
- Previous module 230.11-1, Compressors, separated contents. Relevant material revised. Relevant remaining material transferred to 330.1 course.
- Module 3, Steam Generator Safety Valves, new module added.
- Previous module 230.12-1, Lubrication, transferred to 330.1.
- Previous modules 230.13-1,2,3,4 Shaft Couplings, Belt Drives, Chain Drives, Gears and Gearing, deleted.
- Previous module 230.14-1, Refrigeration, transferred to 330.1.
- Previous module 230.15-1, Diesel Engines, deleted.
- Previous module 230.16-1, Gas turbines, transferred to 330.1.

Revised by: D. Bieman, R. Harding, N. Ritter

*Dec 1972*

R-0. Original Issue.

Prepared by: L. Laplante

## Module 1

# CENTRIFUGAL PUMP AND SYSTEMS

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### OBJECTIVES:

After completing this module, you will be able to:

- |     |  |                    |
|-----|--|--------------------|
| 1.1 | a) Describe how pump head varies with pump capacity for both axial and radial flow pumps.  | <i>Page 4</i>      |
|     | b) Describe how pump efficiency varies with pump capacity.   | <i>Pages 4-5</i>   |
|     | c) Describe how pump power varies with capacity, and state the effect these characteristics may have on start-up technique.                              | <i>Pages 5-6</i>   |
| 1.2 | a) Describe Net Positive Suction Head Required (NPSH <sub>R</sub> ).   | <i>Pages 6, 11</i> |
|     | b) Describe the term Net Positive Suction Head Available (NPSH <sub>A</sub> ).   | <i>Page 12</i>     |
|     | c) State the desired relationship between NPSH <sub>R</sub> and NPSH <sub>A</sub> and explain the consequence of not achieving the desired relationship. | <i>Page 13</i>     |
| 1.3 | Consider a typical centrifugal pump, operating in a liquid system. Explain the effect of each of the following parameters on the pump capacity:          |                    |
|     | a) Discharge tank level/pressure;  | <i>Page 9</i>      |
|     | b) Suction tank level/pressure;  | <i>Pages 9-10</i>  |
|     | c) Fluid friction losses in the system.  | <i>Pages 10-11</i> |
| 1.4 | Explain how the tendency of a centrifugal pump to cavitate is affected by:   |                    |
|     | a) Discharge tank level/pressure;  | <i>Page 15</i>     |
|     | b) Throttling in the discharge piping;   | <i>Page 16</i>     |
|     | c) Suction tank level/pressure;  | <i>Page 17</i>     |
|     | d) Throttling in the suction piping;   | <i>Pages 17-18</i> |
|     | e) Fluid temperature at the pump inlet;  | <i>Page 19</i>     |
|     | f) Changing pump speed.  | <i>Page 20</i>     |
| 1.5 | Consider an arrangement of two centrifugal pumps in series. Explain the effect of starting up or shutting down one pump on:                              |                    |
|     | a) System head;  | <i>Pages 22-23</i> |
|     | b) System flow;  | <i>Pages 22-23</i> |
|     | c) Tendency to cavitate.   | <i>Pages 24-25</i> |

## NOTES &amp; REFERENCES

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|---|--|
| <p><i>Page 27</i></p> <p><i>Page 27</i></p> <p><i>Pages 27, 28</i></p><br><p><i>Page 31-32</i></p> <p><i>Pages 31-32</i></p> <p><i>Page 33</i></p> <p><i>Page 34</i></p><br><p><i>Page 34</i></p><br><p><i>Page 35</i></p><br><p><i>Pages 35-36</i></p><br><br><p><i>Page 36</i></p> <p><i>Page 36</i></p> <p><i>Pages 36-37</i></p> <p><i>Pages 37-38</i></p> <p><i>Page 38</i></p> <p><i>Pages 37-38</i></p><br><p><i>Pages 38-39</i></p><br><br><p><i>Page 39</i></p><br><p><i>Page 39</i></p> | <p>1.6 Consider an arrangement of two identical centrifugal pumps in parallel. Explain the effect of starting or stopping one pump on:</p> <ul style="list-style-type: none"> <li>a) System head;</li> <li>b) System flow;</li> <li>c) Tendency to cavitate.</li> </ul> <p>1.7 Consider an arrangement of a CANDU heat transport system with four running main circulation pumps. Explain the effect of tripping a single pump on:</p> <ul style="list-style-type: none"> <li>a) System flow;</li> <li>b) System head.</li> <li>c) Tendency to cavitate.</li> </ul> <p>1.8 a) Describe two examples of operating practices used to prevent cavitation.</p> <p>b) Describe two examples of operating practices used to prevent air locking.</p> <p>1.9 Describe two typical indications of severe cavitation (vapour locking) or air locking.</p> <p>1.10 Describe three examples of general operating practices used to prevent water hammer.</p> <p>1.11 Explain the indicated number of reason(s) for each of the following prerequisites for starting a centrifugal pump:</p> <ul style="list-style-type: none"> <li>a) Pump/suction piping is primed (1);</li> <li>b) Suction isolating valves fully open (1);</li> <li>c) Discharge valve closed (2);</li> <li>d) Pump lubrication system in service (1);</li> <li>e) Pump gland seal liquid supplied (2).</li> </ul> <p>1.12 Explain why a centrifugal pump overheats when operating at low capacity, and state three operating actions that will prevent overheating.</p> <p>1.13 Explain two methods of avoiding thermal shock in pumps which must be started in high temperature systems.</p> <p>1.14 With respect to the isolation of a centrifugal pump in parallel where one pump has been shut down and the other pump is operating:</p> <ul style="list-style-type: none"> <li>a) Describe the procedure necessary to safely isolate the shutdown pump;</li> <li>b) Explain the possible consequences if the pump is improperly isolated.</li> </ul> |
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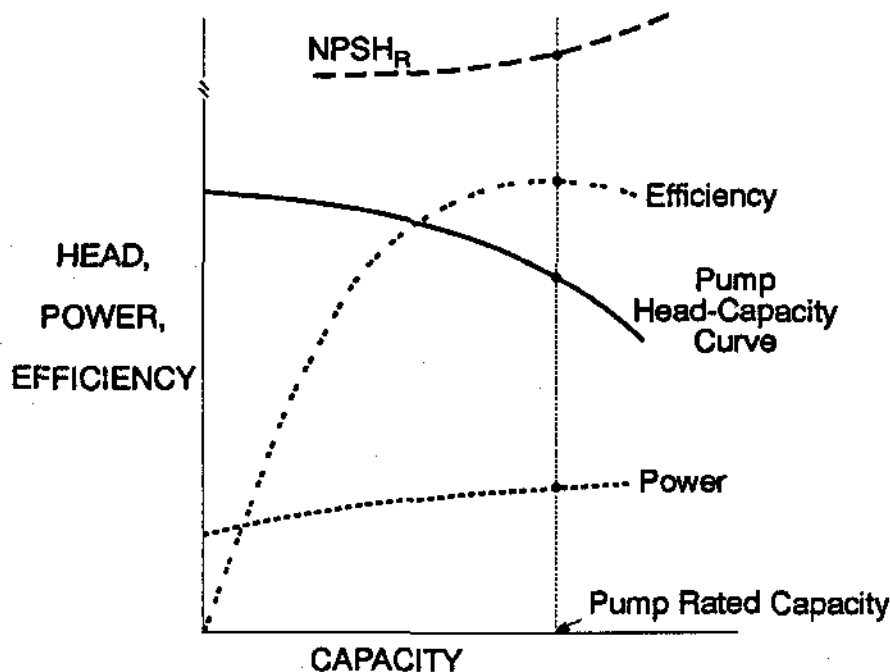
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## INTRODUCTION

A centrifugal pump can be described as one which uses a rotating impeller to add kinetic energy to a liquid, giving it the ability to transfer from one tank to another or to circulate through a closed loop system. In the earlier levels of the Mechanical Equipment course, we discussed the reasons why centrifugal pumps are the ones best suited for use in power stations and described various constructions of these pumps. In this module we will discuss in more detail, the operating characteristics of centrifugal pumps. We will also discuss how pumping system arrangements and operating conditions affect the ability of a pump to provide satisfactory service. We will also describe the general operating practices which promote pump and system life and reliability.

## PUMP OPERATING CHARACTERISTICS

The manufacturer of a pump will provide a set of curves to describe a centrifugal pump's performance at one particular operating speed. General examples of these curves are shown in Figure 1.1 below. Figure 1.2, the foldout diagram at the end of the module illustrates actual pump curves for a DNGS primary heat transport (PHT) pump. The curves in both figures show changes in a given parameter as a function of capacity.



**FIGURE 1.1**  
**TYPICAL CENTRIFUGAL PUMP PERFORMANCE**  
**CURVES**

## NOTES &amp; REFERENCES

**Obj. 1.1 a)**

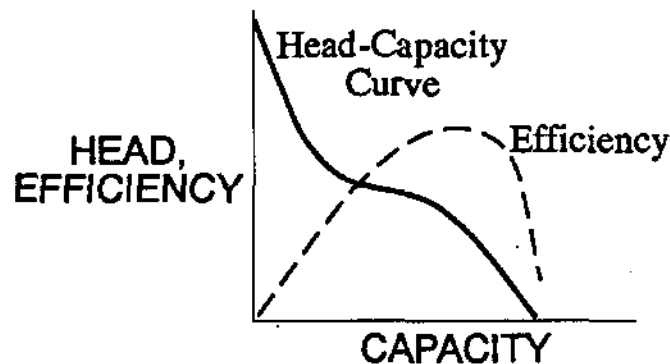
Capacity is usually stated as a volumetric flowrate such as litres per second, or gallons per minute.

Different type of impellers used were discussed in earlier Mechanical Equipment courses.

**Pump Head-Capacity**

In discussing fluid flow, the term **Head** is frequently used. Pump head refers to the energy content per unit weight that a pump is capable of transferring to the liquid. The units are usually metres in the metric system. Notice, from Figure 1.1 and 1.2, that the pump head-capacity\* curve droops with increasing capacity. As the flow through a centrifugal pump increases, it develops less head because of increased friction losses and turbulence within the pump.

The shape of the head capacity curve is determined by the type of pump impeller used. Figures 1.1 and 1.2 show head-capacity curves typical of a radial flow design. As the impeller shape becomes more axial, the droop in the head capacity curve becomes steeper. The head-capacity curve in Figure 1.3 shows an example of an axial impeller where the curve flattens out over a small range of capacity. Operating in this flattened capacity range will result in instability (ie. small fluctuations in operating head result in large changes in capacity), indicated by heavy surges in flow and vibration. Large mixed flow and axial flow pumps such as the low pressure service water and condenser circulating water pumps may be unstable at certain capacities, hence operation in these unstable regions must be avoided.



**FIGURE 1.3**  
**HEAD-CAPACITY & EFFICIENCY-CAPACITY CURVE**  
**FOR AN AXIAL FLOW PUMP**

**Pump Efficiency****Obj. 1.1 b)**

The ability of a pump to convert the mechanical energy of a rotating shaft into pressure and kinetic energy of a flowing liquid (i.e. the ratio of output to input energy or power) is called "pump efficiency" and is a function of capacity. The shape of the efficiency curve shown in Figures 1.1 and 1.3 is typical of all types of centrifugal pumps, although the peak efficiency tends to fall as the impeller design changes from radial flow to axial flow. At no flow conditions, the pump is at zero efficiency since output power is zero. In any pump efficiency curve, the pump becomes more efficient at transferring power as flow increases, until a maximum efficiency is reached. As flow increases further, the pump becomes less efficient at transferring power to the fluid.



Some of the shaft energy (power) is "lost" in overcoming bearing and packing friction and turbulence in the liquid. Friction losses are relatively small and constant, but as pump capacity increases, losses due to turbulence gradually decrease to a minimum value, then increase as flowrate increases further.

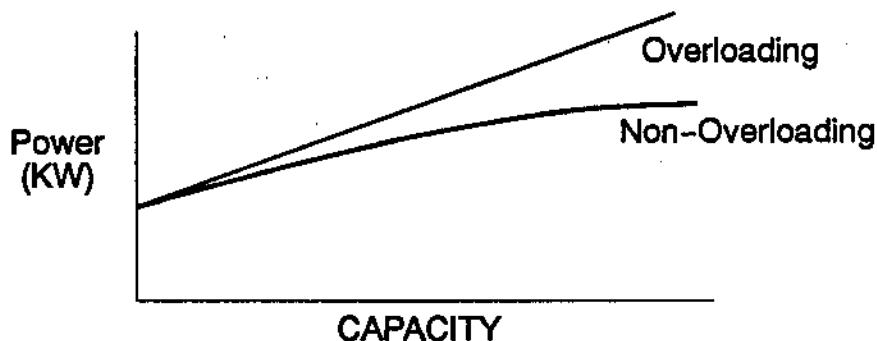
The **rated capacity** of a pump is defined as the pump capacity at its peak efficiency, or as the capacity at which pump losses are a minimum. Pumps are selected to operate at their rated capacity whenever possible, however the top of the efficiency curve tends to be flattened and reasonably large changes in capacity can usually be made without much reduction in efficiency.

### Pump Power

Power supplied to a pump impeller shaft is described as shaft power, and is measured in kilowatts. Pump power curves indicate how shaft power requirements change with capacity. The shape of pump power curves depend upon the type of centrifugal pump impeller.

For radial flow pumps, minimum power occurs at zero flow and increases as capacity increases. Typical power characteristics of radial flow impellers are shown in Figure 1.4. which indicates they may be described as "overloading" or "non-overloading". Thus, the characteristic is "overloading" if the power continues to increase after the rated capacity is surpassed. Alternatively, if power levels off (ie reaches a peak value) after rated capacity is exceeded, it is called "non-overloading". Both characteristics are useful. The overloading type would initiate a pump motor trip if excessive flow occurs, whereas a non-overloading type avoids a pump trip on excessive flow. The latter situation is important if safe operation requires that flow must not stop under any conditions, as in the case of reactor cooling.

*Obj. 1.1 c)*



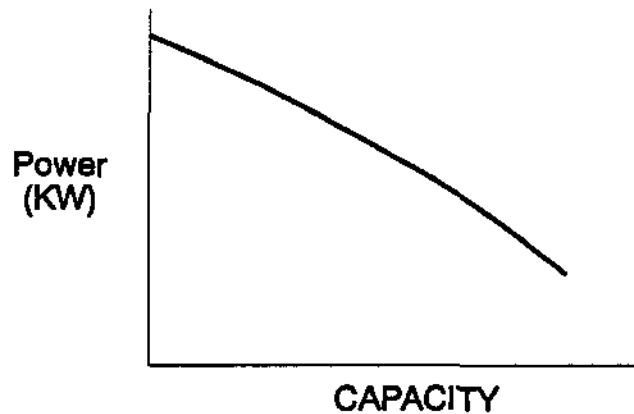
**FIGURE 1.4**  
**POWER CURVE FOR A TYPICAL RADIAL FLOW PUMP**

## NOTES &amp; REFERENCES

We prefer to start-up a radial flow pump with its discharge valve closed, because starting torque and hence motor current are minimal when flow is zero. However this procedure is not possible when there is no pump discharge valve, as is the case in the PHT systems.

For axial flow pumps, maximum power occurs at zero flow and decreases as capacity increases. A typical power characteristic for an axial flow pump is shown in Figure 1.5. This characteristic suggests that in order to minimize the duration of high power at start-up, the pump should be started with its discharge CV open. However, this procedure is not followed, because water hammer is of major concern during start-up conditions (discussed later in the "General Operating Practices"). Also, to minimize the power required, axial flow pumps are not usually operated continuously in a throttled condition.

In any pump application, the electrical power supply is sized to accommodate the largest demand, even if it exists only for a short time period.



**FIGURE 1.5**  
**POWER CURVE FOR A TYPICAL AXIAL FLOW PUMP**

### Net Positive Suction Head Required ( $NPSH_R$ )

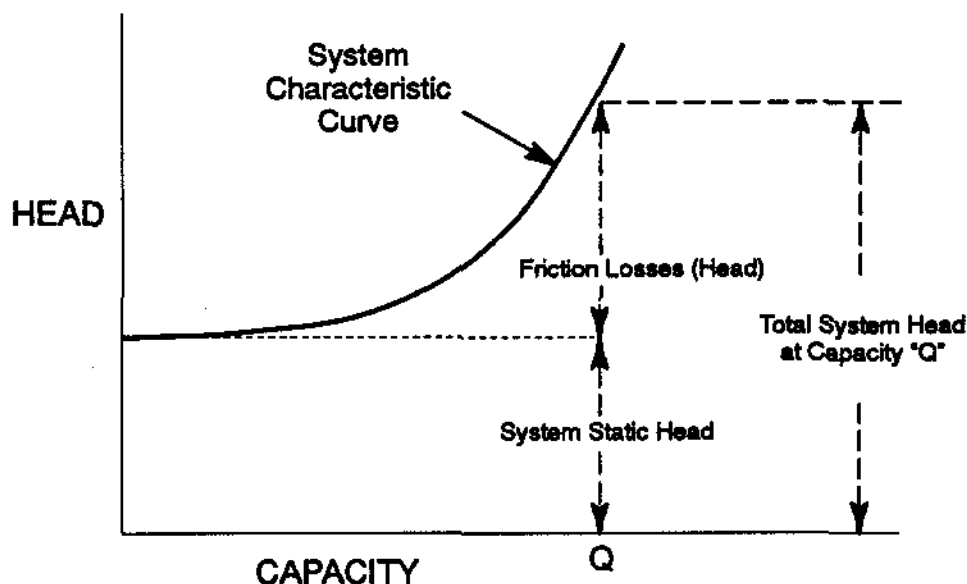
*Obj. 1.2 a)*

The  $NPSH_R$  curve is provided by the pump manufacturer to assist in avoiding cavitation in application of the pump. As shown in Figure 1.1,  $NPSH_R$  increases as capacity increases.

$NPSH_R$  is defined as the minimum amount of energy in excess of vapour pressure energy, that must be contained in the liquid as it enters the pump, in order to prevent cavitation occurring inside the pump. The topic of  $NPSH$  is discussed in more detail, later in the module.

## SYSTEM CHARACTERISTICS

Pumping systems in our stations are most often used to transfer water from one area to another (eg. the Feedwater System), or circulate water in a closed loop (eg. the Primary Heat Transport System). A pumping system characteristic illustrates the energy (head) that must be added to a liquid to ensure it flows through the system at a specified flow rate. Figure 1.6, a foldout diagram, shows a liquid transfer system, and Figure 1.7 illustrates a system head curve for various flows in the system. The curve has two main components, namely the **System Static Head**, and **System Friction Head**.



**FIGURE 1.7**  
**TYPICAL SYSTEM CHARACTERISTIC CURVE**

### System Static Head

From Figure 1.7, the system static head gives the head required at the zero flow condition. The system static head may be composed of a lift component and a pressure difference component. The lift component will be a factor when the discharge tank liquid level is higher than the suction tank liquid level. The pressure difference component will be a factor when the suction and discharge tanks are pressurized at different pressures. For example, the Condensate Extraction Pumps transfer water from the condenser, below atmospheric pressure, to the deaerator, which operates above atmospheric pressure and at a much higher elevation. In this case, the system static head would involve a lift component and a pressure difference component.

## NOTES &amp; REFERENCES

**System Static Head = Pressure Head + Elevation Head**

or mathematically expressed as:

$$= \frac{\Delta P}{\rho g} + \Delta H \quad \text{meters}$$

Where:  $\Delta P$  = Discharge tank pressure ( $P_d$ ) – Suction tank pressure ( $P_s$ ),

$\rho$  is the density of the fluid;

$g$  is the acceleration due to gravity;

$\Delta H$  = Discharge tank level elevation – Suction tank level elevation;

In closed loop systems, the static head is eliminated because the liquid always returns to its starting pressure and elevation.

### System Friction Head

The piping lengths and diameter, the fittings and valves, all offer resistance to the flow of water through a system. It turns out that the energy required by the water to overcome the system's total resistance, or in other words, the head loss in the system due to friction, is proportional to the flow velocity squared. This characteristic is called the System Friction Head ( $h_f$ ) and can be expressed by the relationship:

$$\text{Friction Head} = h_f = C_L \frac{v^2}{2g}$$

The ( $v^2$ ) term causes the curve to get steeper as capacity increases. The "loss coefficient" ( $C_L$ ) represents the sum of all system resistances and remains constant unless the system configuration changes.

Note that the frictional head loss relationship is also applicable to any part of the total system, eg. the Suction System or the Discharge System, as we shall see in a later section.

The friction head is added to the total static head to form the system head curve. This is illustrated graphically in Figure 1.7.

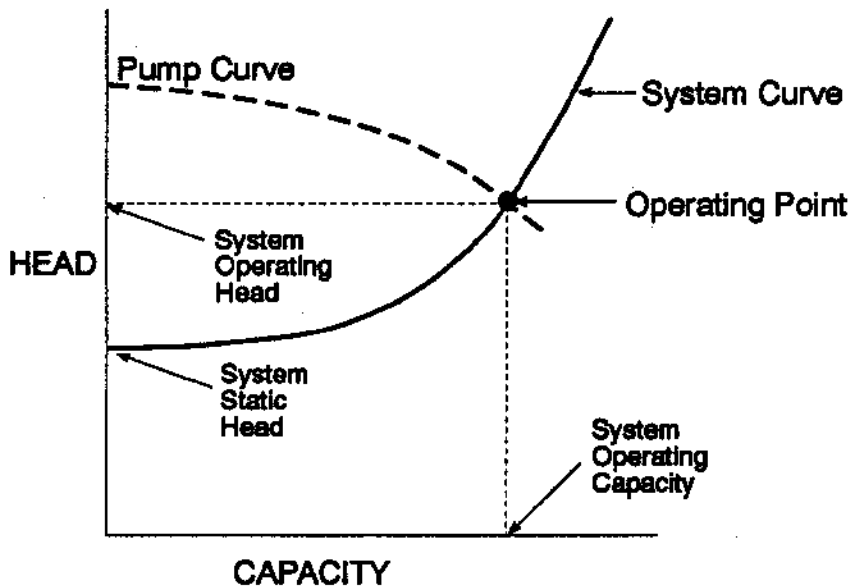
### Operating Point

The system head curve, described above, represents the energy the liquid needs to flow through the system at any flowrate. The pump head–capacity curve represents the ability of the pump to supply energy to the liquid at any flow rate. The intersection of the system head curve and the pump head–capacity curve gives the operating point (capacity) of the system. At this point, the pump provides sufficient energy (head) to meet the needs of the system at a particular capacity. The operating point is illustrated in Figure 1.8.

The pump curve shape is constant for a given pump, and its position remains fixed if pump speed is constant, which is usually the case.

Consequently, system flowrate is adjusted by altering either the shape or position of the system curve. Three ways of changing the system curve, and hence the operating point, will be discussed, namely:

- i) Discharge tank level or pressure
- ii) Suction tank level or pressure
- iii) System friction losses.



**FIGURE 1.8  
SYSTEM OPERATING POINT**

### Changing Discharge Tank Level or Pressure

By increasing the discharge tank level or pressure, the system static head will increase. This can be visualized from the fold out diagram Figure 1.6. Since the system friction loss should remain the same for any given flow, the slope of the new system curve will remain the same but is shifted slightly higher. By shifting the system head curve upward, the operating point moves upward on the pump curve to a lower capacity as illustrated in Figure 1.9. Thus by increasing the discharge tank level or pressure, there will be a reduction in pump capacity. Similarly a decrease in discharge tank level or pressure will increase pump capacity.

*Obj. 1.3 a)*

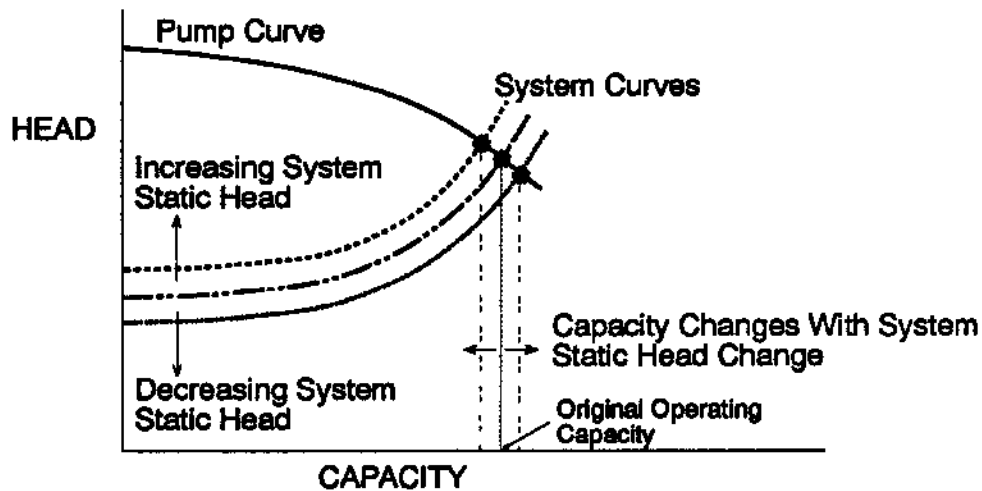
### Changing Suction Tank Level or Pressure

If the suction tank level or pressure increases, the system static head will decrease. This can be visualized from the fold out diagram Figure 1.6. The new system head curve will shift lower but retain the same slope. The net effect on pump operation is that the operating point moves further along on the pump curve, resulting in an increase in pump capacity. Lowering the pump suction tank level or pressure will have the opposite effect. (ie. system static head will increase and the capacity will decrease.

*Obj. 1.3 b)*

## NOTES &amp; REFERENCES

The effect on system capacity, due to suction tank static head or pressure changes is also included in Figure 1.9. However, the influence of changing suction static conditions in the overall system static head is usually quite small.



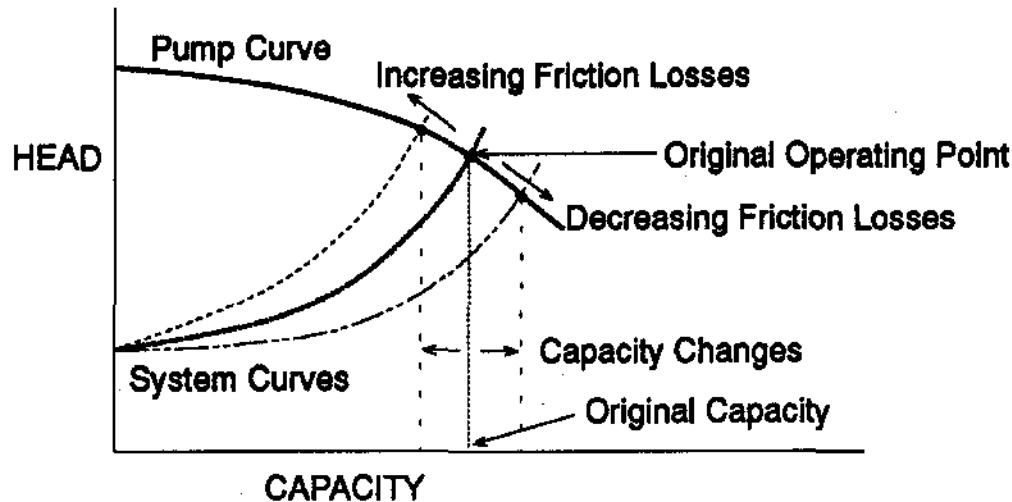
**FIGURE 1.9**  
**EFFECT OF CHANGING SYSTEM**  
**STATIC HEAD ON CAPACITY**

### Changing System Friction Head

*Obj. 1.3 c)*

Friction head losses may occur on either the suction side or the discharge side of the pump. By examining the sketch of a typical water transfer system on the foldout diagram, Figure 1.6, we can identify various components capable of changing friction losses. A change in the valve opening of either the suction or discharge valve, or accumulation of dirt in the suction strainer, will change the system flow by changing system flow resistance. Since the system static head is unaffected, a change in the friction head will change the slope of the system curve as shown in Figure 1.10.

An increase in friction losses (eg. resulting from valve throttling action) will shift the pump operating point to a lower capacity. A decrease in friction loss (caused by opening a throttled valve) will shift the operating point to a higher capacity. The effects of increasing friction losses on the suction side of the pump will be discussed when the topic of net positive suction head is considered.



**FIGURE 1.10**  
**EFFECT OF CHANGING FRICTION LOSSES ON THE**  
**SYSTEM CURVE**

## SUCTION SYSTEM OPERATING CONDITIONS

Suction system operating conditions must always allow the liquid to reach the pump inlet at the highest possible pressure, in order to minimize the possibility of cavitation occurring inside the pump. Suction system operating conditions control the magnitude of liquid energy as it enters the pump, which in turn establishes the liquid's margin to cavitation. This parameter is known as the Net Positive Suction Head.

### Net Positive Suction Head (NPSH), (NPSH<sub>R</sub>) and (NPSH<sub>A</sub>).

NPSH is generally defined as the difference between the pressure plus velocity energies of the liquid (ie, the Pump Suction Head) and its vapour pressure energy at the flowing temperature. Cavitation will occur when the NPSH falls to zero. That is, when the liquid's energy becomes equal to the energy level at which boiling occurs.

In practice, liquid pressure and velocity is measured at the pump inlet flange, because it is the interface between the suction piping and pump. But further energy losses occur inside the pump, downstream of this location, causing the liquid's energy to approach the energy level where cavitation would be initiated. Consequently, the pump manufacturer measures the friction losses in the pump suction passages, over a range of capacities. These energy losses are then known, and values of NPSH<sub>R</sub> are provided in the form of a curve, as shown on Figure 1.1.

Given an NPSH<sub>R</sub> curve specified by the pump manufacturer, the suction system must be designed and operated to ensure the NPSH of the liquid entering the suction piping/pump interface, is larger than the NPSH<sub>R</sub>.

*Obj. 1.2 a)*  
*(continued)*

## NOTES &amp; REFERENCES

*Obj. 1.2 b)*

The NPSH resulting from system configuration and operation is called the NPSH available ( $NPSH_A$ ). Thus to avoid cavitation inside the pump,  $NPSH_A$  must be larger than  $NPSH_R$ . Note that  $NPSH_R$  is an empirical quantity, obtained from the pump manufacturer's tests. However, because  $NPSH_A$  is a function of the suction system, it can be expressed mathematically and the general form is as follows:

$NPSH_A = \text{Pump Suction Head minus Liquid vapour Pressure Head.}$

or:

$$NPSH_A = \frac{P_s}{\rho g} + \frac{v_s^2}{2g} - \frac{P_{vap}}{\rho g}$$

Where  $P_s$  and  $P_{vap}$  are absolute pressures and ( $v_s$ ) is the average velocity at the pump inlet.

The vapour pressure ( $P_{vap}$ ) is dependent on the temperature of the liquid. For water, the vapour pressure is low at low temperatures, however above  $\approx 50^\circ\text{C}$  its magnitude becomes significant.

In systems which take liquid from a suction tank, conditions in the tank and in the suction piping can be used to calculate the  $NPSH_A$ . Conditions which affect the suction head are;

- i) pressure on the stored liquid surface ( $P_t$ ).
- ii) elevation difference between the stored liquid level and the centreline of the pump impeller ( $h$ ).
- iii) pressure drop due to flow in the suction piping ( $h_f$ ).

Using these conditions,  $NPSH_A$  can be expressed mathematically as follows;

$$NPSH_A = \frac{P_t}{\rho g} \pm h - h_f + \frac{v_s^2}{2g} - \frac{P_{vap}}{\rho g}$$

Note: assume  $v_s$  is constant throughout entire

Recall that ( $h_f$ ) can be expressed by the relationship;

$$h_f = C_L \frac{v_s^2}{2g} \quad \text{therefore;}$$

$$NPSH_A = \frac{P_t}{\rho g} \pm h - C_L \frac{v_s^2}{2g} + \frac{v_s^2}{2g} - \frac{P_{vap}}{\rho g}$$

Note that ( $h$ ) is positive when the liquid level is above the pump impeller, and negative when below the pump.

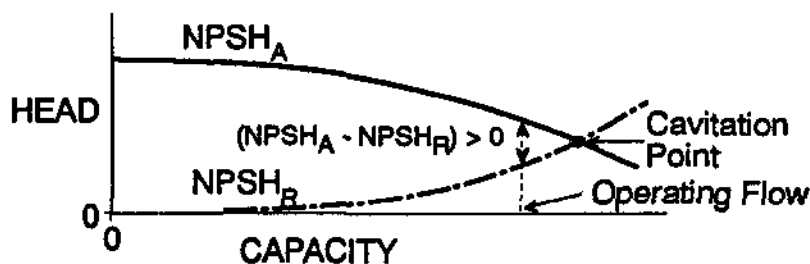
If velocity in the system is small (ie. friction and velocity components of suction head are small), then  $NPSH_A$  can be estimated mathematically as follows;

$$NPSH_A \approx \frac{P_t}{\rho g} \pm h - \frac{P_{vap}}{\rho g}$$



Since frictional effects have a large effect on the magnitude of both  $NPSH_A$  and  $NPSH_R$ , when plotted against capacity, their curves are not linear. As we see in Figure 1.1 the  $NPSH_R$  curve steepens as flowrate increases. On the other hand, for  $NPSH_A$ , the flow friction acts to reduce the static head components, hence the curve droops, with its steepness increasing as capacity increases. The characteristic shape of  $NPSH_A$  and  $NPSH_R$  versus Capacity curves are shown on Figure 1.11. We can see that when flowrate is zero,  $NPSH_R$  is zero and  $NPSH_A$  is a maximum. As capacity increases, the margin between the curves decreases, and the point of intersection indicates the capacity at which cavitation occurs. Points to the right of the intersection are indicative of severe cavitation (vapour-locking) conditions.

*Obj. 1.2 c)*



**FIGURE 1.11  
NPSH CURVES vs. FLOW  
FOR A TYPICAL PUMPING SYSTEM**

The question arises as to how relevant the concept of NPSH is to the jobs of operating personnel.

The concept of NPSH is of direct use to the system designer, because it provides information enabling him design the suction system to enable operating personnel to avoid cavitation problems.

For the operator, the concept of NPSH illustrates the conditions which can cause cavitation and suggests actions which prevent its onset.

### **SUMMARY OF THE KEY CONCEPTS**

- Centrifugal pump head generally has a maximum value at zero capacity and decreases as capacity increases. The characteristic slope is steeper for an axial impeller than for a radial type.
- A pump becomes more efficient at transferring power as flow increases until a maximum efficiency is reached. As flow increases further, the pump becomes less efficient at transferring power to the fluid.
- For radial flow centrifugal pumps, minimum power occurs at startup and increases as capacity increases. For axial flow centrifugal pumps, maximum power requirements occur at pump startup and decrease as pump capacity increases.

## NOTES &amp; REFERENCES

- If the power for a radial flow pump continues to increase beyond its rated capacity, the characteristic is called “overloading”. If, as capacity increases past the rated value, power levels off at a maximum value, the characteristic is called “non-overloading”.
- An increase in the discharge tank level or pressure, will shift the system curve higher. The operating point moves upward on the pump curve to a lower capacity. Similarly a decrease in discharge tank level or pressure will increase pump capacity.
- An increase in the suction tank level or pressure, decreases the system static head, shifting the system curve lower. The operating point moves further along on the pump curve, increasing capacity. Similarly a decrease in suction tank level or pressure will decrease pump capacity.
- An increase in friction losses will steepen the slope of the system curve, moving the operating point to a lower capacity. Similarly a decrease in friction losses will move the pump operating point to a higher capacity.
- $NPSH_R$  is an indication of the energy the liquid requires to overcome friction losses from the pump suction opening to the impeller.  $NPSH_R$  increases as capacity increases. Manufacturers supply this information by pump curves.
- $NPSH_A$  refers to the liquid energy at the inlet of the pump in excess of the vapour pressure energy.  $NPSH_A$  decreases as capacity increases.
- A pump must always operate with  $NPSH_A$  greater than  $NPSH_R$ , otherwise cavitation will occur.

*Page 41*

You can now do assignment questions 1-7.

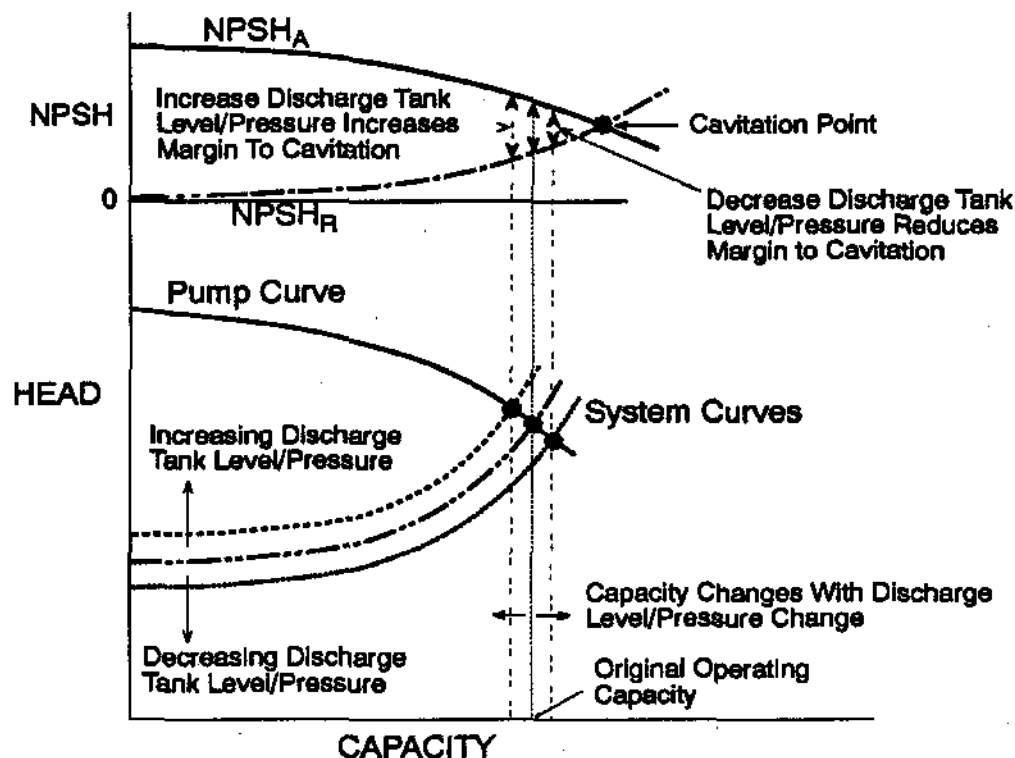
## FACTORS AFFECTING THE PROBABILITY OF CAVITATION

In this section, we will discuss how changes to the discharge system configuration, the suction system configuration, fluid temperature, and pump speed will affect the approach to cavitation conditions. To assist our discussions, will refer to Figure 1.6. and use the graphs, Figures 1.9 and 1.10 in conjunction with the  $NPSH_A$  and  $NPSH_R$  graphs (Figure 1.11). In addition, you will be able to relate the discussions to the mathematical expression for  $NPSH_A$  given previously on Page 1-12.

### Changing Discharge Tank Level or Pressure

By changing the discharge tank level or pressure, the system curve changes, but the NPSH curves are not affected. Figure 1.12 shows that as capacity increases (caused by a decrease in discharge tank level or pressure), the margin between the  $NPSH_A$  curve and the  $NPSH_R$  curve decreases. The pump now operates closer to cavitation conditions. Similarly, an increase in discharge tank level or pressure will cause the pump to operate at lower capacity, hence further from cavitation conditions.

*Obj. 1.4 a)*



**FIGURE 1.12**  
**EFFECT OF CHANGING DISCHARGE TANK LEVEL/  
 PRESSURE ON MARGIN TO CAVITATION**

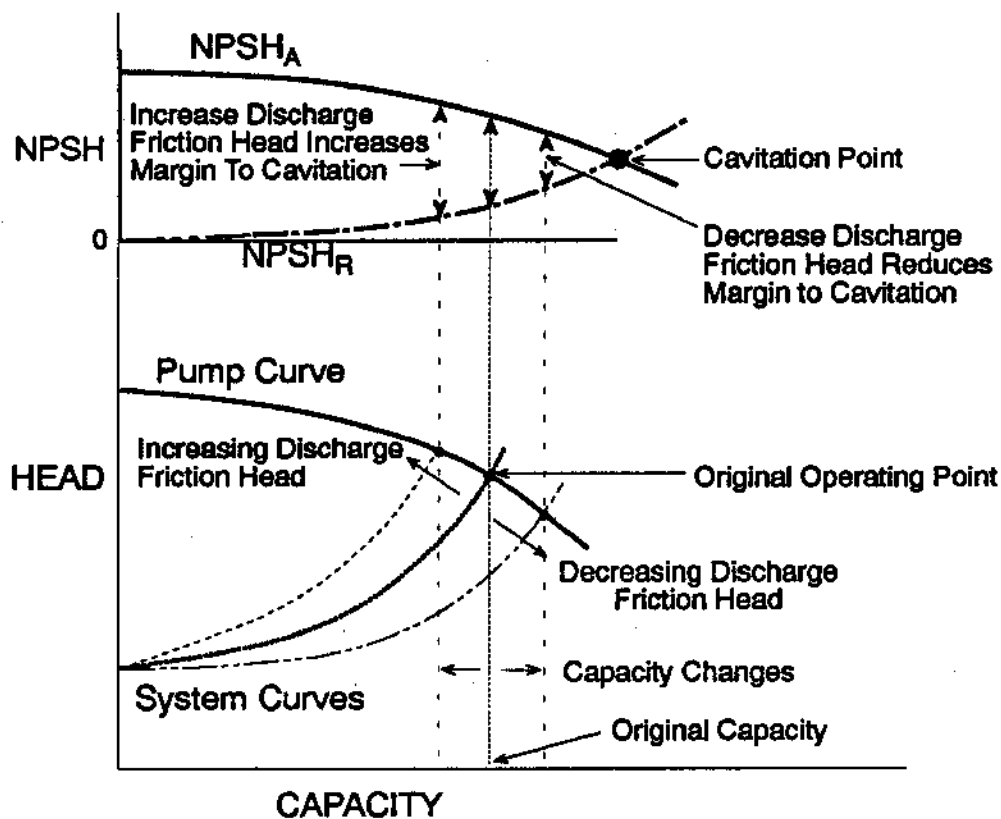
## NOTES &amp; REFERENCES

*Obj. 1.4 b)***Changing Discharge System Friction Head**

When the pump discharge valve is throttled, the system friction loss coefficient ( $C_L$ ) increases and the pump operating point shifts to a lower capacity. There is no effect on the NPSH curves, however, a shift to a lower capacity increases the margin to cavitation between the  $NPSH_A$  curve and the  $NPSH_R$  curve.

Similarly, as the pump discharge valve is opened, the margin to cavitation is reduced. These effects are shown in Figure 1.13.

Note that other modifications to the discharge system configuration such as a line blockage or introduction of a bypass flow will also affect the discharge friction head.



**FIGURE 1.13**  
**EFFECT OF CHANGING DISCHARGE FRICTION**  
**HEAD ON MARGIN TO CAVITATION**

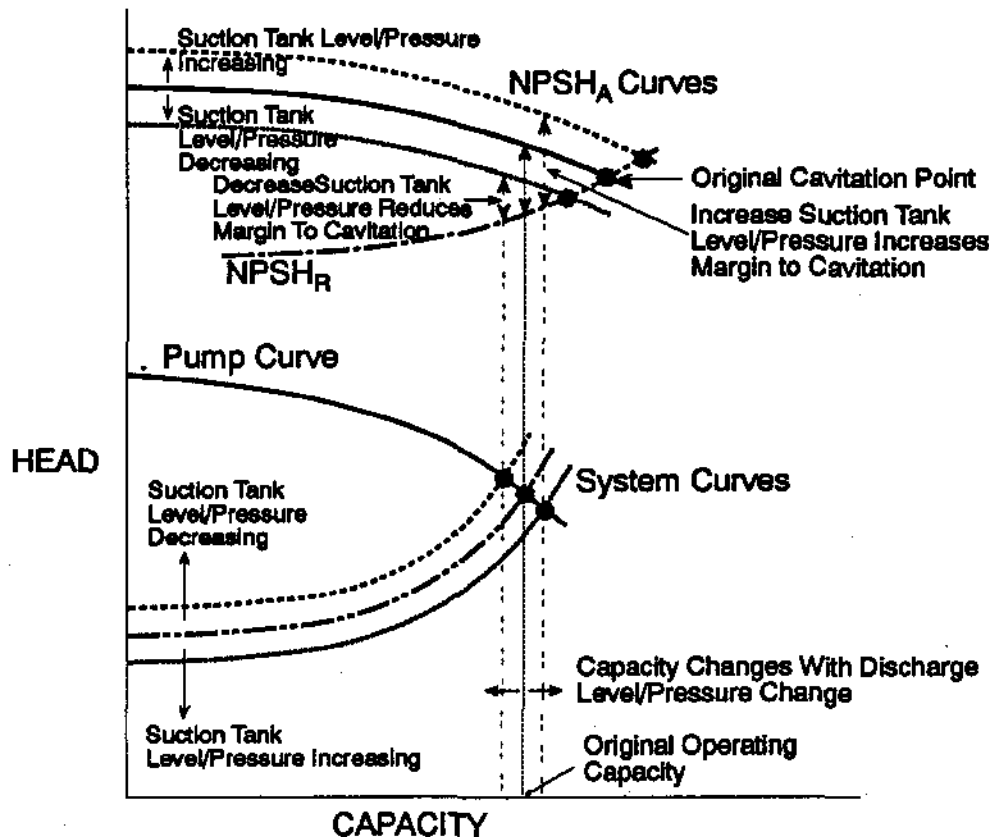
**Changing Suction Tank Level or Pressure**

Changing the suction tank level or pressure affects the overall system curve and the net positive suction head available (see Figure 1.14). Because only the static conditions are changed, the starting point of the  $NPSH_A$  curves will raise or lower but the curve shape will remain the same.

## NOTES &amp; REFERENCES

*Obj. 1.4 c)*

As the level or pressure in the suction tank falls, the margin to cavitation is reduced because the  $NPSH_A$  curve shifts lower. The reduction is offset somewhat by the beneficial effect of lower capacity at lower suction tank level or pressure conditions. Conversely, the margin to cavitation increases as the suction tank level or pressure increases. The increase is offset by the adverse effect of increased capacity. However, the effect on the system curve due to changes in suction system static conditions, is usually small.



**FIGURE 1.14**  
**EFFECT OF CHANGING SUCTION TANK**  
**LEVEL/PRESSURE ON MARGIN TO CAVITATION**

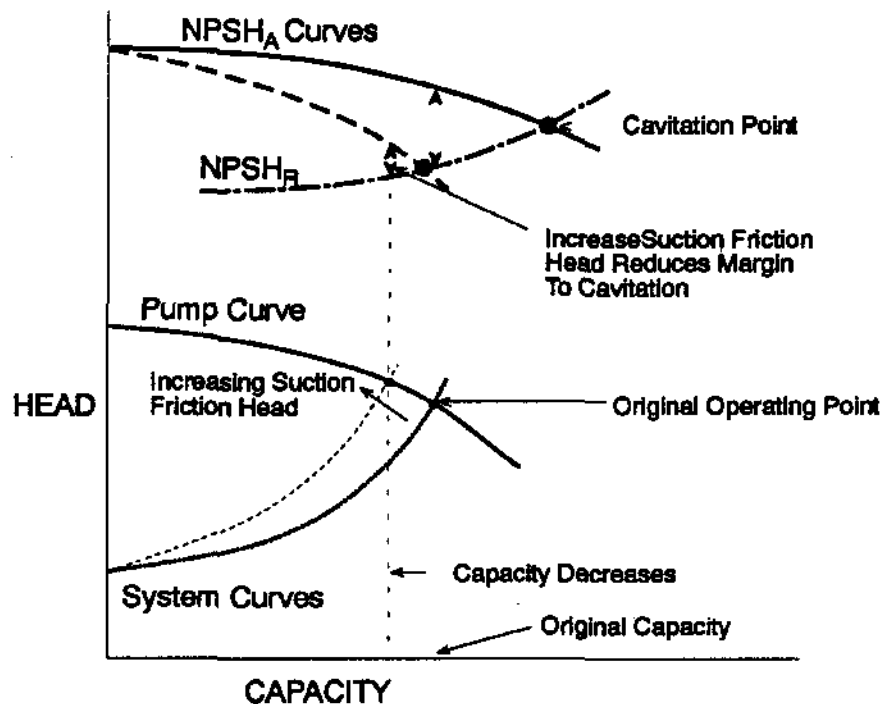
### Changing Suction System Friction Head

The suction system friction head can be changed adversely by accumulation of dirt in the strainer of the pump suction line or beneficially by cleaning the strainer. Adjustment to the isolating valve in the pump suction line would also change the suction system friction loss coefficient ( $C_L$ ). Since the pump suction isolating valve is normally fully open and strainers start out clean, we will assume this as the original operating state, and discuss only an increase in suction system friction.

*Obj. 1.4 d)*

## NOTES &amp; REFERENCES

By increasing the suction system friction coefficient, the  $NPSH_A$  will be adversely affected; but since the suction static head is unchanged, the net positive suction head available at zero capacity will not change. However, with a suction friction increase, less suction head will be available at the pump under flow conditions. The  $NPSH_A$  curve will droop downward as flow increases in the suction line. The margin to cavitation is reduced as suction friction head increases. This effect is shown in Figure 1.15. The small beneficial effect of reduced capacity with an increase in suction friction head is overwhelmed by the adverse effect of increased suction line friction.



**FIGURE 1.15**  
**EFFECT OF INCREASING SUCTION LINE FRICTION**  
**ON MARGIN TO CAVITATION**

### Changing Pump Inlet Temperature

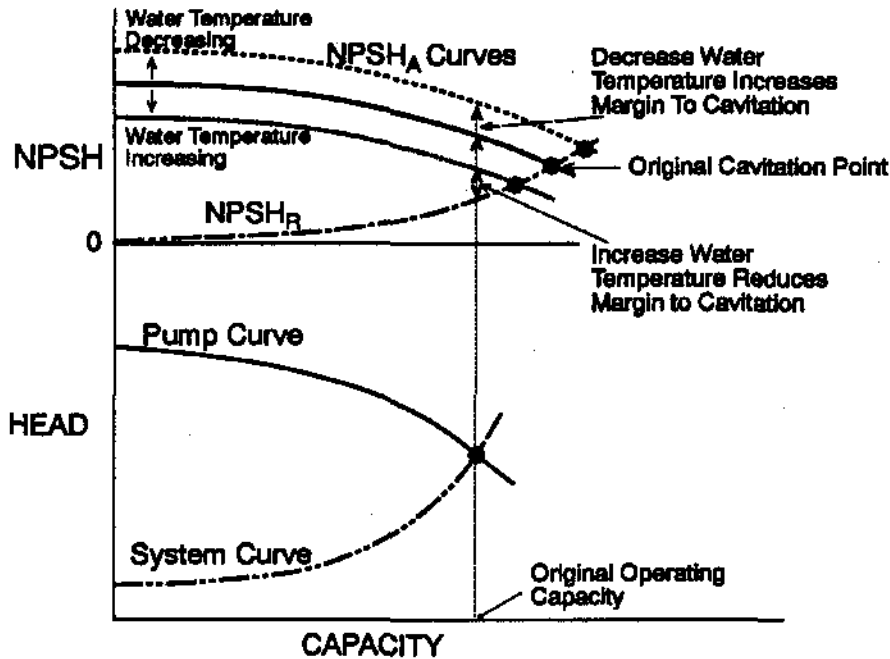
Two fluid properties, which change with temperature and influence  $NPSH$ , are viscosity and vapour pressure. Liquids become less viscous as temperature increases, hence decrease the frictional losses. Since water is the usual fluid involved in our pumping system, and since the viscosity of water is low, we will assume the friction head remains constant at all system operating temperatures.

On the other hand, vapour pressure of water increases as temperature rises. For the purpose of this module, we will assume that the vapour pressure change is significant, and hence has a measurable effect on  $NPSH_A$ .

NOTES & REFERENCES

Obj. 1.4 e)

Recall that NPSH represents the pump suction head less the vapour pressure head. As vapour pressure changes, the  $NPSH_A$  curve will retain its shape, but will shift up or down. Figure 1.16 shows that as the pump inlet temperature increases, a corresponding increase in vapour pressure will result in a reduced margin to cavitation. Similarly, a reduction in fluid temperature will reduce vapour pressure, and increase the margin to cavitation of the pump. Note that the  $NPSH_R$  curve is not affected by the temperature change.



**FIGURE 1.16**  
**EFFECT OF CHANGING PUMP INLET TEMPERATURE**  
**ON MARGIN TO CAVITATION**

## NOTES &amp; REFERENCES

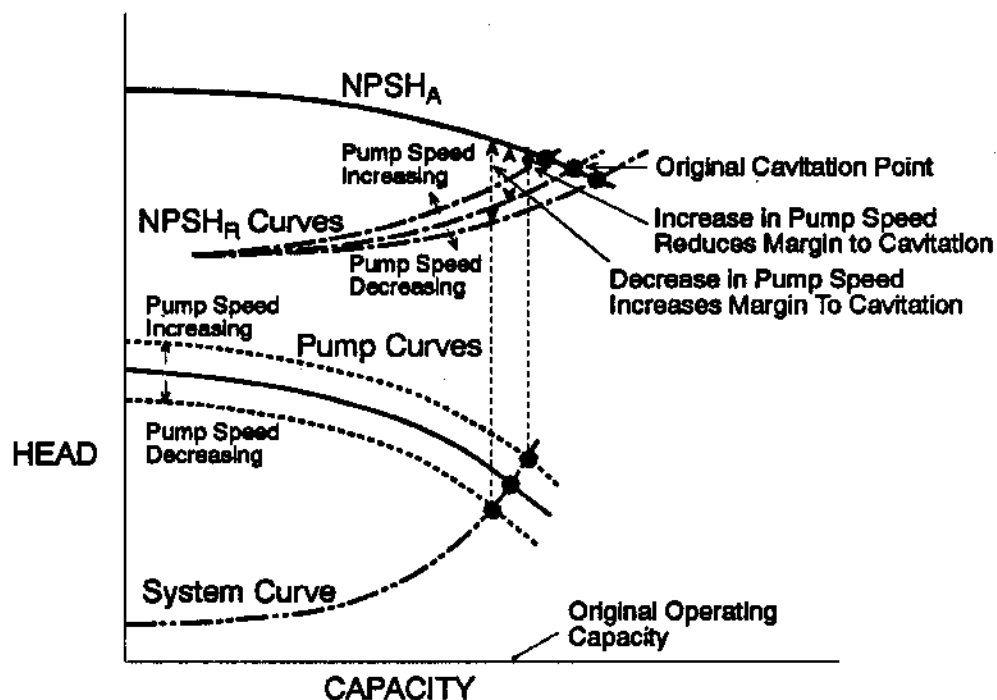
### Changing Pump Speed

For most of the centrifugal pumps in a generating station, changing the pump speed is not a normal operation possibility. However, where the speed can be changed (or the pump impeller changed to a different size), there will be an effect on pump capacity and margin to cavitation.

*Obj. 1.4 f)*

Changing pump speed will affect the pump operating characteristics. An increase in pump speed will allow the pump to supply more energy to the fluid, so the pump curve will shift up over the capacity range. With increased pump speed, the frictional resistance within the pump will increase, so the  $NPSH_R$  curve will increase in slope. The effects of increasing pump speed are shown on Figure 1.17. An increase in pump speed will reduce the margin to cavitation, while a reduction in pump speed will increase the margin to cavitation.

In pumps where the pump impeller size is changed, the impeller vane tip speed will also change. A larger impeller will have an increased vane tip speed which will produce an effect similar to increasing the pump speed. A smaller impeller will produce the same effect as a reduction in pump speed.



**FIGURE 1.17**  
**EFFECT OF CHANGING PUMP SPEED ON MARGIN TO**  
**CAVITATION**



## SUMMARY OF THE KEY CONCEPTS

- As system capacity is increased, the pump's margin to cavitation decreases.
- Decreasing the discharge tank level or pressure will reduce the margin to cavitation. Increasing the discharge tank level or pressure will increase the margin to cavitation.
- Increasing the discharge friction coefficient ( $C_L$ ) shifts the pump to a lower capacity. The pump will operate with an increased margin to cavitation. A decrease in discharge friction results in a decreased margin to cavitation.
- As suction tank level or pressure decreases, the margin to cavitation decreases, because the  $NPSH_A$  curve shifts lower. Raising the suction tank level or pressure will increase the margin to cavitation.
- Increasing the suction friction coefficient ( $C_L$ ) causes the droop of the  $NPSH_A$  curve to increase. This reduces the margin to cavitation of the pump.
- Increasing the process fluid temperature will increase the vapour pressure of the liquid, shifting the  $NPSH_A$  curve downward. The pump will operate with decreased margin to cavitation. A decrease in the process liquid temperature will increase the margin to cavitation.
- A change in liquid temperature does not affect the  $NPSH_R$  curve.
- An increase in pump speed or impeller size will shift the pump curve upward to a higher operating capacity as well as increase the slope of the  $NPSH_R$  curve. The margin to cavitation of the pump is reduced. Reduced pump speed will increase the margin to cavitation.

You can now do assignment questions 8–10.

*Page 41*

## NOTES &amp; REFERENCES

## CENTRIFUGAL PUMPS IN SERIES

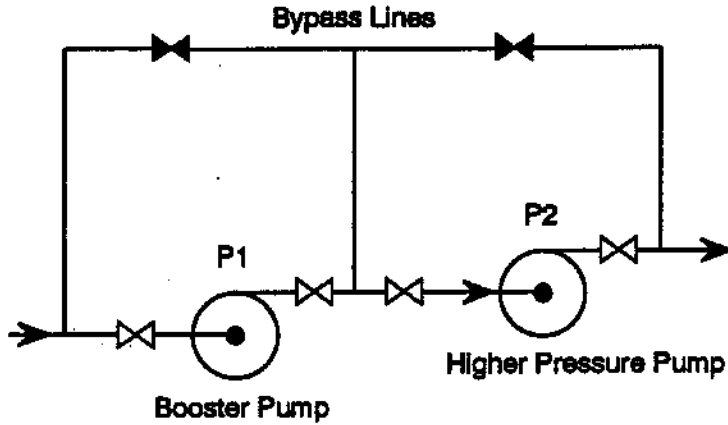
In cases where the pump  $NPSH_R$  is higher than the available  $NPSH$ , a booster pump can be used to increase the  $NPSH_A$ . An example of this type of circuit is the turbine lube oil system, where a booster pump provides the  $NPSH_A$  for main lube oil pump. The booster pump must have a larger rated capacity than the process pump and a  $NPSH_R$  smaller than the  $NPSH_A$  at its suction port. An illustration of a series type of pump arrangement is shown in Figure 1.18. The bypass line around each pump may allow operation of the system while one pump is isolated\*.

\* *Caution: Bypass lines that remain unused for an extended time span are likely to become contaminated with corrosion products. If so, in emergency operation, they would likely be ineffective, since the contamination could cause the other remaining pump or downstream equipment to quickly fail. Bypass lines must be regularly flushed in order to ensure their effectiveness when required.*

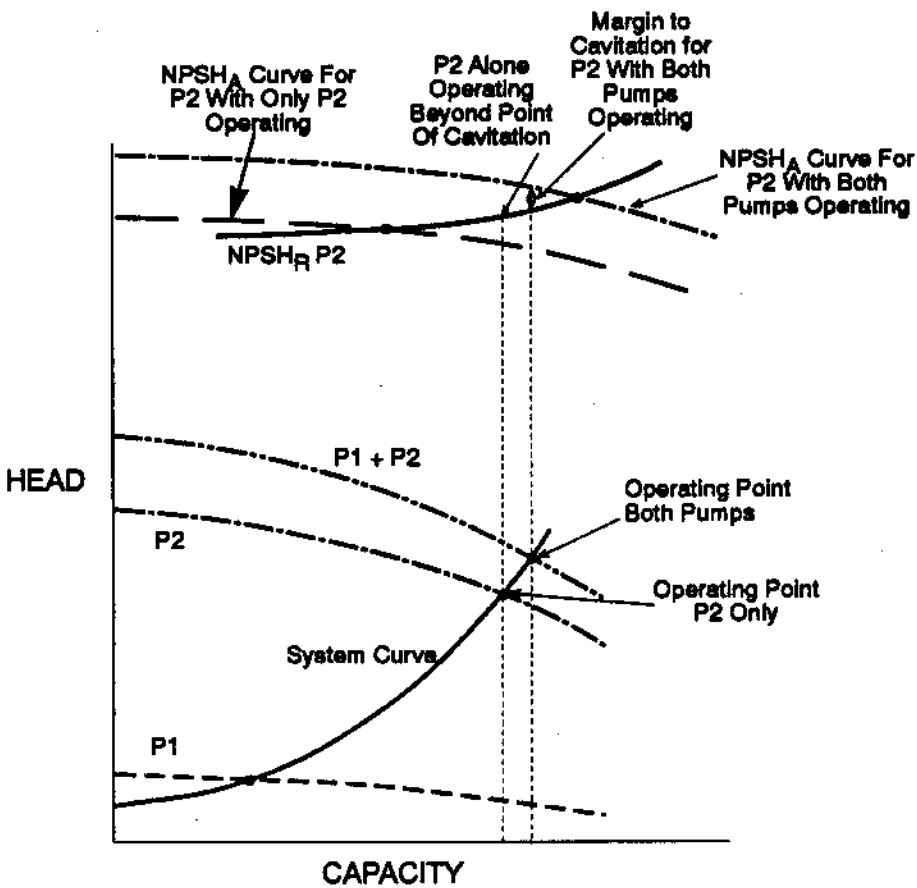
*Obj. 1.5 a & b)*

Figure 1.19 illustrates the pump curves for the booster pump, P1, the main pump, P2, and both pumps operating together, P1+P2. The operating points of both pumps 1 and 2 together are indicated as well as the operating point of pump 2 alone. Note that in our example, we have assumed that the booster pump P1 can supply a head greater than the system static head, but this condition is not an essential one. However, in our assumed case, P1 could supply some flow to the system if P2 was unavailable. Note also that the head developed by both pumps in series, the P1+ P2 curve, is the sum of the individual P1 and P2 heads at the same flow.

As indicated in Figure 1.19, when P1 is shutdown, the system head will decrease to that of P2 alone because the energy (head) contribution from P1 is lost. Also, system capacity is reduced as the operating point moves down the system head curve.



**FIGURE 1.18  
PUMPS IN SERIES**



**FIGURE 1.19  
PERFORMANCE CHARACTERISTICS  
OF SERIES PUMPS**

## NOTES &amp; REFERENCES

**Obj. 1.5 c)**

The  $NPSH_R$  curve for P2 and the  $NPSH_A$  curves for P2 with and without P1 running are also shown in Figure 1.19. With both pumps running, an acceptable margin to cavitation is shown. When P1 is shutdown, the net positive suction head available to P2 is reduced. The new net positive suction head available curve for P2 shifts downward keeping the same slope as the original conditions. Even though a lower capacity results in a lower  $NPSH_R$ , P2 will be operating beyond the point of cavitation, which we know is highly undesirable. To maintain continuing, safe operation, system flow has to be significantly throttled. The cavitation in P2 may be acceptable for a short period of time (ie. if immediate plant safety requires continued flow).

### Series Mounted Pumps In a Closed Loop Circuit

Heat transport systems have series pumps of equal capacity in a closed loop arrangement (Figure 1.20 shows the basic circuit used in some CANDU stations). The flowrate in the loop is constant, and each pump is only required to overcome half the loop resistance. The inlet of the pumps is boosted to an adequate NPSH by the pressurizing system. Note that the HTS pressure at the pump inlet establishes the system datum, with pump head and frictional resistance being related to it (Figure 1.21).

If the HTS pressure falls, the system curve falls. The system flowrate remains constant because the pump curve also falls an equal amount, and the pump head remains unchanged relative to the system pressure "datum". However,  $NPSH_A$  will be affected in such cases. Recall that:

$$NPSH_A = \frac{P_s}{\rho g} + \frac{v^2}{2g} - \frac{P_{vap}}{\rho g}$$

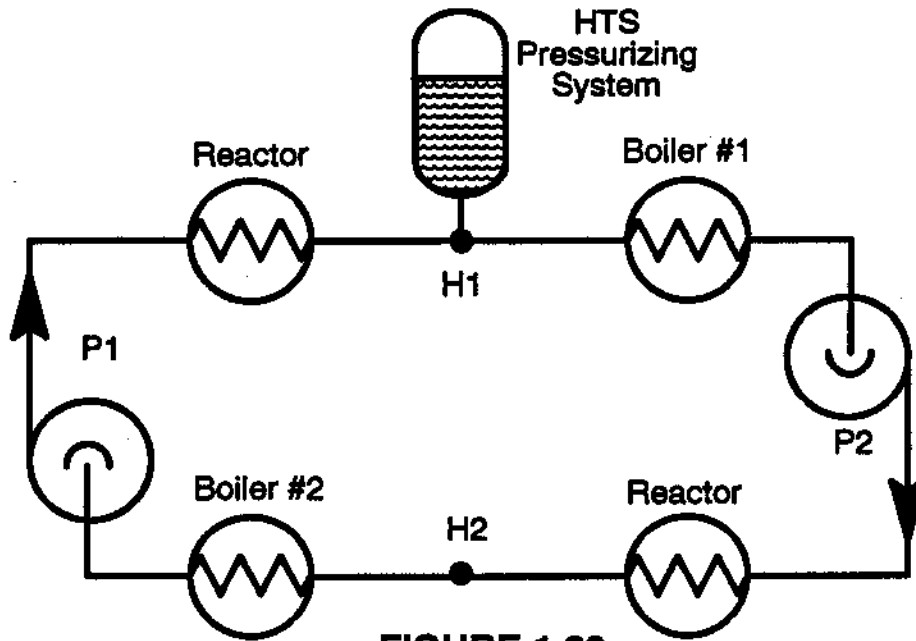
The  $NPSH_A$  falls if pressurizing pressure falls, or HTS temperature rises, hence under these conditions, pump cavitation becomes more probable (Figure 1.21 (b)).

The effect of increasing system resistance on  $NPSH_A$  is to swing the system curve upward and reduce flow, as shown in Figure 1.21 (a). Additional friction inserted anywhere in the circuit will introduce a pressure difference between the reactor outlet headers (points H1 and H2). The HTS pressure control scheme controls to the highest of these pressures\*, hence the other header's pressure will decrease\*, resulting in a lower pressure at the suction of its downstream pump, reducing its margin to cavitation. This effect is shown in Figure 1.21 (b).

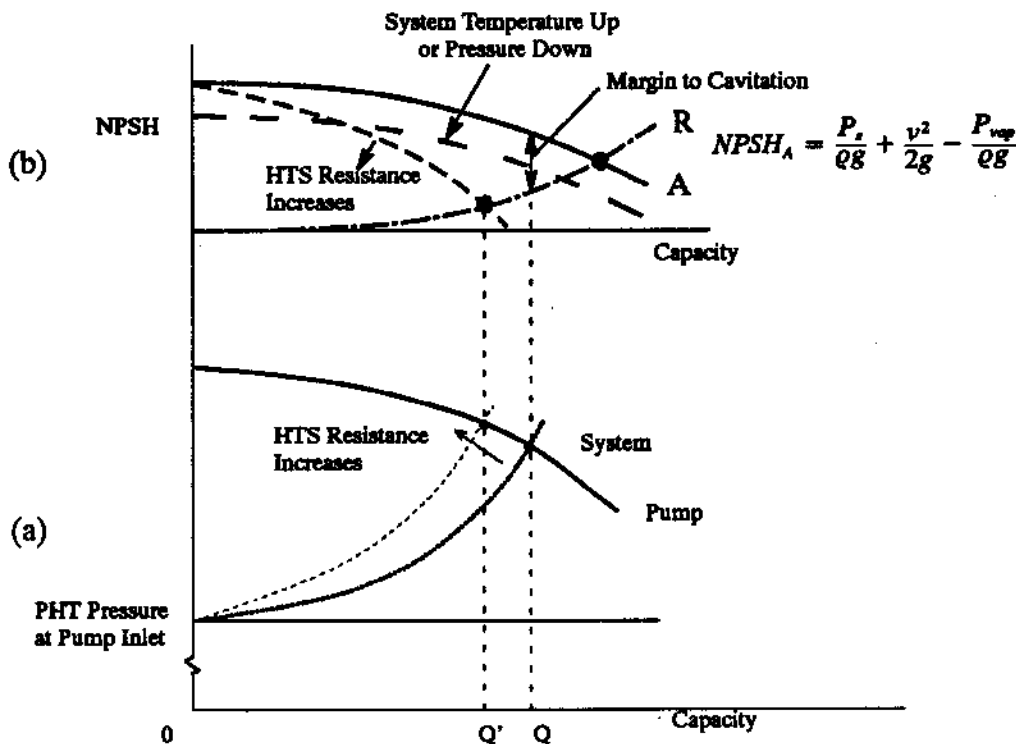
Additional resistance added downstream of the reactor outlet header (eg. plugging too many boiler tubes) will reduce  $NPSH_A$  at the downstream pump without a noticeable reduction in flowrate, and hence is probably of greater concern to operating staff (for example, the pump is more susceptible to severe cavitation or vapour lock during a low HT pressure transient, which could cause a reduction/loss of HT flow).

By controlling to the highest of the header pressures, excessive pressure anywhere in the system is prevented. But, because pressure control is based on the highest pressure, pressure elsewhere in the system will be allowed to drop.

The amount of decrease will be the amount caused by the resistance "imbalance" in the system.



**FIGURE 1.20**  
**SERIES MOUNTED PUMPS IN A CLOSED**  
**LOOP (HEAT TRANSPORT) SYSTEM**



**FIGURE 1.21**  
**CLOSED LOOP SYSTEM - PUMPS IN SERIES**  
**PERFORMANCE CHARACTERISTICS**

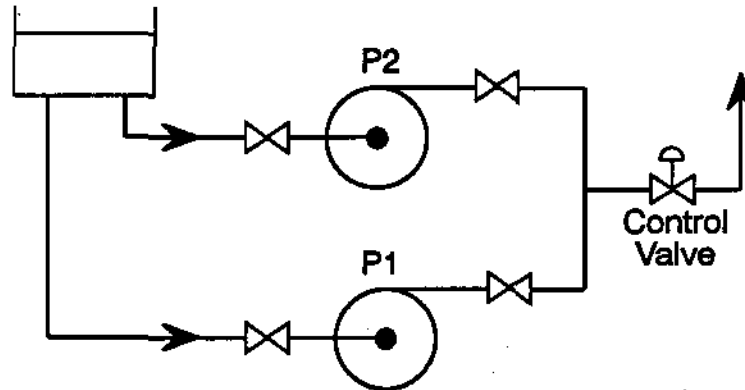
## NOTES &amp; REFERENCES

## CENTRIFUGAL PUMPS IN PARALLEL

### Parallel Pump Operation

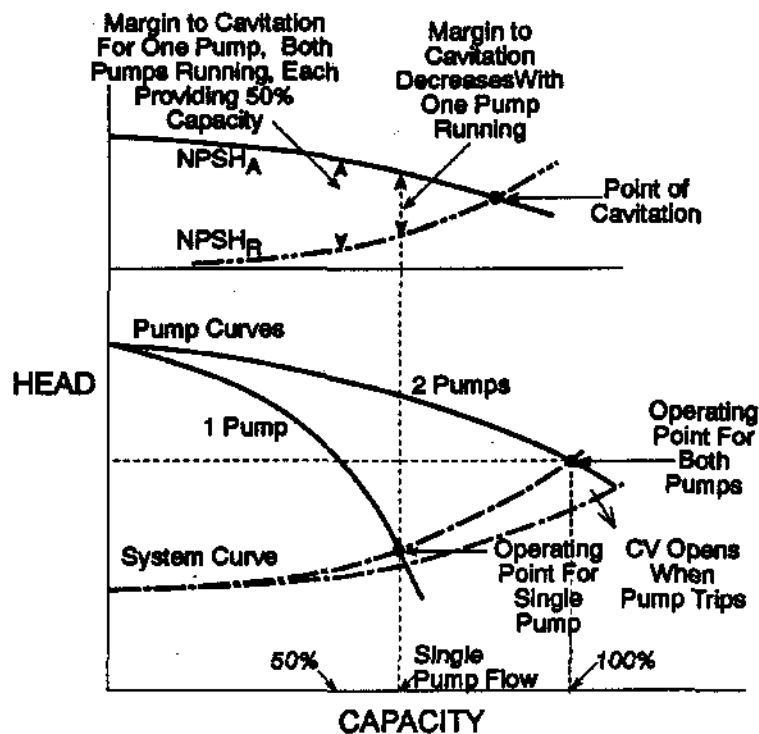
Almost all pumping systems in our generating stations have two or more pumps mounted in parallel. The reasons for this arrangement are to improve system reliability and to allow the pumping system to operate efficiently over a wide range of capacities. At reduced flow rates, one pump can be shutdown allowing the remaining pump(s) to continue to operate close to rated capacity and hence maintain optimal efficiency. It is also important that pumps operating in parallel are of the same size and have similar, "stable" pump characteristic curves.

We will discuss two types of parallel pump suction arrangements since the suction configuration has a significant influence on the pump performance. In the first case, there are separate supply lines to each pump. Figure 1.22 illustrates the first case where there is a common discharge header and control valve. We assume that flow conditions through both pumps are identical.



**FIGURE 1.22**  
**PARALLEL PUMP SYSTEM WITH**  
**SEPARATE SUCTION LINES**

Performance curves of two pumps with this parallel arrangement are illustrated in Figure 1.23. The diagram shows curves for either pump, both pumps, the system curve, the operating points for single and double pump operation, and the NPSH curves (related to the single pump Head/Capacity curve). The 100% flow at the operating point for both pumps is double that for a single pump operating at the same head. In fact, for any value of head along the two pumps curve, the flow will be double that of an individual pump at the same head.



**FIGURE 1.23**  
**PARALLEL PUMP PERFORMANCE IN A**  
**SEPARATE SUCTION LINE SYSTEM**

Figure 1.23 also illustrates the effect of shutting down a pump. When one pump is shut down, the operating point moves down the system curve. From the diagram, it is evident that the system head will reduce to the new operating point.

*Obj. 1.6 a)*

As shown in Figure 1.23, although system flow will be reduced by shutting down a pump, the resulting flow is greater than half the two pump flow. This is because the friction head of the system is lower. Thus, the remaining pump is able to operate at a higher capacity than with two pump operation.

*Obj. 1.6 b)*

Consider the  $NPSH_A$  for an operating pump. There is no change to the suction conditions, other than flow, when its operating twin is shut down. Because there are separate suction lines, the  $NPSH_A$  curve remains the same. The  $NPSH_R$  curve will not change from one mode of operation to the other. However, the operating pump will see an increased flow in going from parallel operation to single pump operation. As shown in Figure 1.23, the margin to cavitation decreases when one pump is shutdown, for a separate suction line system.

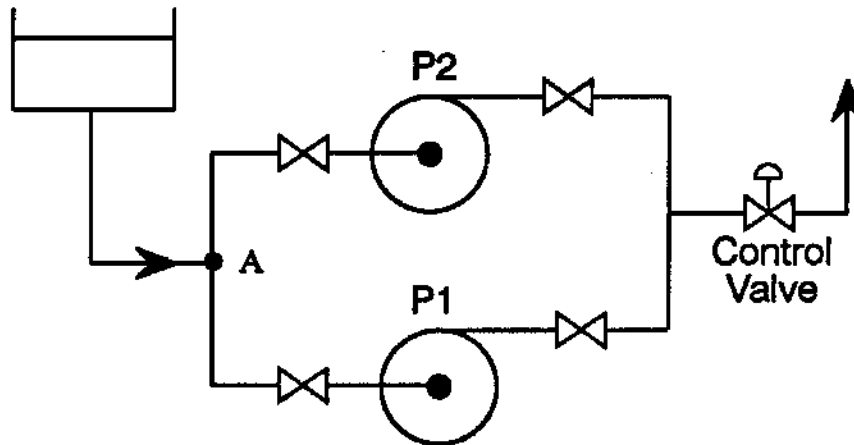
*Obj. 1.6 c)*

If the control valve (Figure 1.22) is not fully open when one pump is tripped, the valve will probably move to the fully open position in attempting to maintain the flowrate. Such action will cause the system head/capacity curve slope to decrease, increasing the capacity of the remaining pump further, hence further decreasing the  $NPSH$ , and increasing the possibility of cavitation.

## NOTES &amp; REFERENCES

**Parallel pumps, common suction line dominant.**

We will now discuss the second case, where a single suction line supplies the pumps mounted in parallel, as shown in Figure 1.24. We know that when one of the pumps is shutdown, flowrate to the remaining pump increases, reducing the  $NPSH_A$ . However, at the same time, flowrate in the common part of suction line decreases substantially, thus increasing the  $NPSH_A$ .

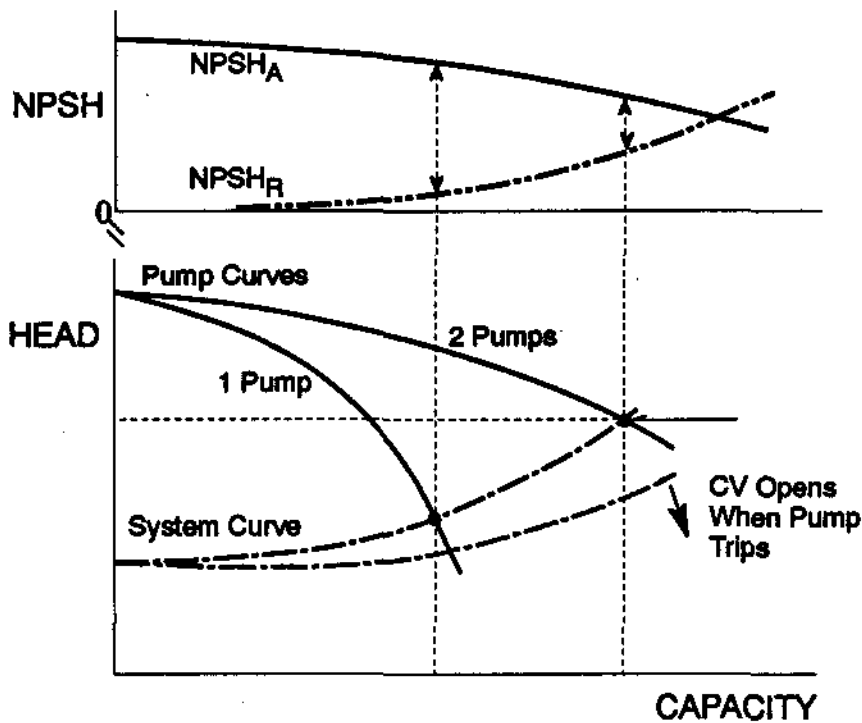


**FIGURE 1.24  
PARALLEL PUMP SYSTEM WITH  
COMMON SUCTION LINE**

*Obj. 1.6 c)  
continued*

When the common suction line is dominant, we can visualize the parallel pumps as a single unit, incorporating two independent impellers with point A taken as the interface between the suction piping and pump. The individual suction lines are then assumed to be part of the pump. Hence we are interested in the values of  $NPSH$  at point A. We can illustrate the difference between one and two pump suction line characteristics, by sketching head/capacity curves for one and two pumps (impellers) as before, but drawing the  $NPSH$  curves related to the two pump curve. The curves are shown in Figure 1.25 and correctly indicate a net increase in  $NPSH_A$  when one of the pumps is shutdown, even though the remaining pump's capacity has increased.





**FIGURE 1.25**  
**PARALLEL PUMP PERFORMANCE IN A COMMON**  
**SUCTION LINE LOSS DOMINANT SYSTEM**

As in case #1, a partially open discharge CV will probably open fully when system capacity decreases, in attempting to maintain flowrate. But when the *frictional characteristics of the common suction line dominate those of an individual pump suction line*, the adverse effect on the margin to cavitation is negligible.

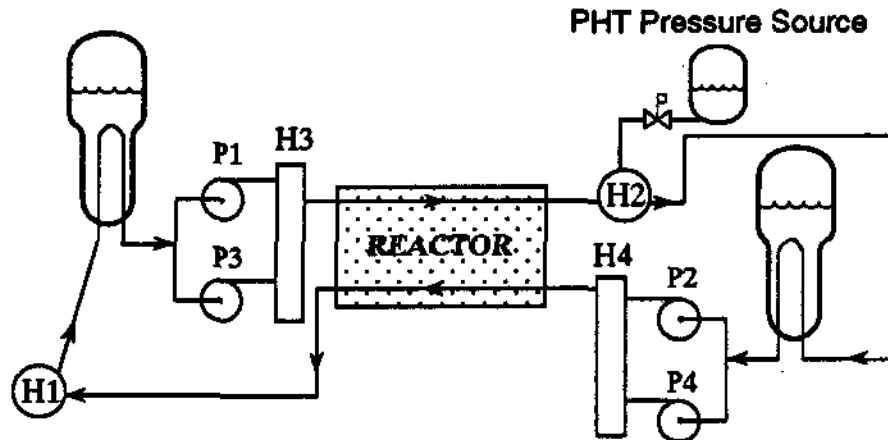
## NOTES &amp; REFERENCES

**PUMPS IN SERIES AND PARALLEL**

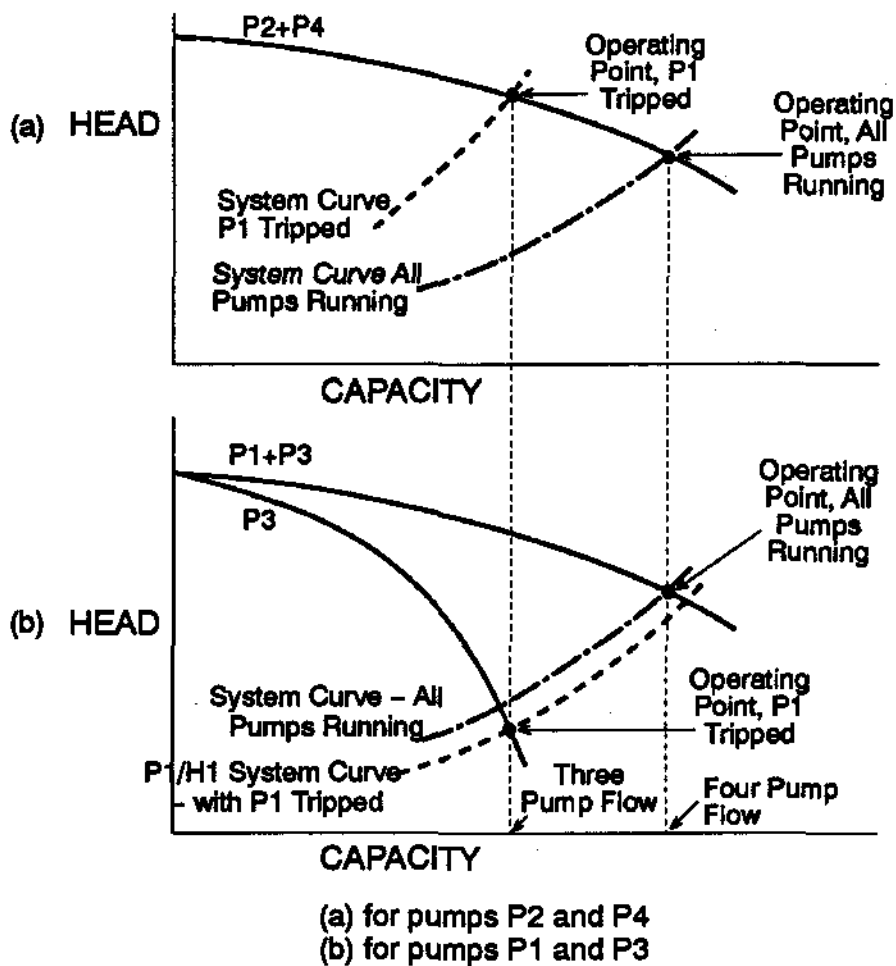
We will now expand the discussions by looking at parallel pumps mounted in series with other pumps. Some CANDU stations have this type of arrangement for their heat transport systems, and a simplified circuit is shown in Figure 1.26. Note that the pressurizing system is connected to header #2 (H2), but it controls system pressure to the higher of the pressure in reactor outlet headers H1 or H2\*. When the system is operating normally, each pair of pumps supplies equal flows and system performance will be the same as discussed earlier\*\*. We will now look at the performance changes when one pump (say P1) is shutdown, leaving the remaining three in operation. Figure 1.27 indicates graphically how capacity and head of the pumps differ from normal operation.

This was explained in the sidenote on page 23.

This was discussed on Page 1-26 in the parallel pumps section.



**FIGURE 1.26  
CANDU PHT PUMP ARRANGEMENT**



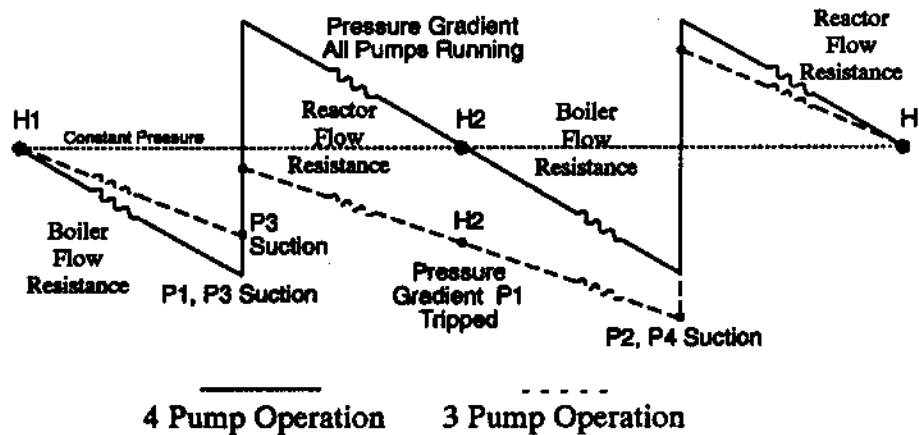
**FIGURE 1.27**  
**CANDU PHT PUMP CURVES SHOWING**  
**THE EFFECT OF TRIPPING ONE PUMP**

In normal operation, each pair of pumps supplies a head which is sufficient to overcome half of the loop resistance. When P1 is shutdown, system flowrate decreases, but P3 flowrate increases to a new operating point as shown in Figure 1.27 (b). The combined flow from P2 and P4 reduces to match it (Figure 1.27 (a)).

*Obj. 1.7 a & b)*

## NOTES &amp; REFERENCES

Figure 1.28 shows how the pressure gradient through the loop changes as P1 is shut down and capacity is reduced.



**FIGURE 1.28**  
**PRESSURE GRADIENTS AROUND THE PHT CIRCUIT**  
**FOR 4 AND 3 PUMP OPERATION**

In effect, P3 creates a barrier to flow and establishes a static head large enough to increase the head of P2 and P4, (despite less flowrate resulting in reduced circuit resistance). Pumps P2 and P4 now act as booster pumps to P3. More simply (and neglecting any pressure control for a second), piping downstream of P3 and upstream of P2/P4 have one pump feeding it, and two pumps taking suction from it, and logically, causing its pressure is to decrease\*. Similarly, piping downstream of P2/P4 and upstream of P3 have two pumps feeding it, and only one pump taking suction from it, causing its pressure to increase\*.

Recall that flow resistance varies with the flow velocity squared, hence when P1 shuts down, system flowrate and velocity drop to around 60% and the head loss due to friction falls to about 36% of its original value. You will see the effect of decreased flow/resistance on the slope of the pressure profile lines in Figure 1.28.

The pressurizing system responds to the higher pressure in H1 or H2. In this case (P1 is shut down), pressurizer pressure decreases to control pressure in H1 to setpoint, but allows a further pressure decrease in H2. If P2 or P4 were shutdown (instead of P1), H2 pressure will tend to increase, hence its pressure would be controlled, and H1 pressure will be allowed to decrease.

The response of the pressure control system may mask the size of system pressure changes, but the magnitude of the head changes can be detected.

When a pump is shutdown, a brake is applied to prevent it from spinning due to flow through it. The pump is not isolated from the flow, hence some of the flow from the running pump will recirculate through the stationary partner. For this example, if the brake is not applied, P1 will rotate backwards. Figure 1.28 shows that the P2/P4 suction head has decreased, thereby reducing the margin to cavitation (the actual change depends on H2 pressure reduction and the decrease in losses in the inlet piping of P2/P4, which act to increase the suction pressure).

To summarize, the effect of tripping P1 is to decrease the system flow, as indicated by the new operating points on Figure 1.27. The system pressure will decrease throughout the PHT circuit as indicated on Figure 1.28, except for the one header controlled to setpoint by the pressurizer (where it remains constant) and the suction line to the single pump (where it increases). In the case of P1 or P3 tripping, Header 1 is controlled to setpoint pressure.

*Obj. 1.7 c)*

## SUMMARY OF THE KEY CONCEPTS

- When two pumps are operating in series and one is shutdown:
  - the system head will decrease because the contribution of one pump is lost;
  - the system (pump) capacity will decrease.
- When the booster pump of series mounted pumps is shutdown or the pressurizing system of a closed loop circuit fails, cavitation will occur in the remaining pump.
- When two pumps are operating in parallel and one is shutdown:
  - the system head will decrease;
  - the system flow will decrease;
  - the remaining pump will operate at a higher capacity.
- When one parallel pump is shutdown, and where there are separate suction lines, the operating pump will have a reduced margin to cavitation.
- When one parallel pump is shut down, and where there is a common suction line and it is the dominant factor, the margin to cavitation will increase for the operating pump.
- When one pump is tripped in a series/parallel arrangement, system flow will be reduced. Head for the running "pump pair" will increase, and the head for the "single running pump" will decrease. HTS pressure setpoint will be maintained in the ROH whose pressure tends to increase.

You can now do assignment questions 11-14.

*Page 42*

## NOTES &amp; REFERENCES

## GENERAL OPERATING PRACTICES

Since the pump is the most likely component to cause system malfunction, general operating practices are directed toward ensuring the pump's integrity. We will discuss system operational practices to avoid pump damage from cavitation, air-locking, and water hammer. Precautions dealing with the start of a pump on a shutdown system will be discussed. Checks on auxiliary services such as bearing lubrication and gland sealing will be discussed. Precautions on isolating a shutdown parallel pump will be dealt with. All of these practices will result in safer operation with beneficial effects to pump performance.

### Obj. 1.8 a)

#### Preventing Cavitation

To avoid cavitation,  $NPSH_A$  must be greater than  $NPSH_R$ . The factors affecting  $NPSH_A$  are pump suction pressure and fluid temperature. In practice, pump suction pressure can be maximized by:

- i) Operating with the suction tank level or pressure at or near their maximum permitted values. Either or both of these actions may be practical. For closed loop systems, pump suction pressure is maintained by correct operation of the pressurizing system.
- ii) Minimizing suction line energy losses by:
  - cleaning strainers;
  - fully opening isolating valves;
  - ensuring common line suction flow rate is not excessive. This could result, for example, from three 50% pumps operating in parallel, simultaneously.
- iii) Maintaining fluid temperature in the correct temperature range specified for the system, to minimize the vapour pressure component of  $NPSH_A$ .

### Obj. 1.8 b)

#### Preventing Air Locking

Air locking occurs when an excessive amount of air enters the pump suction port. A common cause of air ingress is a low liquid level in the suction tank. In suction systems that operate under a vacuum, such as the Condensate Extraction Pumps, an open air vent (or drain valve), leaking valve packing, or a leaking joint in the suction line could let air in. Where a pump operates under suction lift conditions, air can be drawn in through the pump shaft packing. To prevent air locking conditions, operations staff should:

- i) Ensure the suction tank level is maintained at the correct value so that air in-leakage is eliminated;
- ii) Ensure suction line drain and vent valves are tightly closed;
- iii) Watch for and report signs of deteriorating joints and packing;
- iv) Confirm seal water is in service (where installed).

## Indications of Cavitation and Air-Locking

In any operating pumping system, indications of cavitation and air-locking are the same, and increase with the condition's severity. Severe cavitation and air-locking conditions are both indicated by noise, heavy vibrations, pulsating flow and pump-motor current fluctuations. For large systems, the three latter symptoms may be indicated on control room instrumentation, otherwise the field operator is responsible for their detection. Less severe conditions of cavitation or air-locking could be indicated by a noticeable reduction in system capacity and/or an increase in pump operating temperature.

*Obj. 1.9*

## Preventing Water Hammer

The possibility of water hammer is always a concern in liquid process operations, since piping and storage vessels have been moved from their supports by the phenomena. Water hammer occurs when a liquid column is subjected to a sudden change in velocity. The following examples briefly describe conditions which can result in water hammer, and the actions recommended to minimize its adverse effects. In all cases, the preferred actions are directed towards minimizing acceleration or deceleration of the liquid column, and hence the forces applied to system components.

*Obj. 1.10*

- i) A sudden change in liquid velocity is caused by rapidly opening or closing a valve. Consequently slow operation of valves is the recommended practice whenever possible.
- ii) Water hammer can be induced by suddenly starting a pump against a stationary column of liquid. Therefore in such cases, the preferred procedure is to start the pump against a closed discharge valve, then slowly open it, causing the liquid column to accelerate slowly.
- iii) Water hammer can be induced when a pump, energizing a long liquid column, is shut down suddenly. Momentum of the column causes it to separate, and hydraulic shocks occur when the separated parts come together again. Thus the preferred procedure is to slowly close the discharge valve fully, before tripping the pump.
- iv) Water hammer can be initiated by refilling a drained system too quickly. Severe hydraulic shocks can occur when the system becomes full and flow stops suddenly. Thus the correct action is to either throttle the flowrate from the process pump, slowly into the system, or use a small filling pump.

## NOTES &amp; REFERENCES

- v) Water hammer effects can be created when vapour-locking or air-locking conditions exist in a pump suction line. Either the sudden formation and collapse of vapour pockets in the liquid (vapour-locking) or the sudden introduction of air bubbles (air-locking), will cause rapid acceleration and deceleration of the liquid column, and hence large forces to be applied to the piping system.

*NOTE: Piping systems may also contain devices to minimize water hammer effects. Water hammer arrestors reduce pressure surges. An arrestor is an assembly of a steel cylinder with an internal pressurized bladder. Also, orifice plates may also be placed throughout the system to throttle excessive pressure fluctuations.*

### **Centrifugal Pump Start and Operation Checks**

In starting centrifugal pumps from shutdown conditions, it is important that various pump checks are performed. Often, for important systems, most of these checks are incorporated in the pump start permissive logic. Where not included, they will be part of the system operating procedure. Some of the more important checks are described, with reasons for performing them.

*Obj. 1.11 a)*

A process system is often drained, when shutdown for maintenance. Hence before re-starting the system, the suction piping and pump bowl must be primed. The priming procedure requires the suction piping and pump bowl to be completely filled with process liquid, and this condition implies the need for all trapped air to be vented (via "vent cocks" installed at high points in the suction system). Even when the pump is primed by gravity (which is usually the case in our stations) air venting is still essential, because it eliminates the air-locking conditions and its water hammer effects.

*Obj. 1.11 b)*

Before a process system is started, the suction isolating valves must be fully open. In addition, verify that suction strainers are clean. The reasoning here is for cavitation prevention.

### **Starting the Pump against a Closed Discharge Valve.**

*Obj. 1.11 c)*

Prevention of water hammer is usually the main reason why pumps are started with their discharge valves closed. Once the pump has reached running speed, or sometimes before, the discharge valve is slowly opened, allowing the liquid to accelerate through the system in a controlled manner.

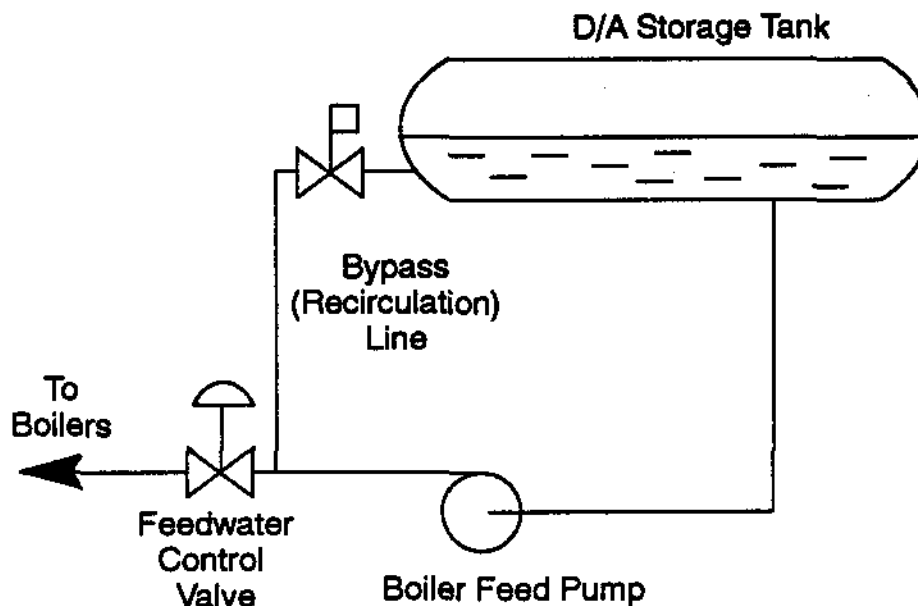


Another reason for starting radial flow pumps with their discharge valve closed is to minimize the starting power required. Recall that radial type impellers require least power at zero capacity, (refer to Figure 1.1). Unfortunately, we cannot claim the same benefit for the axial flow pump, since it draws maximum power at zero capacity, as can be seen in Figure 1.5. Nevertheless, in the interest of avoiding water hammer, the axial flow CCW pumps are started with their discharge valves closed. This practice is reinforced by the fact that these pumps operate in parallel without discharge check valves installed, hence when shutdown, depend on their closed discharge valves to avoid being driven backwards by flow from the running pump.

### Preventing Over-Heating.

Immediately after starting a pump against a closed discharge valve, the valve should be opened. This is because the energy losses, due to friction and turbulence inside the pump, will heat the pump and its contents very quickly. At rated capacity, the pump efficiency is maximum and flowrate is large, therefore the heating effect is small. At reduced capacities, efficiency is also reduced (as shown in Figure 1.1) so pump heating is intensified and the probability of cavitation increased. Thus, when operating in the shut-off condition, even though the power input is relatively small, the pump can quickly overheat. Large pumps (such as the Condensate Extraction or the Boiler Feed pumps), which sometimes operate at very low capacities for long time periods, are provided with a bypass line, which permits them to always operate at a capacity sufficient to avoid over-heating. Figure 1.29 shows a basic arrangement for the boiler feed pumps.

*Obj. 1.12*



**FIGURE 1.29**  
**BASIC CIRCUIT USED TO PREVENT OVER-HEATING**  
**OF THE BOILER FEED PUMPS**

## NOTES &amp; REFERENCES

While we are discussing pump overheating, recall that cavitation and air entrainment reduce pump capacity, and will promote pump overheating. Therefore, practices which prevent cavitation and air entrainment will also reduce the chances of pump overheating.

### **Pump Bearing Lubrication Verification**

*Obj. 1.11 d)*

Where a pump bearing lubrication system is installed, verification that it is in service, is required. Inadequate lubricant flow will cause over heating and *damage to the bearings, reducing pump life*. Most large radial flow pumps have an auxiliary bearing oil lubrication system and trip logic activated after about 20 seconds of low bearing flow. Most other pump's bearings employ a self contained oil lubrication system. In a few large pumps (eg. the CCW pumps), process water is used for bearing lubrication. In all cases, regular monitoring of the lubricant flow, as well as the flow of cooling water used to control lubricant temperature, is essential.

### **Gland Seal Supply Verification**

*Obj. 1.11 e)*

Gland seal liquid is supplied where it is necessary to provide cooling and lubrication for the pump glands (shaft seals). Most pumps employ process water to provide seal flow. The system maintains the gland cavities at a higher pressure than the surroundings to prevent outflow from the pump or inflow of air from ambient. Reliability of the gland seal liquid supply is particularly important when the escape of process liquid is a threat to the safety of personnel or plant operation. For example, the primary heat transport pump gland seal prevents the escape of hot HTS D<sub>2</sub>O from the system, and hence the escape of radioactive steam. In the case of the boiler feed pumps, gland seal liquid flow prevents the hazard of escaping steam and hot condensate.

Also, for suction systems operating under a vacuum (such as the condensate extraction systems), gland seal liquid prevents the ingress of air into the system. Air-locking of the condensate pumps and corrosion in the condensate, feedwater and boiler systems are the major concerns.

In addition, the packings of isolating valves in vacuum lines receive a seal water supply to prevent air in-leakage at the valve stems.

Regular monitoring of all components using a gland seal liquid flow is required.

### **Preventing Thermal Shock**

*Obj. 1.13*

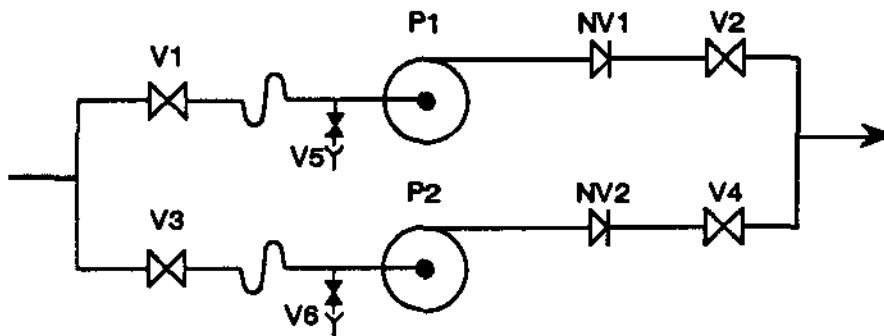
Thermal shock implies the creation of excessive thermally induced stresses in a component, and results when it is subjected to a sudden change in temperature. Therefore, pumps that are valved into high temperature systems, must be protected from rapid temperature changes. The two methods used to provide this protection are:

1. Slowly pre-warm the pump prior to start up, using the process liquid, or:
2. Keep the pump warm at all times, by means of a supplementary heating system.

Also, in the operation of piped liquid systems, a suddenly applied temperature difference (particularly in heat exchangers) will increase the probability of localized boiling, which can cause severe mechanical shocks (ie. water hammer) to the system. Consequently during system start up and operation, staff must be constantly on the alert for operating conditions which could promote thermally induced shock. Actions which gradually adjust a liquid's temperature and/or pressure, are generally the safest and therefore the most desirable.

### Safe Isolation of One Pump in a Parallel Configuration

Consider the configuration in Figure 1.30 as an example. Suppose that pump P2 has been shutdown and is to be isolated for repairs. Pump P1 continues to supply the system with flow. NV2 closes automatically to prevent a significant re-circulation of flow through P2 (however, NV2 is not leak-proof).



**FIGURE 1.30  
ISOLATION OF ONE PUMP IN A  
PARALLEL ARRANGEMENT**

The correct method of isolating P2 is as follows;

- Fully close discharge valve V4 (V4 can be guaranteed closed, but not guaranteed leak-proof) then;
- Fully close suction valve V3, then;
- Immediately, fully open valve V6, to allow any leakage through V3, NV2 and V4 to drain away.

If the stationary pump is isolated incorrectly, (ie, V3 is closed before V4 is closed) the suction piping of P2 can be over-pressurized by leakage past NV2 (ie, the pressure will increase to P1 discharge pressure) and the suction piping could rupture. In at least one case, improper isolation of a pump has resulted in a worker's death.

*Obj. 1.14 a)*

*Obj. 1.14 b)*

## NOTES &amp; REFERENCES

**SUMMARY OF THE KEY CONCEPTS**

- Operating practices to avoid cavitation are:
  - maximizing pump suction pressure by maximizing suction tank level or pressure, and minimizing suction line energy losses;
  - maintaining fluid temperature in the correct operating range.
- For a pump with severe cavitation or air locking, the operator may detect any or all of the following:
  - a reduction in process flow;
  - rapid oscillations of the pump motor operating current;
  - severe noise or vibration;
  - an increase in pump temperature.
- Water hammer is prevented by:
  - slowly opening and closing flow control valves;
  - ensuring that the pump discharge valve is closed during pump starts;
  - ensuring a shutdown system is primed and vented before pump startup; and
  - taking actions to prevent air or vapour locking.
- Before a process system is started up, the pump suction piping must be primed and vented to ensure that air or gas in the system will not air lock the pump.
- Before a centrifugal pump is started, the suction isolating valves must be fully open to prevent cavitation, and;
- The pump bearing lubrication system and its cooling water supply must be in service to prevent overheating and subsequent bearing damage, and;
- The gland seal liquid supply must be in service, to provide cooling and lubrication to the pump glands and bearings. Gland seal systems are used to prevent the escape of process liquid from pumps operating above atmospheric pressure (usually for operational safety reasons) or to prevent the ingress of air into pumps and valves subjected to vacuum conditions. Therefore, verification of their correct operation is essential.
- When a centrifugal pump operating in parallel with others, is shutdown, the pump discharge valve must be closed before the suction line valve, to prevent discharge pressure being applied to, and hence over-pressurizing, suction system components.

You can now do assignment questions 15-19.

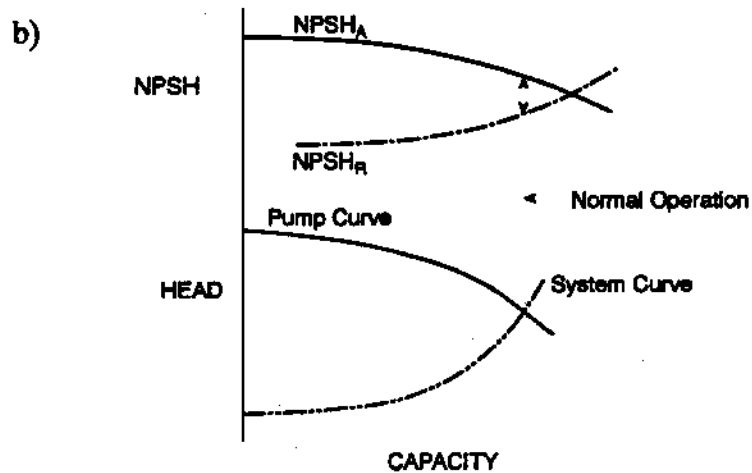
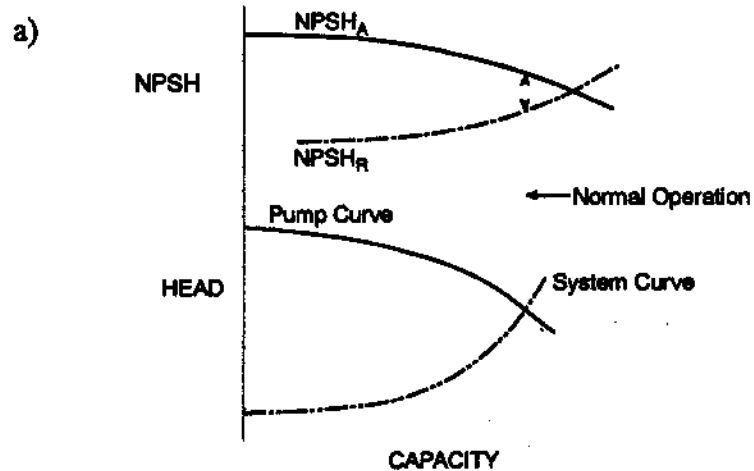
## ASSIGNMENT

1. Describe how pump efficiency varies with pump capacity.
2. Describe the two general ways pump horsepower varies with pump capacity.
3. Define  $NPSH_R$  for a pump.
4. Explain the term  $NPSH_A$ .
5.
  - a) State how  $NPSH_A$  and  $NPSH_R$  change in relation to each other as pump capacity changes.
  - b) State the normal operating relationship between  $NPSH_A$  and  $NPSH_R$ .
6.
  - a) Explain how an decrease in discharge tank level will affect pump capacity.
  - b) Explain how a increase in suction tank level will affect pump capacity.
  - c) State two ways fluid friction losses can increase in a system containing process pumps.
  - d) Explain how increased fluid friction losses will affect pump capacity.
7. As suction tank pressure decreases, the pump capacity will \_\_\_\_\_ . As discharge tank pressure increases the pump capacity will \_\_\_\_\_.
8.
  - a) Explain how suction tank level can affect pump cavitation.
  - b) Explain how discharge tank level can affect pump cavitation.
  - c) Explain how discharge valve throttling can affect pump cavitation.
  - d) Explain how suction valve throttling can affect pump cavitation.
  - e) Explain how fluid temperature at the pump inlet can affect pump cavitation.
  - f) Explain how pump speed can affect cavitation.
9. Explain the effect of throttling the pump discharge valve as the suction strainer becomes partially plugged.

## NOTES &amp; REFERENCES

10. Sketch new pump curves on the diagrams provided, for the following:

- a) Throttling in the suction piping;
- b) Increasing the fluid temperature at the pump inlet.



11. Two different centrifugal pumps are operating in series, then one pump is shut down.

- a) Explain how the capacity of the system changes.
- b) Explain how the tendency of the operating pump to cavitate will change.

## NOTES &amp; REFERENCES

12. With one pump operating, a second similar centrifugal pump, operating in parallel on a common suction line, is started up. The system head will \_\_\_\_\_ and the system flow will \_\_\_\_\_ . The capacity of the first operating pump will \_\_\_\_\_ . The tendency for cavitation to occur will \_\_\_\_\_ .
13. With one pump operating, a second similar centrifugal pump, operating in parallel with a separate suction line, is started up. The system head will \_\_\_\_\_ and the system flow will \_\_\_\_\_ . The capacity of the first operating pump will \_\_\_\_\_ . The tendency for cavitation to occur will \_\_\_\_\_ .
14. a) For the four pump arrangement of a CANDU heat transport system, when one pump is tripped, the system flow will \_\_\_\_\_ .
- b) Explain, how the remaining pumps' head will change.
15. a) Describe four indications of severe cavitation.
- b) Describe four indications of air or vapour locking.
16. Give two examples of good operating practices used to prevent cavitation.
17. State three examples of general operating practices used to prevent water hammer.

## NOTES &amp; REFERENCES

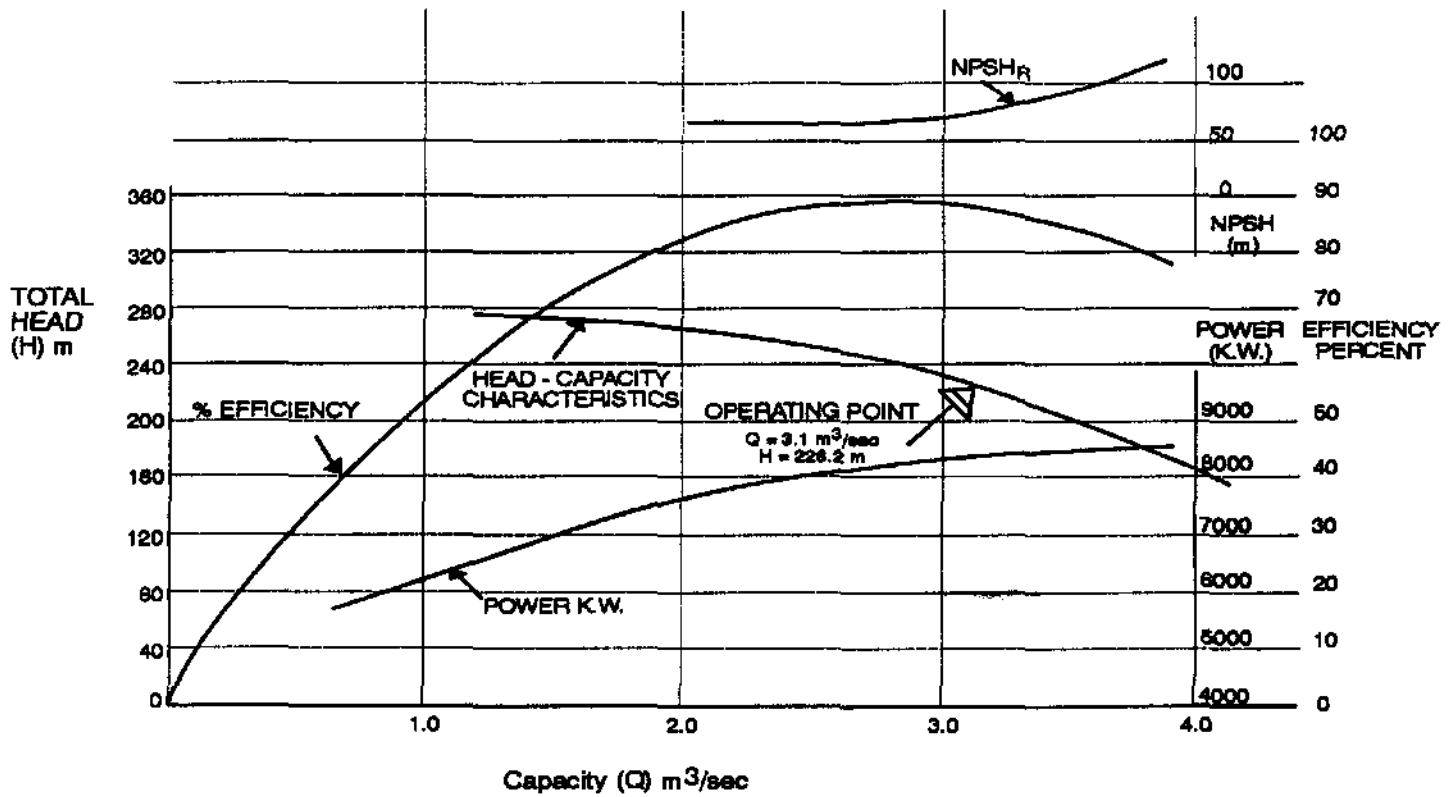
18. a) State one reason why the suction system is primed before starting a pump.
  - b) State two reasons why a pump discharge valve is usually fully or partially closed during pump start-up.
  - c) State one reason why suction isolating valves are normally fully open when a pump is started.
  - d) State one reason why the pump lubrication system must be in service prior to starting a pump.
  - e) State two reasons why the pump gland seal liquid system should be in service prior to pump start-up.
19. a) State the hazard and possible consequences of shutting down and isolating a centrifugal pump when a second pump continues to operate in parallel.
  - b) Describe the procedure necessary to safely isolate the shutdown pump.

Before you move on to the next module, review the objectives and make sure that you can meet their requirements.

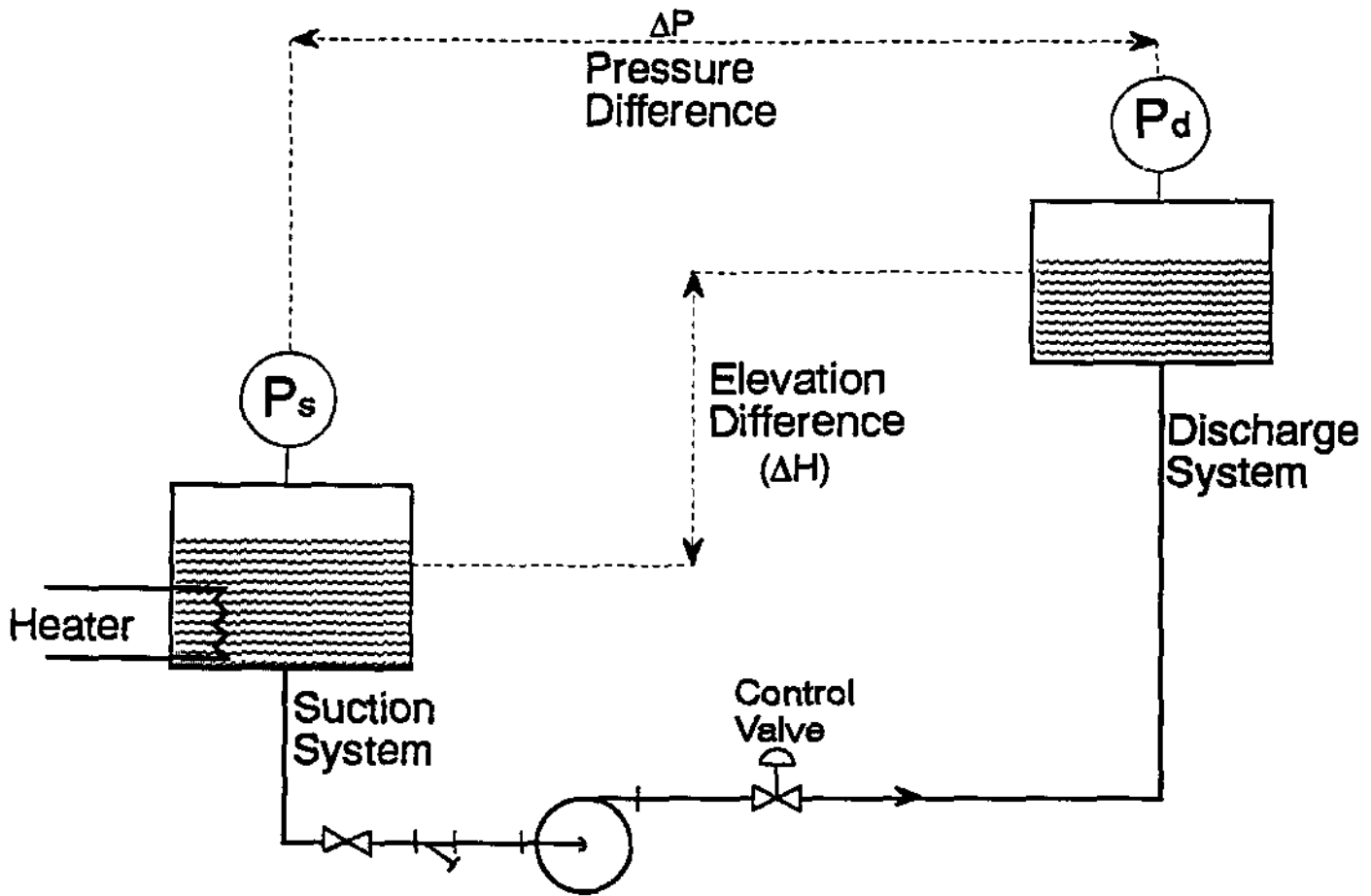
Prepared by: D. Bieman, WNTD  
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Revision: R-2, Feb 1994





**Figure 1.2**  
**ORIGINAL DARLINGTON NGS PHT PUMP CHARACTERISTICS**  
**(BORG-WARNER CANADA LTD. - 5 VANE IMPELLER)**



**FIGURE 1.6**  
**TYPICAL LIQUID TRANSFER SYSTEM**

## Module 2

# POSITIVE DISPLACEMENT COMPRESSOR OPERATION

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## OBJECTIVES:

After completing this module you will be able to:

- 2.1 Explain how Isothermal and Adiabatic compression processes affect the work (energy) required to compress and deliver a given mass of air.
- 2.2 Explain the effect of clearances and valve leakage on compressor performance.
- 2.3 Explain, using a pressure-volume diagram, the effect of each of the following on compressor performance:
  - a) Jacket cooling,
  - b) Two-stage compression with intercooling.
- 2.4 Explain three benefits of using an aftercooler on compressed air system operation.

*Page 2*

*Page 3*

*Page 4*

*Pages 4-5*

*Page 6*

\* \* \*

## INTRODUCTION

Various types of compressors are described in levels four and three mechanical equipment courses. This module will deal with the positive displacement compressors used in our stations.

As an introduction to this module, let's start by looking at the cycle used for the compression of the air that we use in our instruments, air tools, valve actuators, etc. Figure 2.1 shows a simplified air compression cycle as a pressure/volume diagram.

The shaded area in Figure 2.1 is representative of the work required to compress and deliver a particular mass of air, and for steady operation is representative of the power required. In this module, emphasis will be placed on how we compress and deliver air, and its effect on the work required.

## NOTES &amp; REFERENCES

Shaded area is representative of the work (energy) required to compress and deliver a particular mass of air. In this case, the cycle uses an adiabatic compression process.

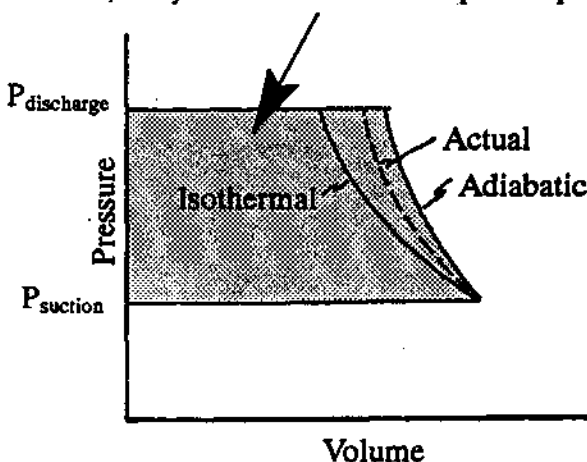


Figure 2.1: Simplified Air Compression Cycle

## THE COMPRESSION CYCLE

In theory, air may be compressed **adiabatically** (without the addition or removal of heat) or **isothermally** (at constant temperature).

It is difficult to compress gases at a constant temperature. The expense and size of heat removal equipment required for this makes the process impractical. Likewise, we cannot compress gases adiabatically, since some heat always escapes and friction heat generated during the compression process is added to the gas.

### Obj. 2.1

These two compression processes are illustrated on pressure/volume axes in Figure 2.1. We can see that the area of the overall cycle, bounded by the adiabatic compression process, will be larger than an area bounded by the isothermal compression process. This means that the work required to compress and deliver the air using an adiabatic process is more than the work required using an isothermal process.

From the definitions of adiabatic and isothermal, we can deduce that an actual compression process lies somewhere between them (ie. some heat is lost during compression, but not all of it). The process will lie closer to adiabatic than to isothermal\*, and a lot of heat is generated during the process. For example, air at 4°C if compressed from atmospheric pressure to 690 kPa (100psi), it will reach over 200°C\*.

To keep the work needed per cycle (or power) as small as possible, we try to make our compression process as close to isothermal as possible (ideally, it is better to shift the process over even further, but again, this is not practical). Therefore, improvement to the cycle will involve removal of heat during the compression process.

*These numbers are provided for illustrative purposes only. You are not expected to remember them or perform any calculations to derive these numbers.*



## NOTES &amp; REFERENCES

The increased clearances caused by wear will allow more high pressure air to remain in the cylinder at the end of the compression process. Due to an increased amount of air remaining in the cylinder (which must expand during the suction stroke), the point at which the suction valve opens will now be delayed. This reduces the amount of air drawn in (shortening section DA on Figure 2.2), reducing capacity. Similarly, section BC on Figure 2.2 will be shortened due to the increased clearance volume.

## METHODS USED TO IMPROVE COMPRESSOR EFFICIENCY

To approximate an isothermal process, and therefore reduce the compressor power requirements, we remove the heat of compression in two ways:

- i) By removing the heat during the actual compression process and,
- ii) By compressing the air in two stages, and removing heat between stages.

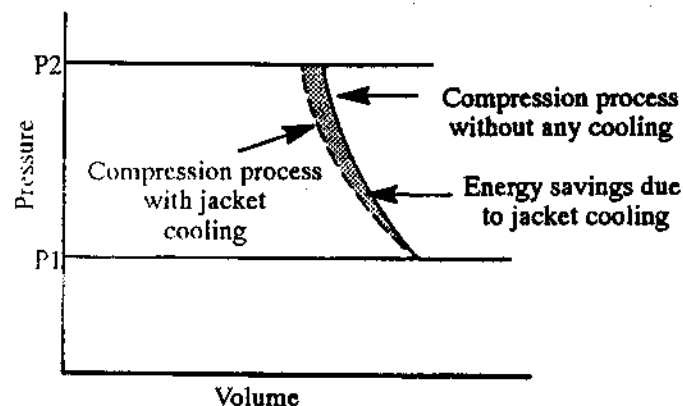
Each of these will be discussed below.

### Obj. 2.3 a)

#### Jacket Cooling

The removal of heat during the compression process involves cooling of the cylinder or compressor element (rotary screw type) with a water jacket. This cooling jacket surrounds the compressing component, and removes some of the heat of compression and friction generated during the process.

Thus, the compression process is shifted toward the isothermal process (ie. shifted left on the pressure-volume diagram). The shaded area labelled in Figure 2.3 represents the energy (power) saved during the compression process with jacket cooling.



**Figure 2.3: Jacket Cooling Effects**

Jacket cooling, as with all other forms of compressor cooling, helps keep compressor metal temperatures at practical levels, prolongs lubricant life and consequently reduces machine maintenance.

## NOTES &amp; REFERENCES

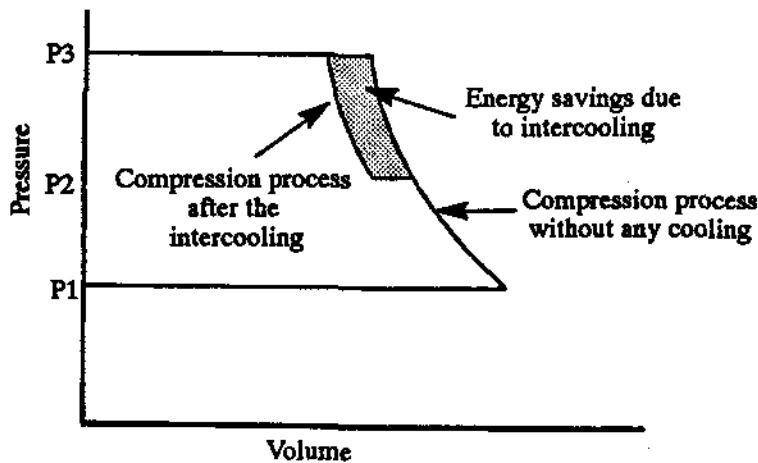
Obj. 2.3 b)

## Two Stage Compression With Intercooling

The second method of heat removal is to compress the air part way to its final pressure, extract some of the heat then continue compression. This is done by **compression in two stages with cooling of the air between stages using an intercooler**. Intercoolers were discussed in your previous mechanical equipment courses.

Figure 2.4 shows a two stage compression process. The overall compression between pressure  $P_1$  and pressure  $P_3$  is done in two stages, with the intermediate pressure being  $P_2$ . At  $P_2$ , the intercooler will cool the air while its pressure remains constant\*, causing the volume of the air to decrease. This moves the process line shown in the diagram horizontally toward the isothermal process (starting point for the high pressure stage (HP) is now cooler). The energy saved per cycle, as a result of intercooling, is represented by the shaded area in Figure 2.4.

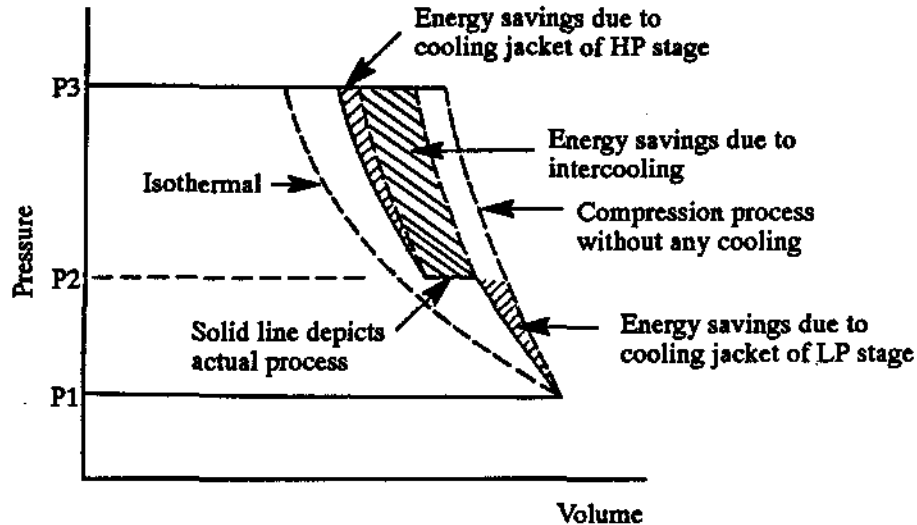
\* Pressure is maintained by steady state air flow through the compressor (otherwise, cooling would cause shrinkage and a corresponding pressure drop).



**Figure 2.4: Effect of Intercooling**

Figure 2.5 shows the overall energy saved (shaded area) using jacket cooling and intercooling on a pressure-volume diagram for a two stage compressor. As you can see, this combination more closely approximates an isothermal process.

## NOTES &amp; REFERENCES



**Figure 2.5:**  
**Combined Jacket and Intercooling Energy Savings**

*Obj. 2.4***AFTERCoolING**

Recall from your previous mechanical equipment courses that the aftercooler removes heat from the air after it leaves the compressor. Heat removed after the compression process does not contribute to any savings in work required to compress and deliver the air. But, there are three major benefits obtained from the use of aftercoolers, and each will be discussed below.

**Minimize Storage Volume and Compressor Cycling**

If we take air that has just been compressed after the last stage of compression, it will still be quite hot. If not cooled before entering the receiver, this hot air will cool in the receiver, and will **shrink in volume, causing its pressure to fall**. Aftercoolers cool the air before it reaches the air receiver, and hence **minimize compressor cycling on and off**.

The aftercooler cools the air, causing its volume to decrease, **allowing the receiver to hold more air mass**. For example, if the air is cooled from 200°C to 40°C, the volume will decrease to approximately 65% of its original volume (ie. an air receiver at 40°C will hold about 1.5 times more mass than if it was at 200°C)\*. Thus, air receiver temperature is a factor in the amount of air available after failure of the air compressor system (ie. pressure drops due to usage, leakage and due to cooling/shrinking).

*These numbers are provided for illustrative purposes only. You are not expected to remember them or perform any calculations to derive these numbers.*



## Moisture Separation

The second benefit for using aftercoolers is to help reduce the moisture content of the air after compression. By reducing the temperature of the air/water vapour mixture to below the vapour's saturation temperature (dew point), water will be condensed and then removed via a trap, before reaching the receiver. The resulting air will be saturated (ie. contain 100% of its water holding capacity) after cooling/condensation.

By removing moisture with the aftercooler and trap combination, less water will condense in the air receiver(s) and piping as the air cools further, and consequently reduce the corrosion problem\*. If the air must be dried further (ie. for use as instrument air), moisture removal via aftercooling will reduce the load on the air driers.

## Personnel Hazard Reduction

The air temperature after a compression cycle can be high enough to cause serious burns. By providing aftercoolers, the hot air is cooled immediately after the high pressure compression stage. This practice keeps piping and other system equipment at close to ambient temperatures, thus preventing injuries to personnel.

\* Water will also be condensed during the intercooling process if temperature is reduced below the saturation temperature at the intercooler pressure.

## NOTES &amp; REFERENCES

**SUMMARY OF THE KEY CONCEPTS**

- For a given mass of air, an isothermal compression and delivery cycle will require the compressor to do less work. Compressors are cooled to more closely approximate an isothermal process.
- Clearances and valve leakage will reduce volumetric efficiency, and will tend to cause compressor overheating.
- Jacket cooling removes heat during the actual compression process.
- Two stage compression with intercooling will remove heat of compression between stages, at constant pressure.
- Aftercoolers reduce air storage volume and compressor cycling. They reduce the moisture content of the air and cool the air to prevent a thermal hazard to personnel.

*Page 9***You can now do assignment questions 1-5.**

## ASSIGNMENT

1. Explain how an isothermal compression and delivery process compares to an adiabatic compression and delivery process in achieving minimum work per cycle.
2. Explain the effect of valve leakage and clearances on compressor performance.
3. Explain, with the aid of the pressure-volume diagram, the effects of jacket cooling on compressor performance.
4. Explain, with the aid of the pressure-volume diagram, the effects of two stage compression with intercooling on compressor performance.
5. Explain three benefits of using aftercoolers on air compressors.

Before you move on to the next module, review the objectives and make sure that you can meet their requirements.

Prepared by: Nick Ritter, WNTD  
Reg Harding, ENTID

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