TURBOMACHINES

A device that transfers energy between the moving fluid and rotating parts, rotating vanes, a change in the flow direction, velocity, and/or pressure of the fluid and the work is done either by the system or on the system. They may be classified as Pumps and Turbines

Centrifugal Pumps

Two most used pumps are the Centrifugal and Axial pumps.

- The **centrifugal pumps** are most widely used in *domestic and small commercial applications*.

- Whereas the **axial pumps** are used for *large and commercial applications*. The horizontal axial flow pump is designed for pumping large capacities with low lift requirements. It is designed for reliable, efficient pumping operation and for ease of maintenance.

- The principle operation of a centrifugal pump is to convert fluid velocity into pressure energy. The pump consists of three components, an inlet duct, an impeller, and a volute. Fluid enters the inlet duct (D). As the shaft (A) rotates, the impeller (B), which is connected to the shaft, also rotates. The impeller consists of a number of blades that project the fluid outward when rotating. This centrifugal force gives the fluid a high velocity. Next, the moving fluid passes through the pump case (C) and then into the volute (E). A cutout view of a single stage centrifugal pump is shown in the figure 1.
Figure 2 shows the pump characteristics curve. These curves are primarily supplied by the manufacturer. The pump characteristic curves can be defined as 'the graphical representation of a particular pump's behavior and performance under different operating conditions'. The operating properties of a pump are established by the geometry and dimensions of the pump's impeller and casing. A typical characteristic curve shows the total dynamic head, brake horsepower, efficiency, and net positive Suction head all plotted over the capacity range of the pump.

Total Pump Head = Discharge Head ± Suction Head or Lift or alternatively \( H_p = H_d \pm H_s \), Where \( H_d \) is the discharge pressure head, \( H_s \) is positive with suction head shown in Figure 3, and \( H_s \) is negative with suction lift shown in Figure 4.

**Net Positive Suction Head (NPSH)**

- NPSH is the total suction head required to avoid cavitation problems in pumps.
- In other words, NPSH is the minimum fluid energy required at the pump inlet for satisfactory operations.
- The NPSHR (NPSH required) must be positive to avoid cavitation. In general a higher NPSH is desirable. The manufacturer for all pumps provides NPSHR.
- NPSHA is the net positive suction head available or actual energy available at the pump inlet, which is calculated using equation 1.
- The pump is less prone to cavitation if the NPSHA is greater than the NPSHR. The NPSHA is calculated using equation 1 given below.
NPSHA = \( \frac{P}{\gamma} + \frac{V^2}{2g} - h_{vp} \pm (h_s) - h_L \) (On suction side only) \hspace{1cm} (1)

Where, \( h_{vp} \) = vapor pressure of the fluid corresponding to the liquid temperature (m or ft).

- \( \frac{P}{\gamma} \) = pressure head at water level on suction side,
- \( \frac{V^2}{2g} \) = velocity of water in the suction pipe at the entrance,
- \( h_s \) = Suction head or suction lift shown below, the distance from center line of pump to fluid level (m or ft);
- \( h_L \) = head loss considered in the suction line only (m or ft).

Note that if the reservoir is above the center line of the pump (Figure 3 with suction head) positive sign is considered, while for a reservoir below the pump (Figure 4 with suction lift) negative sign is be considered.

<table>
<thead>
<tr>
<th>Water Temp. °C</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
<th>60</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapor Pressure, ( h_{vp} ) (m)</td>
<td>0.125</td>
<td>0.239</td>
<td>0.435</td>
<td>0.759</td>
<td>1.275</td>
<td>2.068</td>
<td>10.78</td>
</tr>
</tbody>
</table>

**Pumps in Series or Parallel Setting**

Often in many engineering applications a single pump cannot deliver the head necessary for a particular need, and two (or more in practice) can be combined in series to increase the height to which the fluid can be pumped. If both pumps are identical then for a given flow rate, the pressure rise is doubled normally. Two pumps in series are shown in the figure 5. This is useful when a high pressure is needed but the required flow rate is sufficient from a single pump.

Similarly when the identical pumps are in parallel the flow rate is doubled but the pressure head remains the same as shown in the figure 6. When pumps run in parallel the flow is increased and the pressure head produced is around the same as a single pump.
Cavitation

Cavitation is the local vaporization of a liquid. This occurs when the absolute pressure falls to less than (or equal to) the liquid’s vapor pressure at a given temperature, resulting in small pockets of vapor and boiling occur as shown in figure 7. If the vapor bubbles come in contact with the walls at the time they collapse, pitting of the surface can occur due to the high local pressures generated. The entire process of the formation, growth, and collapse of a bubble can occur in milliseconds in a turbo-machine. There is no possible way to predict when cavitation will occur and how it can be controlled. The following parameters may be considered to avoid cavitation in pumps.

1. **NPSHA > NPSHR**.

2. **σ < σc**

   Another parameter which is used by the pump manufacturers is the cavitation number (σ). The cavitation number σ is often compared with a known parameter called critical cavitation parameter σc (a dimensionless factor provided by manufacturer), where σ is given as

   \[
   \sigma = \frac{P_{ps} - P_{vp}}{\frac{\rho V^2}{2g}} = \frac{P_{ps} - h_{vp}}{\frac{\rho V^2}{2g}}
   \]

   where

   \[P_{ps} = \text{Pressure at pump suction point}\]

3. **Nss < 8,500 (US units), and 3.0 (SI units)**

   The Hydraulic Institute recommends that for cold-water flow in a pump the Critical Suction Specific Speed \(N_{ss}\) should be less than 8,500 (US units), and 3.0 (SI units) for cavitation free operation.

   \[
   N_{ss} = \frac{\text{RPM} \sqrt{Q \text{ per eye}}}{NPSHR^{3/4}} \quad (\text{US sys.})
   \]

   \[
   N_{ss} = \frac{\text{rpm} \sqrt{Q}}{(g \ast NPSHR)^{3/4}} \quad (\text{SI sys})
   \]
- **Specific Speed**

It is a performance factor. It is a *non-dimensional speed* of the pump at which discharge and head can be estimated at best efficiency. Specific speed can be thought of as the speed required for a pump to produce unit head at unit volume flow rate. It is calculated as

For US units, \( N_s = \frac{NQ^{1/2}}{H^{3/4}} \) where, \( N = \text{rpm}, Q = \text{gpm}, H = \text{feet} \)

For SI units, \( N_s = \frac{nQ^{1/2}}{(gH)^{3/4}} \) where, \( n = \text{rps}, Q = \text{m}^3/\text{sec}, H = \text{m} \)

- **Pump Power**

\[
\begin{align*}
    h_p &= \frac{(P_2-P_1)}{\gamma} + \frac{(V_2^2-V_1^2)}{2g} + (Z_2 - Z_1) + h_L \\
    P_{\text{power}} &= \frac{\dot{m}g h_p}{\eta_p} = \frac{\gamma \dot{Q} h_p}{\eta_p} \text{ Where } \eta_p \text{ is the pump efficiency}
\end{align*}
\]

![Figure 14.14](image)

Figure 14.14, text [5]
**Affinity Laws**

Often it is important to operate a pump at a speed or head other than the published speed or head. To determine how pump will perform at speeds other than the published speed we can either look at the manufacturer’s published data or use affinity laws. These laws are used to design actual machines (pumps) on the basis of models developed in the laboratories, $Q =$flow rate, $N =$speed in rpm, $D =$diameter, $P =$Power, $H =$head

<table>
<thead>
<tr>
<th>Model (1)</th>
<th>Prototype (2)</th>
<th>Model (1)</th>
<th>Prototype (2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{Q_1}{(N_1 * D_1^3)} = \frac{Q_2}{(N_2 * D_2^3)}$</td>
<td>$\frac{P_1}{(N_1^3 * D_1^5)} = \frac{P_2}{(N_2^3 * D_2^5)}$</td>
<td>$\frac{H_1}{(N_1^2 * D_1^2)} = \frac{H_2}{(N_2^2 * D_2^2)}$</td>
<td>$\frac{Q_1}{(D_{12} * H_{11/2})} = \frac{Q_2}{(D_{22} * H_{21/2})}$</td>
</tr>
</tbody>
</table>

- So using these laws, for same diameter or homologous pumps ($D_1=D_2$) we get

\[
\frac{Q_2}{Q_1} = \frac{N_2}{N_1}, \text{ Capacity varies directly as the speed ratio}
\]

\[
\frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2, \text{ Head varies as the square of the speed ratio}
\]

\[
\frac{HP_2}{HP_1} = \left(\frac{N_2}{N_1}\right)^3, \text{ HP varies as the cube of the speed ratio}.
\]

**SI problems are Example 6 and 7**

**Example 1:** A pump (with NPSHR = 35 ft) runs at 1500 rpm with inlet area of 0.5 ft$^2$. The pump is located 2 ft (suction lift) below the reservoir open to atmosphere and moves 3 cusecs of water ($20 \degree C$) 50 ft above the pump inlet. The friction coefficient of the suction pipe (3.5 ft in length) is $f = 0.03$, the minor loss coefficient $K_s = 0.5$, while friction loss in the discharge side is 4 ft. The water is dumped in the second reservoir which is also open to atmosphere.

Find:
- NPSHA, total pump head, specific speed.
• What type of pump would you recommend? What is the operating efficiency?
• The Suction specific speed $N_{ss}$. Is this pump would be susceptible to cavitation at these conditions and when the pump speed is increased to 5,000 rpm?

Find: $NPSHA$, $Pump$ head, $Specific$ $Speed$, $Suction$ $Specific$ $Speed$, $Pump$ $type$ $recommendation$, $operating$ $efficiency$

$Suction$ $Velocity\:=\frac{Q}{A} = 6$ ft/sec, $D_1 = \sqrt{\frac{4*0.5}{\pi}} = 0.798$ ft

$Suction$ $side$ $head$ $loss$ $h_L = \frac{fL}{D} * \frac{V^2}{2g} = \frac{0.03+3.5}{0.798} * \frac{6^2}{2*32.2} + 0.5 * \frac{6^2}{2*32.2} = 0.353$ ft

$NPSHA = \frac{p}{\gamma} + \frac{V^2}{2g} - h_{vp} \pm h_s - h_L \ (+sign$ $for$ $suction$ $head, -sign$ $for$ $suction$ $lift)$

$= \frac{14.7*144}{62.4} + \frac{6*6}{2*32.2} - 0.239*3.281 - 2 - 0.353$

$= 33.923 + 0.56 - 0.7842 - 2.353 = 31.346$ ft.

The pump operation will be prone to cavitation, since $NPSHA < NPSHR = 35$

$Specific$ $Speed$ $N_s = \frac{NQ^{1/2}}{H^{3/4}}$, here $Q = 3*448.83$ gpm = 1346.5 gpm

So that $N_s = \frac{1500*1346.5^{1/2}}{56.353^{3/4}} = 2676$

Select Radial pump and $Pump$ $Efficiency$ of 90% from the chart on page 5,

$Pump$ $Power$ $P_p = \frac{\gamma Q h_p}{550*\eta_p} = \frac{62.4*3*56.35}{550*0.9} = 21.31$ HP

$Suction$ $Specific$ $Speed$ $@1500 \ rpm$, $N_{ss} = \frac{1500*\sqrt{1346.5}}{35^{3/4}} = 3825$, ∴ $N_{ss} <<8500$

Therefore at this speed, the pump will not be prone to cavitation

$Suction$ $Specific$ $Speed$ $@5000 \ rpm$, $N_{ss} = \frac{5000*\sqrt{1346.5}}{35^{3/4}} = 12,750$, ∴ $N_{ss} >>8500$

Therefore at this speed, the pump will be prone to cavitation
Example 2: If the manufacturer’s critical cavitation parameter for the pump in the
Problem 1 is “60”, whether this pump would cavitate at the given operating
conditions.

The cavitation parameter is \( \sigma = \frac{\frac{P_{ps}}{\gamma} - \frac{h_{vp}}{2g}}{\sigma_c} \) given \( \sigma_c = 60 \)

Apply Energy Equation from reservoir (1) to pump suction(2). \( P_2 = P_{ps} \)

\[
\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + h_L \Rightarrow \frac{P_2}{\gamma} = \frac{P_{ps}}{\gamma} \cdot 2 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + 0 + 0.353
\]

\[
\frac{P_2}{\gamma} = \frac{14.7 \cdot 144}{62.4} - \frac{6^2}{2g} - 2.353 \Rightarrow \text{So that } \frac{P_2}{\gamma} = \frac{P_{ps}}{\gamma} = 33.92 - 0.559 \cdot 2.353 = 31 \text{ ft}.
\]

Thus \( \sigma = \frac{31.0 - 0.784}{0.559} = 54.07 \)

\(\sigma << \sigma_c\), the pump will not be prone to cavitation.

Example 3: A 2-ft diameter pump produces 30 ft head and delivers 500 gpm at
1000 rpm. Determine the performance parameters (pump head, Sp. Speed, Power
etc.) at 1100 rpm and select this pump and its efficiency. Now assume that power
required of two pumps are the same \( (P_1 = P_2) \), determine the new diameter, gpm,
head of the new pump and select a pump and its efficiency for this case.

Given \( D_1 = 2 \text{ ft}, \) \( H_1 = 30 \text{ ft}, \) \( Q_1 = 500 \text{ gpm}, \) \( N_1 = 1000 \text{ rpm} \)

@ \( N_2 = 1100 \text{ rpm}, \) \( \text{Find: } Q_2, H_2, P_2, \text{ and Select a pump} \)

Now if pump power \( P_1 = P_2, \) \( \text{Find: } D_2, Q_2, H_2, \text{ and Select a pump} \)

Use Affinity Laws:

\[
\frac{H_1}{N_1^2 \cdot D_1^2} = \frac{H_2}{N_2^2 \cdot D_2^2} \Rightarrow \frac{H_1}{N_2^2} = \frac{H_2}{N_1^2} \Rightarrow \frac{30}{1000^2} = \frac{H_2}{1100^2} \Rightarrow H_2 = 30 \cdot 1.1^2 = 36.3 \text{ ft}
\]

\[
\frac{Q_1}{N_1 \cdot D_1} = \frac{Q_2}{N_2 \cdot D_2} \Rightarrow \frac{Q_1}{N_1} = \frac{Q_2}{N_2} \Rightarrow Q_2 = \frac{(N_2}{N_1}) \cdot Q_1 = 500 \cdot 1.1 = 550 \text{ gpm}
\]

\[
\frac{P_1}{N_1^3 \cdot D_1^5} = \frac{P_2}{N_2^3 \cdot D_2^5} \Rightarrow \frac{P_1}{N_1^3} = \frac{P_2}{N_2^3} \Rightarrow P_2 = \left(\frac{N_2}{N_1}\right)^3 P_1 \Rightarrow P_2 = (1.1)^3 P_1 = 1.331 P_1
\]

Apply Affinity laws for same pump power.
Example 4: A test is conducted on a pump, which develops a discharge head of 12 meters while delivering 400 gpm of water at 60 °C, when operated at 1000 rpm. Determine the specific speed, suction specific speed and select the pump type and the power required to operate this pump. The losses in the suction and discharge sides of the pump are 2 m and 4 m. (1 m³/sec = 15789 gpm)

\[
\frac{P_1}{N_1^{3/4}D_1^{5}} = \frac{P_2}{N_2^{3/4}D_2^{5}} \quad D_2 = \left[\left(\frac{N_1}{N_2}\right)^3 \ast 2^5\right]^{1/5} = \left[\frac{1000}{1100}\right]^{3/4} \cdot 2^5^{1/5} = 1.8888 \text{ ft}
\]

Also \( N_s = \frac{N_1Q_1^{1/2}}{H_1^{3/4}} = \frac{1000 \cdot (500)^{1/2}}{30^{3/4}} = 1744 \), Radial flow pump, Efficiency= 90%

Now \( P_1 = \frac{\gamma QH_1}{550 \cdot 0.90} = \frac{62.4 \cdot 500 \cdot 30}{448.8 \cdot 550 \cdot 0.9} = 4.21 \text{ hp} \quad : \quad P_2 = 1.331 \ast P_1 = 15.605 \text{ hp}
\]

Again, \( N_s = \frac{N_2Q_2^{1/2}}{H_2^{3/4}} = \frac{1100 \cdot (550)^{1/2}}{36.3^{3/4}} = 2012 \), Mixed flow pump, efficiency= 92%

Here Pump head = discharge head + (suction side + discharge side) head loss

\[
= 12 + 6 = 18 \text{ m}
\]

Specific Speed, \( N_s = \frac{NQ^{1/2}}{(gH)^{3/4}} \), where, \( N = \text{rps}, Q = \text{m}^3/\text{sec}, H = \text{m}, \)

For \( N_s = \frac{1000 \ast (400)^{0.5}}{(9.81 \ast 18)^{3/4}} = 0.0055 \), From Chart the is 83% and it is a radial flow pump, and Pump Power = \( \frac{\gamma QH}{746 \cdot 0.83} = \left(\frac{9810 \cdot 400}{746 \cdot 0.83}\right)^{18} = 7.225 \text{ hp} \)

Example 5: An 80% efficient pump delivers 100 gpm with discharge head of 3 ft. Determine the pump power. The suction and discharge pipe diameter is 3 in. and 1 in. respectively. The pressure difference between inlet (\( P_1 \)) and outlet (\( P_2 \)) is 6 psi and the total head loss in suction side is 2 ft.
Find $V_1 = \frac{100}{448.8} \times \frac{4}{\pi \times \left(\frac{3}{12}\right)^2} = 4.54 \text{ ft/sec}$, $V_2 = \frac{100}{448.8} \times \frac{4}{\pi \times \left(\frac{1}{12}\right)^2} = 40.85 \text{ ft/sec}$,

$P_1 - P_2 = 6 \times 144 \text{ psf}$, $h_{L_{-\text{suction}}} = 2 \text{ ft}$, $h_{L_{-\text{discharge}}} = \text{not given} = 0$,

Applying energy eqn. $h_p = \frac{P_2 - P_1}{\gamma} + \frac{V_2^2 - V_1^2}{2g} + (Z_2 - Z_1) + \sum h_L = \frac{-6 \times 144}{62.4} + \frac{40.85^2 - 4.54^2}{64.2} + 5$

$\Rightarrow h_p = 16.483 \text{ ft}$, so that Power $= \frac{\gamma \times Q \times h_p}{550 \times \eta_p} = 62.4 \times \frac{100}{448.8} \times \frac{16.483}{550 \times 0.8} = 0.53 \text{ hp}$

SI problems are given at the end
SI Unit Problems

Example 6) A test is conducted on a pump and test data show the pump capable of developing a total head of 15 m while delivering 20 liters/sec when operated at 1000 rpm. Estimate the head and capacity when the pump is operated at 800 rpm. Determine the initial specific speed of the pump.

\[
H_1 := 15 \cdot m \quad N_1 := 1000 \quad \text{RPM} \quad n_1 := 16.667 \quad \text{Rev/s}
\]

\[
Q_1 := 20 \cdot \frac{L}{s} \quad Q_1 = 0.02 \cdot \frac{m^3}{s} \quad N_2 := 800 \quad \text{RPM}
\]

Find \(H_2\), \(Q_2\), and \(D_2 = D_1\). Apply affinity laws equations (8 and 9). Since same pump is used \(D_2 = D_1\)

\[
Q_2 = \frac{Q_1}{N_1} \cdot N_2 \Rightarrow Q_2 := \frac{0.02}{1000} \cdot 800 \Rightarrow Q_2 = 0.02 \cdot \frac{m^3}{s}
\]

Also \(\frac{H_1}{N_1^2} = \frac{H_2}{N_2^2}\) Which gives \(H_2 := \frac{800^2}{1000^2} \cdot 15 \Rightarrow H_2 = 9.6 \cdot m\)

Again specific speed

\[
N_{s1} := \frac{n_1 \cdot Q_1}{\frac{3}{4} \cdot g \cdot H_1^4} = \frac{16.667 \cdot 0.02^2}{\frac{3}{4} \cdot 9.81^4 \cdot 15^4} = 0.06
\]

Select Centrifugal pump from Figure 14.14 text

Example 7) A pump delivers 95 liters/min of water at 70 °C (\(\mu = 4.04 \times 10^{-4} \text{ N-s/m}^2\), \(h_{vp} = 31.2 \text{ kPa}, \rho = 978 \text{ Kg/m}^3\)) through 1.5 inch steel pipe (\(K_s = 0.046 \text{ mm}\)). The inlet pressure is 20 Kpa vacuum. The atmospheric pressure is 1 bar. The pump is operating at a suction head of 2.5 m. Consider that there is a wide open globe valve (\(K_v = 10\)) and one bend (\(K_b=0.9\)) in the system. Determine the NPSHA. If the manufacturer’s NPSHR is 6 m, should this pump be recommended for the intended operation? If the pump operates at 2500 rpm would it be prone to cavitation.
For problem description reload Turbomachine notes from my Website

At 70 Deg C Vapor Pressure $h_{vp} = 31200$ Pa. or 31.2 Kpa; Atmosphere Pressure is 100 Kpa

$$NPSHA := \left[ \frac{P}{\gamma} + \frac{V^2}{2 \cdot g} + \text{or} - \left( h_s \right) \right] - h_{vp} - h_L$$

$$P := -20 \cdot 10^3 \text{ pa}$$

$$h_s := 2.5 \cdot m$$

$$h_L = f \cdot \frac{L \cdot V^2}{D \cdot 2 \cdot g} + K_v \cdot \frac{V^2}{28 \cdot g}$$

In absence of actual pipe data assume Sch. 40 Pipe with Inlet dia. = 1.5"=1.5*2.54 cm=3.81 cm

$$Q := 95.0 \cdot \frac{L}{\text{min}} \quad Q := 1.583 \times 10^{-3}$$

$$V = \sqrt{\frac{1.583 \cdot 10^{-3}}{\pi \cdot \left( \frac{1.5 \cdot 2.54}{100} \right)^2}} = 1.39 \frac{m}{s}$$

Pipe (Assume Steel) $K_s := 0.046 \cdot \text{mm}$

$$\frac{K_s}{D} = \frac{0.00046}{0.0381} = 0 \quad \text{Re} = \frac{\rho \cdot V \cdot D}{\mu} = \frac{978 \cdot 1.3885 \cdot 0.0381}{4.04 \cdot 10^{-4}} = 1.28 \times 10^5$$

From Moody Diagram $f := 0.23$

Loss coefficient for fully open valve (Globe) - $K_v := 10 \quad K_{bend} := 0.9 \quad K_{v,\text{inlet}} := 0.5$

$$K_{v,\text{total}} := K_v + K_{bend} + K_{v,\text{inlet}} \quad K_{v,\text{total}} = 11.4$$

Such that:

$$h_s := 0.023 \cdot 12 \cdot \frac{1.3885^2}{2 \cdot 9.81} + 11.4 \cdot \frac{1.3885^2}{2 \cdot 9.81}$$

$$\therefore \ h_L = 1.83 \text{ m},$$

Now Calculate $NPSHA = \frac{80000}{9810} + \frac{1.3885^2}{2 \cdot 9.81} - \frac{31200}{9810} + 2.5 - 1.832 = 5.74$
a) To avoid cavitation the pump selected must have a NPSHA of at least 5.74 meters! Since the required NPSHR is 6 m, this pump will therefore be prone to Cavitation.

b) If pump operates at 2500 RPM
\[
N_{ss} = \frac{\frac{2500}{60} \cdot \sqrt{0.001583}}{\frac{3}{(9.81 \cdot 5.741^4)}} \Rightarrow N_{ss} = 0.08
\]

A pump with \(N_{ss} = 0.0806\), would pose no problem for cavitation. But this does not mean the pump is free from problems, as we have noted in Part a of the problem.

Pump Problems – Assigned

1) A pump delivers water at 30 °C from a lower reservoir to an upper reservoir. The pump outlet is 2 m above the pump exit, and static pressure taps indicate an outlet pressure that is 100 kPa greater than the inlet pump pressure. If the flow rate is 0.05 m³/sec and inlet diameter of the pipe leading to the pump is 100 mm while the outlet diameter is 50 mm, calculate the power added to the water. If the lower reservoir is at the atmospheric pressure equivalent to 1 bar and it is 1 m below the pump level, calculate NPSH. Assume no loss in the pipe lengths except at the inlet and exit points of the system.

2) A pump is to deliver 4000 liters/min against a head of 20 m. If the pump is operated at 3600 rpm, select the type of pump to be used.

3) A pump has an impeller diameter of 1.5 meters and operates at 1200 rpm. If the speed is increased to 1400 rpm, what diameter impeller should be used to maintain a constant power input to the pump?

4) What type of pump should be selected that will deliver 2000 gpm against a head of 25 ft while operating at 3600 rpm?

References
5. Text Book