# Centrifugal Machines

## Table of Contents

1. Introduction and Basic Principles ................................................................. 2
   1.1 Hydraulic Machines ............................................................................... 2
   1.2 Centrifugal Machines .......................................................................... 2
   1.3 Pump Geometry ................................................................................... 2
   1.4 Pump Blade Geometry ......................................................................... 3
   1.5 Diffusers .............................................................................................. 5
   1.6 Pump Losses ........................................................................................ 6
   1.7 Example Pump Calculation ................................................................. 7
      1.7.1 Solution ........................................................................................ 7
2. Dimensional Analysis, Specific Speed and Cavitation. ......................... 9
   2.1 Specific Speed for Pumps .................................................................... 9
   2.2 Using Specific Speeds. ......................................................................... 10
   2.3 Multistage Pumps ............................................................................... 11
   2.4 Cavitation ............................................................................................ 12
   2.5 Basic Mechanism and Impacts ........................................................... 12
   2.6 Net Positive Suction Head .................................................................. 12
   2.7 Pump Cavitation Criterion .................................................................. 13
   2.8 Industrial Practise ............................................................................... 14
3. Machine and System .................................................................................. 15
   3.1 System Matching ................................................................................ 16
   3.2 Machine System Example .................................................................... 17
4. Industrial Practise for Selecting a Pump .................................................. 19
1 Introduction and Basic Principles

1.1 Hydraulic Machines
A hydraulic machine is one which has the working fluid as a liquid.

- The simplification for hydraulic machines is that the fluid density is constant
- The complication is the cavitation may occur in the machine. (More on that later!)

Hydraulic machines can be either turbines or pumps but we consider pumps only in this course although the principles remain exactly the same.

1.2 Centrifugal Machines
A centrifugal machine is strictly one where the flow enters the blades axially and then leaves in the radial direction, however the term is often applied to any machine enters axially and leaves radially regardless of the details of the blading design. Although every centrifugal machine is not a hydraulic machine to make the course content compact we will consider only machines that are both hydraulic and centrifugal in this week’s course.

Note there is no new theory this week just the application of analysis tools already developed.

1.3 Pump Geometry

\[ \text{Pure Radial} \rightarrow \text{Centrifugal / Mixed Flow} \rightarrow \text{Axial} \rightarrow \text{Lower H, Higher Q for the Same Power} \]

3D views should make the difference between a radial and a centrifugal pump clearer.
**Exercise 16.** Draw velocity triangles at impeller inlet and exit for a radial pump impeller. Draw the same velocity triangles for a centrifugal pump impeller.

If 1 is inlet to rotor and 2 the outlet, then Euler Work Equation gives:

Power, \( P = \dot{m} \omega (R_2 V_{\theta 2} - R_1 V_{\theta 1}) \)

Efficiency, \( \eta_p = \frac{\text{Ideal Power}}{\text{Actual Power}} = \frac{\dot{m} g H}{\dot{m} \omega (R_2 V_{\theta 2} - R_1 V_{\theta 1})} \)

So Head, \( H = \eta_p \frac{\omega}{g} (R_2 V_{\theta 2} - R_1 V_{\theta 1}) \)

Thus for high head make \( R_2 > R_1 \), as well as the change in \( V_0 \).

### 1.4 Pump Blade Geometry

Key information in Euler equation is tangential velocities is \( V_{\theta 1} \) and \( V_{\theta 2} \). So velocity triangle at 2:

From exit velocity triangle:

\[ V_{\theta 2} = \omega r_2 + \omega R_2 \tan \beta_2 + \omega R_2 \]

How to get \( V_{\theta 2} \)?

We go back to the continuity equation but this time apply it in the radial direction.
Flowrate:  \[ Q = 2\pi R_1 b_2 (1-t_2)V_{r2} \Rightarrow V_{r2} = \frac{Q}{2\pi R_2 b_2 (1-t_2)} \]

Note that here we have introduced the variable \( t \) which is the blade thickness, this is expressed as a faction of passage area so if the blades occupy 8% of the area \( t = 0.08 \)

Put the two together:

\[ V_{\theta 2} = \frac{Q \tan \beta_2}{2\pi R_2 b_2 (1-t_2)} + \omega R_2 \]

At inlet if we assume \( V_{\theta 1} = 0 \) and recall \( H = \eta_p \frac{\omega}{g} \left( R_2 V_{\theta 2} - R_1 V_{\theta 1} \right) \)

\[ H = \frac{\eta_p}{g} \left( \frac{\omega \tan \beta_2}{2\pi b_2 (1-t_2)} \right) Q + \omega^2 R_2^2 \]

(1)

By inspection of this equation for constant efficiency and blade angle \( \eta_p = \text{const}, \beta_2 = \text{const} \) head is linear with flowrate. If the flowrate is zero the head depends on the rotational speed and the outer radius only. The power in a pump can be shown to be as follows:

\[ P = \rho \left[ \frac{\omega \tan \beta_2}{2\pi b_2 (1-t_2)} Q^2 + \omega^2 R_2^2 Q \right] \]

(2)
Exercise 17. Sketch curves of \( H \) vs \( Q \) and \( P \) vs \( Q \) for forward leaning, backward leaning and straight bladed pumps.

Backward Leaning is generally preferred to Forward Leaning, in spite of the lower head because:

1. Higher efficiency due to lower exit swirl, leading to less diffuser loss.
2. Forward leaning have an increasingly steep power curve – if \( Q \) is underestimated, the power requirement will be greatly underestimated so that an electrical drive motor may burn out.

1.5 Diffusers

For a machine with radial exit flow from the rotor, we may reduce the kinetic energy either by means of a vaned diffuser, or a vaneless diffuser. For a vaneless diffuser, as the flow goes out radially, by continuity:

\[
V_r \times R = \text{Constant} = K_1 \quad \text{or} \quad V_r = K_1 / R
\]

By conservation of angular momentum:

\[
V_\theta \times R = \text{Constant} = K_2 \quad \text{or} \quad V_\theta = K_2 / R
\]

Therefore exit k.e.:

\[
\frac{V^2}{2} = \frac{V_r^2 + V_\theta^2}{2} = \frac{1}{2R^2} (K_1^2 + K_2^2)
\]

so k.e. reduces as

\[
\frac{V^2}{2} \propto \frac{1}{R^2}
\]

But pressure rise may cause separation, so vanes may be required to control the flow.
1.6 Pump Losses

Losses are often expressed as fraction of inlet dynamic head

**Impeller:** \( \Delta H_{\text{IMP}} = K_{\text{IMP}} \frac{W_1^2}{2g} \), use *relative* inlet dynamic head, \( \frac{W_1^2}{2g} \)

**Diffuser:** \( \Delta H_{\text{Diff}} = K_{\text{Diff}} \frac{V_2^2}{2g} \)

Where \( K_{\text{IMP}} \) and \( K_{\text{Diff}} \) are the impeller and diffuser Loss Coefficients

The diffuser is a major source of loss. The loss may also be expressed in terms of *diffuser efficiency*: \( \eta_{\text{Diff}} = \frac{\Delta p_{\text{act}}}{\Delta p_i} \), where \( \Delta p_i \) is the ideal pressure rise with diffusion to zero velocity.

Thus \( \Delta p_i = \frac{1}{2} \rho V_2^2 \) and \( \eta_{\text{Diff}} = \frac{\Delta p_{\text{act}}}{\frac{1}{2} \rho V_2^2} = \frac{p_3 - p_2}{\frac{1}{2} \rho V_2^2} \)

Now \( H_2 - H_3 = \Delta H_{\text{Diff}} \), the diffuser total head loss, where \( H = \frac{p}{\rho g} + \frac{V^2}{2g} \)

\[
\Rightarrow \frac{p_3}{\rho g} - \frac{p_2}{\rho g} = \frac{V_2^2}{2g} - \Delta H_{\text{Diff}} \quad \text{(Ignoring gravity and } V_3 = 0) \]

\[
\Rightarrow \frac{p_3 - p_2}{\frac{1}{2} \rho V_2^2} = 1 - \frac{\Delta H_{\text{Diff}}}{\frac{1}{2} \rho V_2^2} \]

or the relationship between Diffuser Efficiency and Loss Coefficient is

\( \eta_{\text{Diff}} = 1 - K_{\text{Diff}} \)
1.7 Example Pump Calculation

Consider a centrifugal pump with data:

- At impeller inlet, mean radius \( R_{m1} = 0.1 \ m \), blade height \( b_1 = 0.1 \ m \)
- At impeller exit, radius \( R_2 = 0.2 \ m \); blade height, \( b_2 = 0.03 \ m \); blade blockage, \( t = 0.08 \)
- Slightly backward leaning blades with \( \beta_2 = -10^\circ \)
- Flowrate of water, \( Q = 0.2 \ m^3/s \); Rotational speed, \( N = 1450 \ rev/min \)
- Assume for losses: Impeller, \( K_{imp} = 0.1 \); Diffuser efficiency = 60%

Calculate: Relative flow at impeller inlet; and absolute flow at impeller exit, Power required; Head produced; Pump efficiency.

1.7.1 Solution

First Sketch pump and velocity triangle at inlet:

Continuity \( Q = 2 \pi R_{m1} b_1 V_{x1} \)

\[
V_{x1} = \frac{Q}{2 \pi R_{m1} b_1} = \frac{0.2}{2 \pi \times 0.10 \times 1} = 3.183 \ m/s
\]

Angular Velocity:

\[
\omega = \frac{2 \pi N}{60} = \frac{2 \pi \times 1450}{60} = 151.8 \ rad/s
\]

Blade speed:

\[
U_1 = \omega R_{m1} = 15.18 \ m/s
\]

From velocity triangle:

\[
W_i^2 = V_{x1}^2 + (\omega R_{m1})^2
\]

\[
\Rightarrow W_i = \sqrt{3.181^2 + 15.18^2} = 15.51 \ m/s
\]

\[
\beta_1 = \tan^{-1} \left( \frac{W_{\alpha1}}{V_{x1}} \right) = -\tan^{-1} \left( \frac{\omega R_{m1}}{V_{x1}} \right)
\]

Therefore

\[
\beta_1 = -\tan^{-1} \left( \frac{15.18}{3.183} \right) = -78.1^\circ
\]

Sketch pump configuration and velocity triangles at exit.
Centrifugal Machines

Continuity:
\[ Q = 2\pi R_2 b_2 (1-r_2) V_{r_2} \]
\[ V_{r_2} = \frac{0.2}{2\pi \times 0.20 \times 0.03 \times (1-0.08)} = 5.766 \text{ m/s} \]

Blade speed:
\[ U_2 = \omega R_2 = 151.8 \times 0.2 = 30.36 \text{ m/s} \]

From velocity triangle:
\[ V_{\theta_2} = W_{\theta_2} + \omega R_2 \]
and
\[ W_{\theta_2} = V_{r_2} \tan \beta_2 = 5.766 \tan 10^\circ = -1.016 \text{ m/s} \]
\[ V_2 = \sqrt{V_{\theta_2}^2 + V_{r_2}^2} = \sqrt{29.34^2 + 5.77^2} = 29.9 \text{ m/s} \]
\[ \alpha_2 = \tan^{-1} \left( \frac{V_{\theta_2}}{V_{r_2}} \right) = \tan^{-1} \left( \frac{29.34}{5.77} \right) = 78.8^\circ \]

Power required:
\[ w_x = \omega (R_2 V_{\theta_2} - R_{ml} V_{\theta_1}) = 151.8 \times (0.2 \times 29.34 - 0) = 890.7 \text{ W/kg} \]
\[ P = \rho Q w_x = 1000 \times 0.2 \times 890.7 = 178 \text{ kW} \]

Ideal head is given by
\[ w_x = g \Delta H_{\text{ideal}} \Rightarrow \Delta H_{\text{ideal}} = \frac{w_x}{g} = \frac{890.7}{9.81} = 90.7 \text{ m} \]

Actual head is given by
\[ \Delta H_{\text{actual}} = \Delta H_{\text{ideal}} - \sum \Delta H_{\text{losses}} \]
\[ \Delta H_{\text{imp}} = K_{\text{imp}} \frac{W_{\theta_2}^2}{2g} = \frac{0.1 \times 15.51^2}{2 \times 9.81} = 1.22 \text{ m} \]
\[ \Delta H_{\text{diff}} = K_{\text{diff}} \frac{V_2^2}{2g} = (1-\eta_{\text{diff}}) \frac{V_{r_2}^2}{2g} = \frac{0.4 \times 29.9^2}{2 \times 9.81} = 18.2 \text{ m} \] Note this is much larger than the impeller loss.
So \[ \Delta H_{\text{actual}} = 90.7 - 1.22 - 18.2 = 71.28 \text{ m} \]

Efficiency:
\[ \eta = \frac{\Delta H_{\text{actual}}}{\Delta H_{\text{ideal}}} = \frac{71.3}{90.7} = 0.786 \]
2 Dimensional Analysis, Specific Speed and Cavitation.

We use similarity for two reasons. i) given data on one size of machine we can predict performance at different sizes, ii) given data on one set of operating conditions predict behaviour at different operating conditions.

If we ignore viscous effects, for hydraulic machines relevant variables are:

<table>
<thead>
<tr>
<th>Variable</th>
<th>Parameter</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>D</td>
<td>Rotor diameter</td>
<td>L</td>
</tr>
<tr>
<td>Q</td>
<td>volumetric flow rate</td>
<td>L^3T^-1</td>
</tr>
<tr>
<td>N</td>
<td>rotational speed</td>
<td>T^-1</td>
</tr>
<tr>
<td>H</td>
<td>head difference across machine</td>
<td>L</td>
</tr>
<tr>
<td>g</td>
<td>acceleration due to gravity</td>
<td>LT^-2</td>
</tr>
<tr>
<td>ρ</td>
<td>density of fluid</td>
<td>ML^-3</td>
</tr>
<tr>
<td>P</td>
<td>power transferred between fluid and rotor</td>
<td>ML^2T^-3</td>
</tr>
</tbody>
</table>

Application of Buckingham Pi Theorem (left as a job for the reader) yields:

\[ \Pi_1 = \frac{Q}{ND^3}, \quad \Pi_2 = \frac{gH}{N^2D^2}, \quad \Pi_3 = \frac{P}{\rho N^3D^5} \]

2.1 Specific Speed for Pumps

Note that \( \frac{\Pi_1}{\Pi_1 \Pi_2} = \frac{P}{\rho Q g H} = \frac{1}{\eta} \) for a pump. So if for a model pump we obtain test data and plot the dimensional parameters against each other we can predict the performance of larger machines of the same type. For a particular machine the operating conditions are expressed by values of N, Q and H. It would be useful to have a dimensional group which includes N, Q and H but not D.

We create a new non-dimensionless group: \( \frac{\Pi_1^{1/3}}{\Pi_2^{1/2}} = \frac{Q^{1/3}N^{2/3}}{(gH)^{1/2}} \)

Engineers aim for maximum efficiency \( \eta \) and there is generally only set of values \( \Pi_1 \) and \( \Pi_2 \) etc. for which this occurs. For a given shape of machine we are therefore interested in a unique value of \( \Pi_3 \).
For convenience raise $\Pi_5$ to the power 2/3 to get the dimensionless specific speed:

$$K_s' = \frac{NQ^{1/2}}{(gH)^{3/4}}$$

$K_s$ has a **unique** value for **maximum efficiency** for a given shape of machine.

It is a **Shape Parameter** independent of the size, D. By calculating the value of $K_s$ from the design specifications, we can determine the **type of machine required**.

Normally engineers stay on planet earth so $g$ is dropped to give a **dimensional Specific Speed**:

$$N_s = \frac{NQ^{1/2}}{H^{3/4}}$$

For $N$ in rev/min, $Q$ in m$^3$/s and $H$ in m, $N_s = 333 K_s'$

When using dimensional numbers it is essential that you know what units are being used. **Practise varies widely around the globe.**

The term Specific Speed arises from the idea of a given type of machine deliver unit flow rate (1 m$^3$/s) at unit head (1m) - $N_s$ is the speed it would run at in rev/min. **It is much better to think of it as a Shape Parameter.**

For commercial machines there are size limitations: i) upper limit from material strength (centrifugal forces) and ii) lower limit, clearance gets large and low Reynolds number lead to reducing efficiency. This gives the range of useful head.

The process can also be carried out for turbines but this is not part of this course.

### 2.2 Using Specific Speeds.

Pump and turbine manufacturers will provide charts and diagrams that relate pump (or turbine) geometry to specific speed. Be sure to use the same units as the person who made the chart!!
For this chart note that $H = \text{Head in feet}$, $N = \text{RPM}$, $Q = \text{US Gallon Per Minute}$.

### 2.3 Multistage Pumps

So far we have considered single stage pumps, but it is possible to put more than one stage in the casing to get a multi-stage pumps. To keep everything on the same shaft line the flow path is a little convoluted and the impellers are often mounted in the opposite direction balance up axial thrust forces.

The diagram features five impellers, the central one is double flow and the outer two are single flow mounted in opposite directions.

The line diagram below shows the flow through the machine:

Source: Neptuno Pumps
2.4 Cavitation

There are certain phenomena that have not been considered in the dimensional analysis (e.g. the viscosity effects, Re). Another important phenomenon, often encountered in hydraulic machine applications, is cavitation.

2.5 Basic Mechanism and Impacts

If the absolute pressure falls below the saturated vapour pressure, \( p_v \), (for the water temperature) then bubbles of vapour will be formed. As the pressure rises, the bubbles collapse suddenly and very high instantaneous pressures can be created (>500 bar) which can cause significant negative impacts:

- Machine Mechanical integrity: erosion of blades, valves etc (See photos)
- Hydrodynamic flow performance: air bubble blockage, increased flow loss – lower efficiency and reduced range of operation.


2.6 Net Positive Suction Head

Consider the diagram above, the pump inlet station zero is at atmospheric pressure. The impeller inlet station 1 must be at a lower pressure \( p_1 \) to ensure flow into the pump.
For no cavitation \( p_1 > p_v \) you will recall that \( p_v \) is the saturated vapour pressure of a liquid which can be found from steam tables. It is easier to work in terms of head \( h = \frac{p}{\rho g} \)

For satisfactory operation we will need some margin between the saturated pressure vapour and the pressure at inlet to the impeller this is called the Net Positive Suction Head or NPSH for short

\[
\frac{p_1 - p_v}{\rho g} = h_1 - h_v = NPSH
\]

Consider the total head change from pump inlet 0 at pressure \( p_a \) and height \( z = z_0 = 0 m \) to impeller inlet 1 where \( z = z_1 \)

\[
H_0 = H_1 + h_f \quad \text{where} \quad h_f \text{ is the frictional head loss}
\]

\[
\Rightarrow \frac{p_a}{\rho g} = \frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1 + h_f \Rightarrow h_a = h_1 + \frac{V_1^2}{2g} + z_1 + h_f \quad \text{rearranging this:}
\]

\[
h_1 = h_a - \frac{V_1^2}{2g} - z_1 - h_f \quad \text{recall that} \quad NPSH = h_1 - h_v \quad \text{so:}
\]

\[
NPSH = h_a - h_v - \frac{V_1^2}{2g} - z_1 - h_f
\]

Note this is an important value in pump design and selection.

### 2.7 Pump Cavitation Criterion

Academics text books sometimes use Thoma’s Parameter, \( \sigma = \frac{NPSH}{\Delta H} \), \( \text{(non-dimensional)} \) where \( \Delta H \) is the total head rise across the pump. Cavitation occurs when \( \sigma = \sigma_{crit} \) where \( \sigma_{crit} \) is a critical value of Thoma’s parameter, which is empirically given. For safe operation: \( \sigma > \sigma_{crit} \)

Low NPSH (low \( \sigma \), leading to cavitation) can result from: a) too high flow velocity, \( V_1 \) OR b) too high pump elevation, \( z_1 \). e.g. consider a pump characteristic where the outlet is restricted:

As \( Q \) increases NPSH is reduced because the velocity head \( \frac{V_1^2}{2g} \) rises and also \( h_1 \) rises. Cavitation results in the maximum flow being restricted no matter how much the exit valve is opened. (Note difference with Lab experiment where the inlet is restricted)
2.8 Industrial Practise

Industrial practise splits Net Positive Suction Head NPSH into two parts:

**NPSHR** – “net positive suction head required” characterize suction ability of pump and is determined by pump supplier (see below)

**NPSHA** – “net positive suction head available” is done by the pumping and piping system on the suction side of your pump. (See later on Machine and System).

It is necessary for correct pumping: NPSHA > NPSHR
3 Machine and System

This lecture examines how various components in a pipe/pump system work together. Consider a pump operating between Reservoir, R1, and R2, with a height difference $h_s$.

For the whole system to work properly, the head rise produced by the pump $H$, has to overcome $h_s$ and the frictional losses: 

$$H = h_s + \text{losses} \quad \text{(the pump performance has to match the pipe system performance)}$$

Often flow losses are proportional to the velocity head $\frac{V^2}{2g}$

Friction loss: 

$$h_f = 4 f \frac{L}{d} \frac{V^2}{2g}$$

Inlet / exit loss: 

$$h_i = K_i \frac{V^2}{2g}$$

Bend loss: 

$$h_b = K_b \frac{V^2}{2g}$$

Thus often we have a total head rise required for system in a form of: 

$$H = h_s + C Q^2$$

On the other hand, the characteristic for the pump can be very different:

**Exercise 19:** Sketch a system characteristic and a pump characteristic on the same curve and hence determine the operating point of the pump and system.
3.1 System Matching

We can plot $H$ vs. $Q$, the System Characteristic on the same graph as the Pump Characteristic. The intersection gives the operating point (i.e. the condition at which the system matches). The effect of a larger pipe (reduces losses for a given $Q$, or the effect of running the pump at a different speed (or different pump size) can easily be seen.

Notes:
Machine dimensionless parameters can be used to scale the pump characteristic:

$$\frac{gH}{N^2 D^2}$$

$$\frac{Q}{N D^3}$$

Efficiency can also be plotted against $Q$ and the value read off for the operating point. Thus the pump power can be determined also (or the power characteristic may be plotted directly).
### 3.2 Machine System Example

A pump manufacturer has provided data on the performance of a pump with impeller diameter 0.25m operating at 1450 rev/min whilst pumping water at a density of 1000 kg/m$^3$. A geometrically similar pump with an impeller diameter of 0.5 m operating at 960 rev/min is to be used with an oil density of 820 kg/m$^3$. Data for the pump operating in water at 1450 rpm is given as follows:

<table>
<thead>
<tr>
<th>Q / [m$^3$/s]</th>
<th>0.02</th>
<th>0.04</th>
<th>0.06</th>
<th>0.08</th>
<th>0.10</th>
<th>0.12</th>
</tr>
</thead>
<tbody>
<tr>
<td>ΔP$_1$ / [bar]</td>
<td>2.1</td>
<td>2.2</td>
<td>2.25</td>
<td>2.2</td>
<td>2.0</td>
<td>1.6</td>
</tr>
<tr>
<td>η / [1]</td>
<td>70</td>
<td>77</td>
<td>80</td>
<td>80</td>
<td>78</td>
<td>72</td>
</tr>
</tbody>
</table>

The oil is pumped from an open reservoir where the ambient pressure is 1 bar and the pressure loss at inlet to the pump is:

$$\Delta p = 0.5 Q^2 \text{ bar (Q in m$^3$/s)}$$

On exit from the pump oil flow through a plant where the flow rate and pressure loss are related by:

$$Q = 0.25 \sqrt{\Delta p} \text{ m$^3$/s (p in bar)}$$

The oil leaves the plant at ambient pressure, 1 bar. Calculate the flowrate of oil and the power required to drive the pump.

Solution strategy:
- Determine pump characteristic at new operating conditions
- Determine system characteristic
- Plot operating point

**Exercise 20:** Determine pump characteristic at new operating conditions using the flow and pressure rise coefficients for the pump.

Hint: To get conditions for new pump, we need to scale according to dimensionless parameters:

$$\frac{Q}{ND^3} \text{ is the same , thus } \frac{Q_2}{Q_1} = \frac{N_2 D_2^3}{N_1 D_1^3} \text{ and } \frac{\Delta P}{\rho N^2 D^2} \text{ is the same , thus } \frac{\Delta P_2}{\Delta P_1} = \frac{\rho_2 N_2^2 D_2^2}{\rho_1 N_1^2 D_1^2}$$
For the system we can sketch a diagram of what is going on:

So there is an inlet loss  \( \Delta p = 0.5 Q^2 \)

A plant loss given by  \( Q = 0.25 \sqrt{\Delta p} \) \( \Rightarrow \Delta P_{plant} = Q^2/0.25 = 16 Q^2 \)

Thus total  \( \Delta P_{system} = 16.5 Q^2 \), since the inlet and exit pressures are both 1 bar

**Exercise 21:** Plot the system and operating characteristic to determine the operating point.

**Answer:** Plotting the efficiency values and reading off for the operating \( Q \) gives approximately:

- \( Q = 0.44 \text{ m}^3/\text{s} \)
- \( \Delta P = 3.12 \text{ bar} \)
- \( \eta = 0.80 \)

*Hence the Power, \( \text{Power} = \frac{Q \Delta P}{\eta} = \frac{1}{0.80} \times 0.44 \times 3.12 = 171.6 \text{ kW} \)
4 Industrial Practise for Selecting a Pump

This information is taken from the “EUROPUMP Guide to the Selection of Rotodynamic Pumps” - May 2008. Although a number of other publications have similar sorts of guidelines in them and if you are thinking of spending money then a sales engineer can do lot of this for you. The following is an overview of the ideas.

**Key point:** the energy requirements of a pump over the lifetime will considerably exceed the capital cost of purchase. Even a single point of efficiency gain can pay for a new pump! So planning carefully is well worth while.

Selecting a Pump:

A) Choose the pump duty point. This is crucial – there is a tendency to overestimate the pump duty point to allow for “future demand” and for everyone to add a “safety margin” in pump selection so you end up throttling energy away. There are various ways around this:

- Impeller modifications to match the duty
- Mounting pumps in parallel to account of changes in flowrate

B) Pick a type of pump. A general guide is found in the figure below:

C) Pick a specific pump from the type selected above. Getting as near the best efficiency point (BEP) is vital. For best efficiency it is crucial to have the correct pump duty point as even for the “best” pump this varies substantially with flowrate. This requires detailed manufacturer charts (see PDF examples).

D) Benchmark the quoted efficiency. To check you have a reasonable number a procedure using specific speed is suggested by Europump, though it is a bit detailed to go through here.
Finally note that each pump is characterised by a series of simple curves showing the relationship between head, power, efficiency and flowrate. A real example is shown below. Note that with the theory covered in this course we can explain the shape of the head, NPSH and power curves directly.