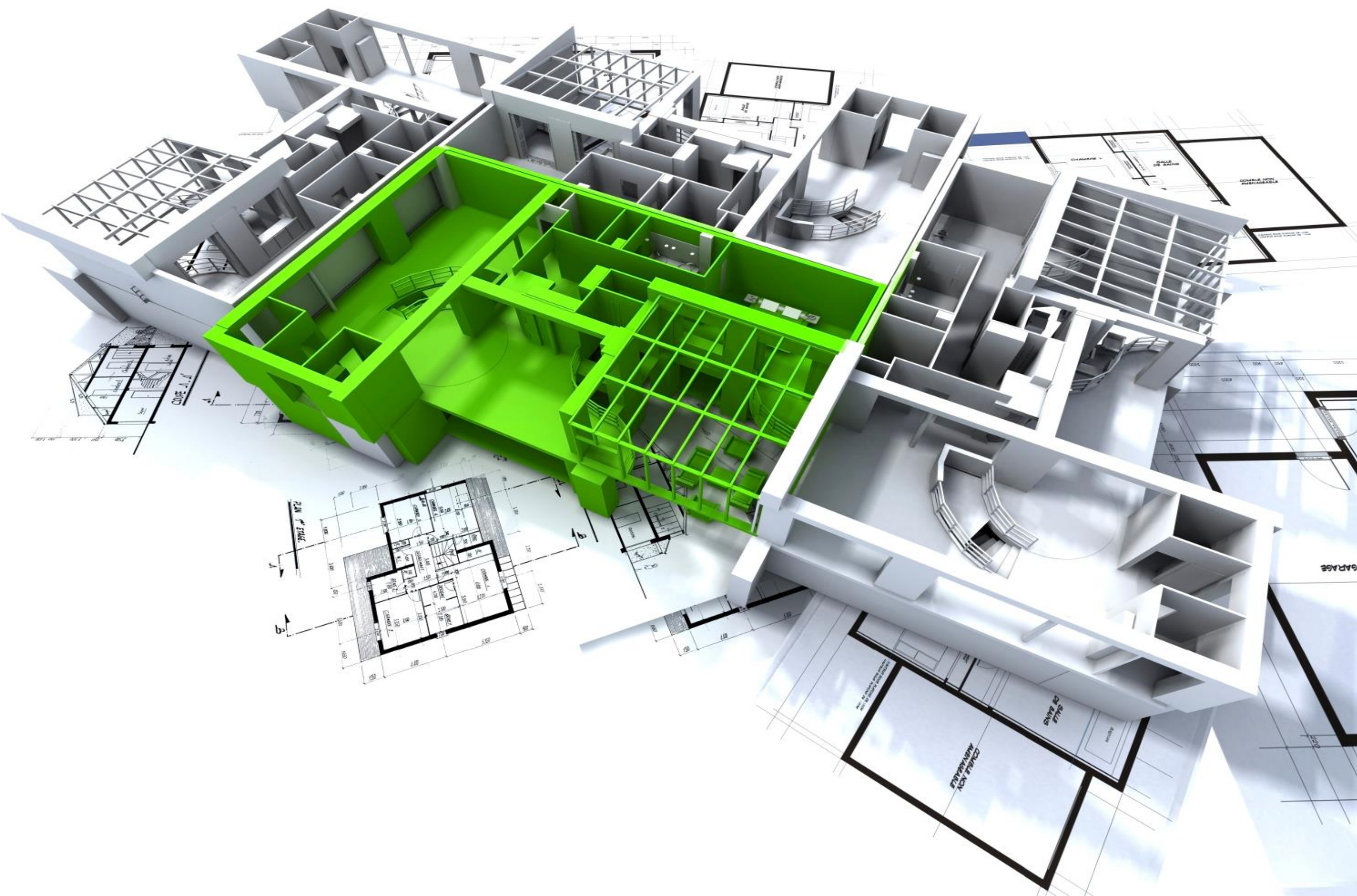


FLUIDS 3

PIPES, PUMPS AND POWER

In this section we will cover the basic flow patterns in pipes and how to calculate pressure drops in pipe systems. It will also cover the basic types of pump and how they are used and matched to pipes. Finally there is a section on power in fluid systems.

FLUIDS 3

PIPES, PUMPS AND POWER

OVERVIEW

In this unit you'll learn about:

- Flow through pipes in general and flow profiles
- The parameters which govern pressure drop in pipes
- How to calculate pressure drop in laminar and turbulent pipe systems
- The effect of fittings
- The different types of pump available and what they are used for
- Matching pipes to the right pump
- Pumps in series and parallel
- Practical issues with pumps
- The general equations of fluid power

ASSUMED KNOWLEDGE FOR THIS MODULE

It is assumed that you already have a knowledge of the following topics:

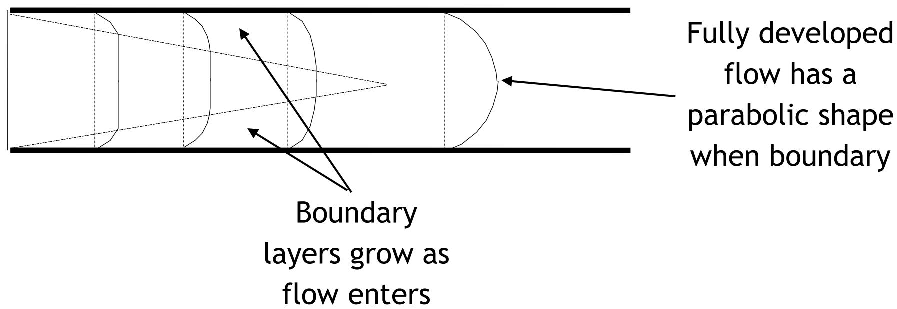
- *Basic fluid Mechanics – The Continuity Equation, Bernoulli's Equation and Forces in Fluids*
- *Fluid parameters – Density, Pressure and Viscosity.*
- *Fluid Statics – Hydrostaic pressure*

OBJECTIVE

To understand flow through pipes and pumps, be able to predict their behavior and design matched pipe and pump systems.

a) Flow in pipes

Bernoulli's equation assumes inviscid flow. In reality, all fluids have viscosity. The flow in a typical real pipe develops as shown below:



Because of this the velocity used in calculations tends to be the average (mean) value.

The flow may be laminar or turbulent, depending on the Reynold's number:

$$Reynold's\ number = \frac{\rho v l}{\mu}$$

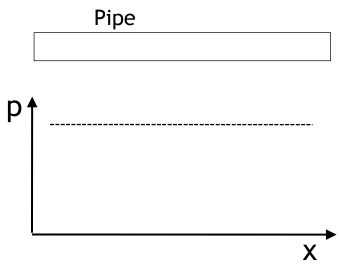
In the case of pipes, the critical length l is the pipe diameter.

TASK 1

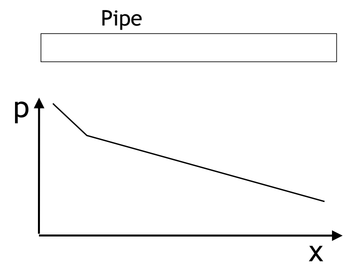
A pipe of 1cm diameter contains water (μ = 1.002 x 10^{-3} Pas). How fast must the water be travelling before the flow becomes turbulent? Express your answer in cm/s.

b) Pressure drop in pipes

Viscosity means that sections of fluid in the boundary layer slip past each other, due to their friction. This causes a loss of energy in the flow and a drop in pressure along the pipe (this effect is not seen in Bernoulli's equation, which assumes inviscid flow). Surface roughness in the pipe and fittings like valves, joints and bends exacerbate the situation. The prediction of a simple application of Bernoulli's equation and the real situation are shown overleaf.



Situation predicted by Bernoulli's



Situation in reality

Although a detailed treatment of this situation requires the use of the Navier - Stokes Equations, they are too complex for general use and engineers look for simpler, more practical, solutions. Fortunately, we can modify Bernoulli to take the pressure loss into account by adding in a new term, known as the *viscous head*. This is basically the pressure loss due to viscosity.

In general, if we have:

$$\rho g z + \rho \frac{v^2}{2} + p + \rho g h_{loss} = const$$

Potential energy (due to height) — Kinetic energy — Pressure — Loss due to friction (viscous head)

For circular pipes we can replace h_{loss} by a function of the important parameters (the equation has also been divided through by ρg):

$$const = z + \frac{p}{\rho g} + \frac{v^2}{2g} + \left[\int_0^x \frac{f v^2}{2Dg} dx \right]$$

Where x is the length along the pipe, D is the pipe diameter and f is a parameter called the *Friction factor*.

This may be integrated and the equation rearranged so that the pressure at any point in the pipe can be calculated:

$$p_B = p_A - \rho g \left[\Delta z + f \frac{L}{D} \frac{v^2}{2g} \right]$$

Where L is pipe length and Δz is the change in pipe height between A and B (negative if B is lower than A).

c) Friction Factor

All that remains is to define the Friction Factor f. This depends on the Reynold's number of the flow and how rough the pipe is. The roughness of the pipe wall is not so important if it is small in large pipes (and also not important in laminar flow); conversely, it's very important if a small bore pipe has big bumps on its walls. So we define the *Relative Roughness* as:

$$\frac{e}{D}$$

Where e is the average size of bumps in the pipe and D is the pipe diameter.

Now, for Laminar flow, f can be determined analytically as:

$$f = \frac{64}{R}$$

TASK 2

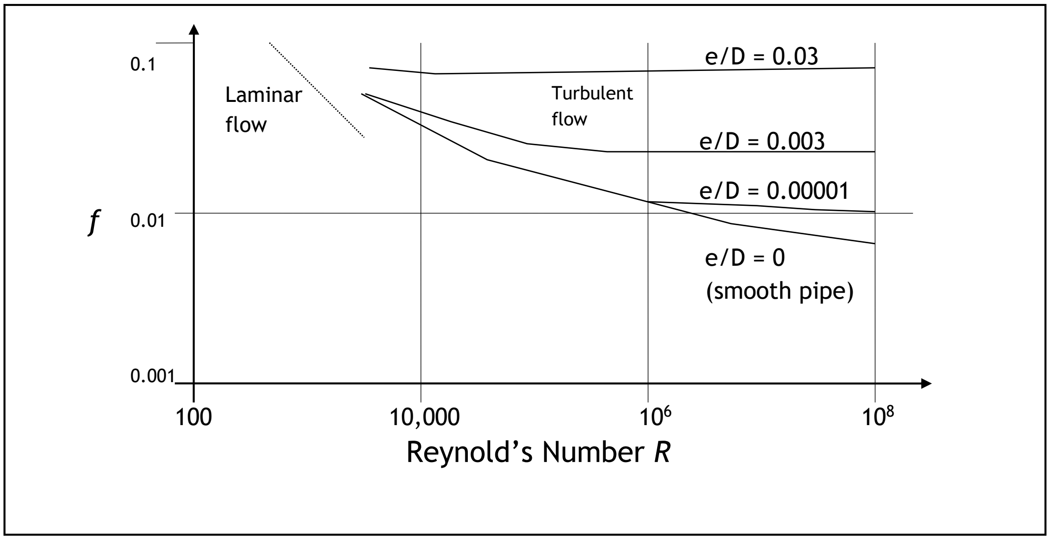
For the same pipe described in task 1, with a flow of water at 7cm/s calculate the pressure drop of a straight pipe 50m long which rises by 1m over its length. How much of this drop is due to the rise of height of the pipe and how much is due to the frictional loss. Express the frictional loss approximately as a head.

Where R is the Reynold's number.

For turbulent flow f can be determined empirically. One such determination is that by Colebrook:

$$\frac{1}{\sqrt{f}} = -2\log\left[\frac{e/D}{3.7} + \frac{2.51}{R\sqrt{f}}\right]$$

Solutions of this are plotted as the *Moody Chart*, shown overleaf. Ask your lecturer for an accurate copy of this chart (or find it on-line).



TASK 3

For the same pipe system used in task two, except this time at a velocity of 50cm/s, find the pressure drop. Assume the pipe is made from cast iron.

Look at the left of the Moody chart for similar values of laminar and turbulent flow - consider what happens as the flow gets faster and transitions - can you make any comments or observations about this?

d) Fittings

Having dealt with pressure loss in straight pipes, let's now consider the effect of fittings like valves and bends. The effect of these may be specified in one of three ways:

- As a stated pressure-drop per fitting.
- As an equivalent length per fitting
- As a *K* (Resistance Coefficient) value

The first method is not accurate unless there is a graph of flow rates, pipe diameters and losses. This is because the pressure drop across the fitting depends on all these parameters.

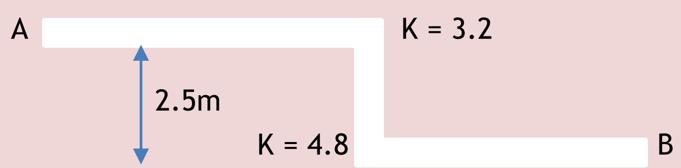
The second method is better, in that it gives an equivalent length of straight pipe (L_{eq}) per fitting. This length can be added to the straight length of pipe to give a total equivalent length.

The final method is probably the most widely used and defines a value *K* which is related to the pipe diameter *D*, the equivalent length L_{eq} and the friction factor *of the fitting* (not of the pipe as a whole) f_t by:

$$K = f_t \frac{L_{eq}}{D}$$

TASK 4

Look at the diagram below and calculate the total pressure drop between the ends (A and B) of the pipe.



The total length of pipe (not including joints) is 5.5m, it falls by 2.5m over its length as shown in the diagram. The pipe has a circular cross-section of diameter 2cm. The flow velocity is 5cms^{-1} and the pipe is carrying water (ρ = 1000 kg, $\mu = 1.002 \times 10^{-3}$ Ns/m^2). You may assume that the friction factor f_t of the joints is the same as for the rest of the pipe.

e) Pipe networks

Before leaving the subject of pipes, it should be noted that networks of pipes, which include pipes both in parallel (branching systems) with each other, and in series, are common in more advanced hydraulic systems (for example the national water supply). These can be modelled using electrical analogy (the pressures or heads being equivalent to voltages, flow rates to currents and resistance depending on friction factors, flow and length). This is generally termed *Deterministic Network Analysis* (read the Wikipedia page on “pipe network analysis” as an exercise at this point). One such classical system for calculating the resulting flow rates and pressure drops is the *Hardy-Cross method*. However, in more modern times, electronic simulators (like Multisim) are more commonly used, and pipe networks can be set up as circuits in these. In the course-book by Douglas (Fluid Mechanics), you can find equations using head as voltage, volumetric flow-rate as current and the term KQ^n as resistance (where the symbols have the conventional meaning and n is a constant which depends on flow type).

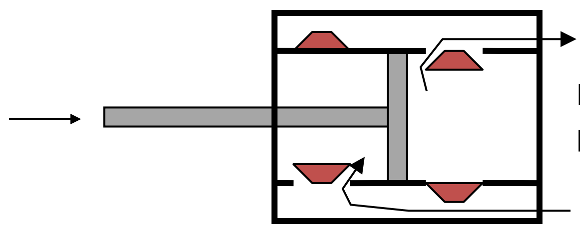
TOPIC 2 – PUMPS

a) Types of pump

Pumps come in a variety of types and sizes, but they can be broadly divided into two classes.

- Rotodynamic pumps
- Positive displacement pumps

A typical displacement pump is the piston or plunger type.

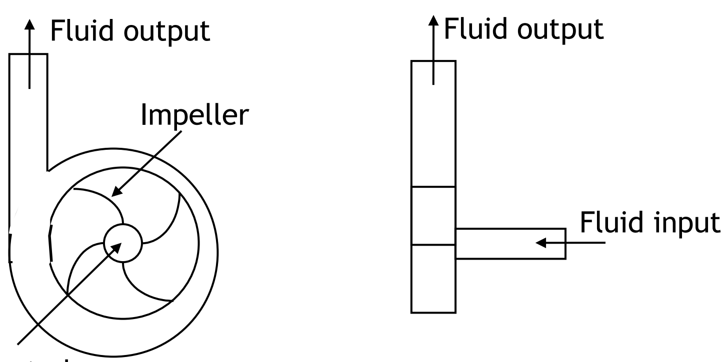


Double acting piston pump

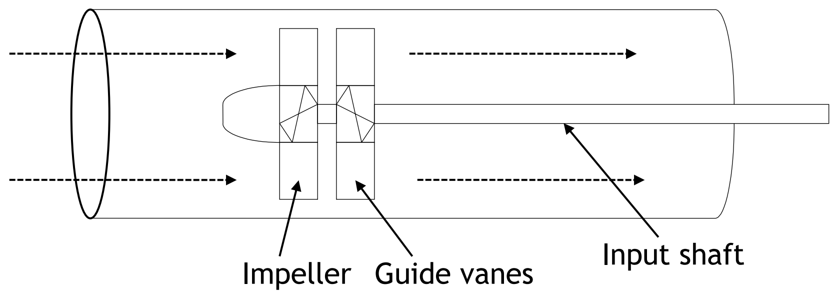
Other similar types include diaphragm pumps and gear pumps (look up a list of displacement pumps on the internet as an exercise).

Such pumps tend to be rather bulky, expensive and need a safety relief valve in case of blockage. They are also low speed and deliver their load in pulses (and vibrate). They are best suited to viscous fluids.

A typical rotodynamic pump is the Centrifugal type shown below:



Another common form is the Axial pump

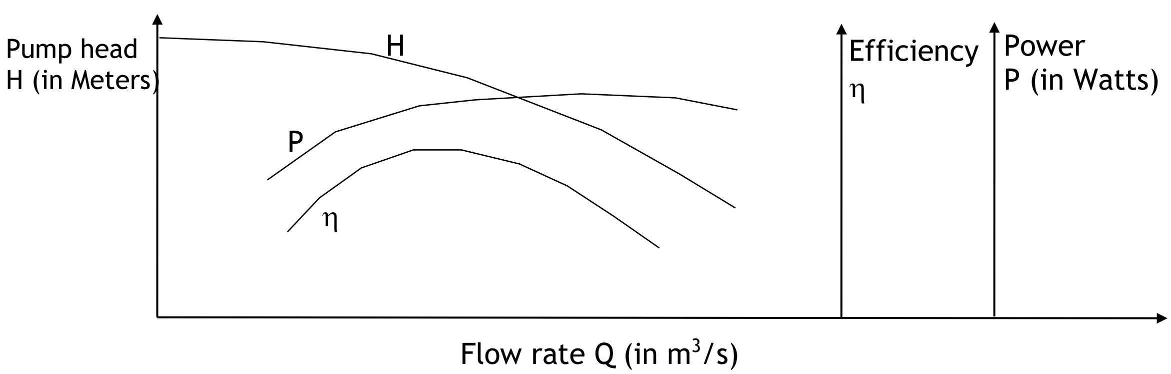


The guide vanes ensure that the fluid leaves the pump with no swirl.

Rotodynamic pumps have a high fluid velocity. They also give a steady flow output, are mechanically simple and are particularly effective in systems of low viscosity and high fluid speed. However, they are not self-priming. The centrifugal type works well at lower flow rates, but can build up a high pressure; the axial type is good a high flow rates, but lower added pressure. As an exercise look up a list of rotodynamic pumps on the internet.

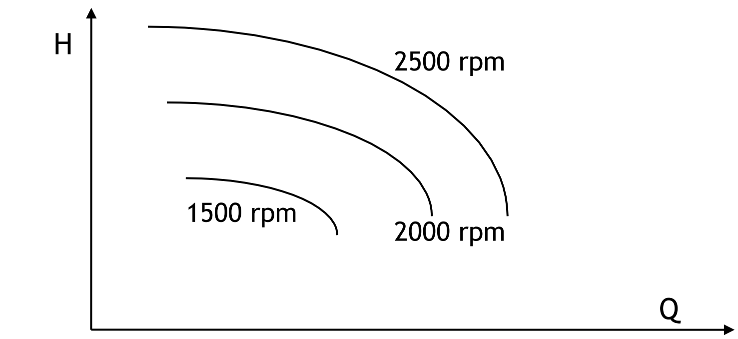
b) Pump specifications

Pumps are often specified using a graph like that shown below (this is typical for a centrifugal pump):

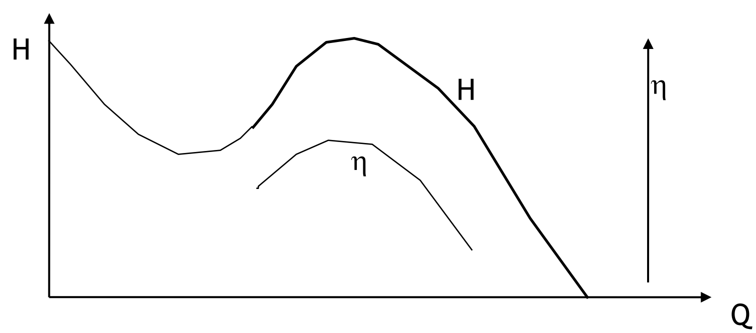


We can see from this that (as might be expected) the pressure is at a maximum when the flow rate is zero. The power is either the external supplied electrical power or power into machine shaft and the efficiency is *Power increase of fluid / Power of machine shaft*.

The Specifications are often given for a number of pump speeds:



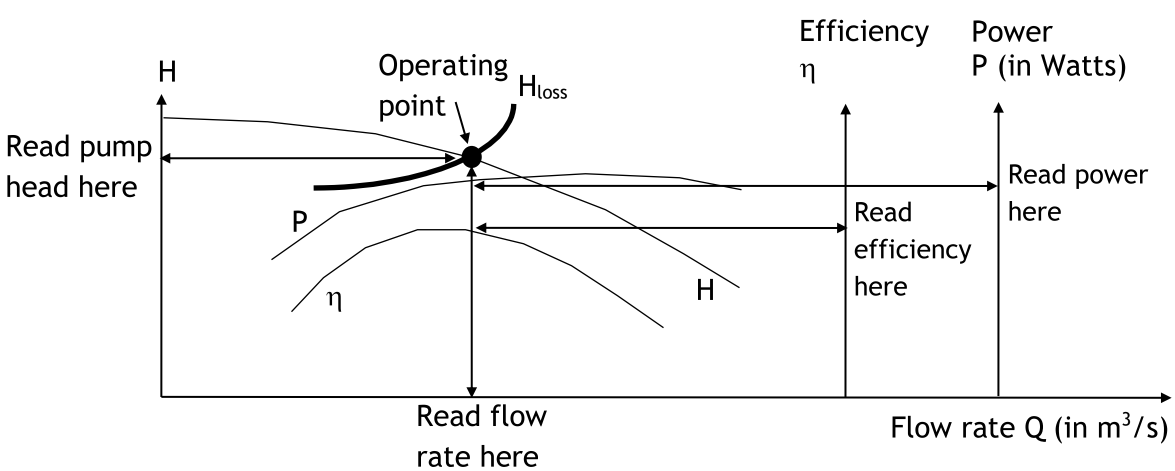
Axial pumps have a slightly different looking curve. Their efficiency often decreases steeply and this must be considered in the design. As mentioned earlier, such pumps can handle very high flow rate, but at low heads.



c) Matching the pump to the pipe system

As engineers, one of the main things we want to know is how to match a pump to an external system (typically a system of pipes).

We can plot the loss-head (from the equation on page 3) on the pump specification and the operating point of the system can be read off.



From basic fluids we know the flow rate is related to the average fluid velocity by:

$$Q = Av$$

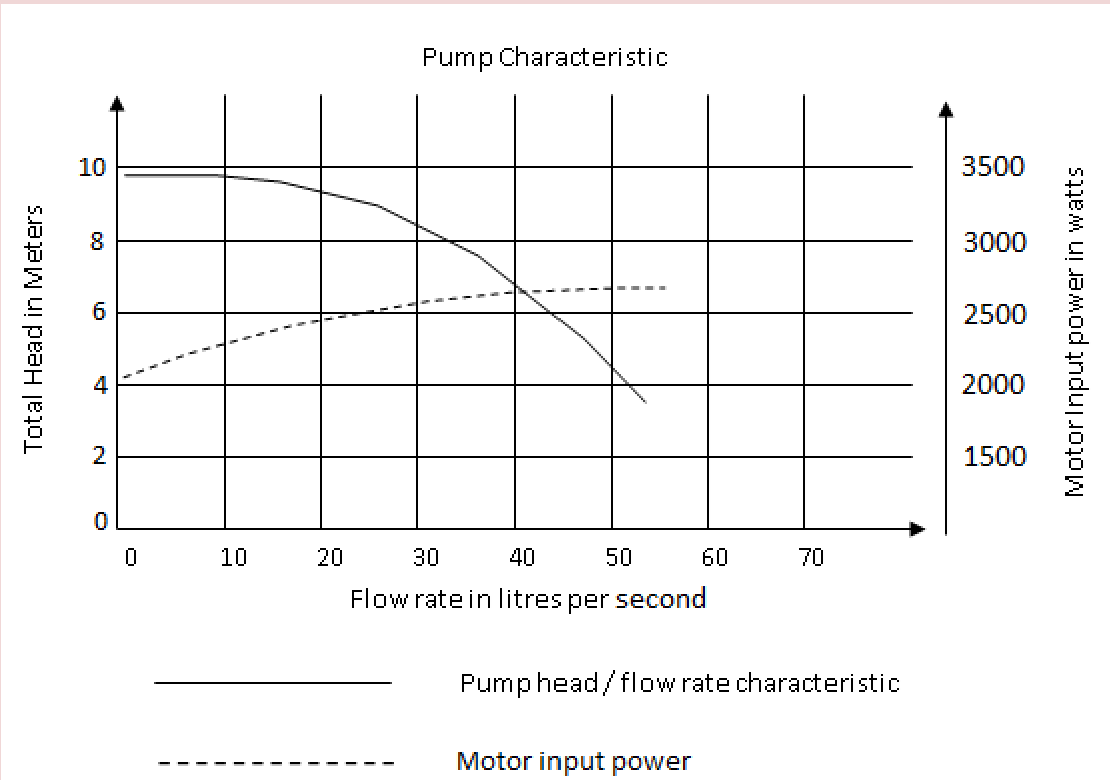
so, for a circular pipe

$$Q = \frac{\pi D^2}{4} v$$

Where D is the pipe diameter.

TASK 5

Consider a pump with the characteristic shown below:



Assume that this pump is connected to a pipe system in which the measured gauge-pressure drops were 98KPa at a volumetric flow rate of $0.03 m^3s^{-1}$. 88KPa at $0.025m^3s^{-1}$ and 78KPa at $0.02m^3s^{-1}$. (continued overleaf)

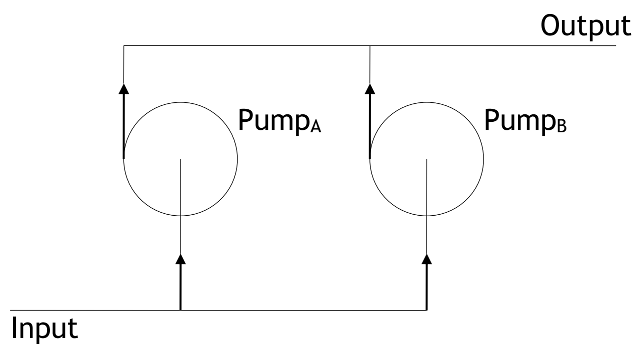
TASK 5 (continued)

Determine:

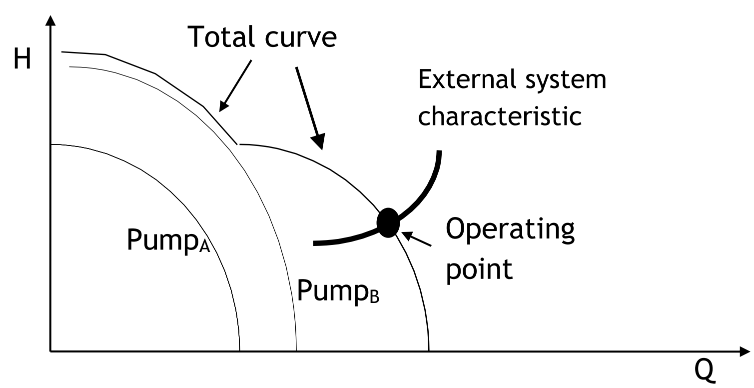
i) The operating point of the system (pressure and flowrate)
ii) The Input power to the pump

d) Pumps in series and in parallel

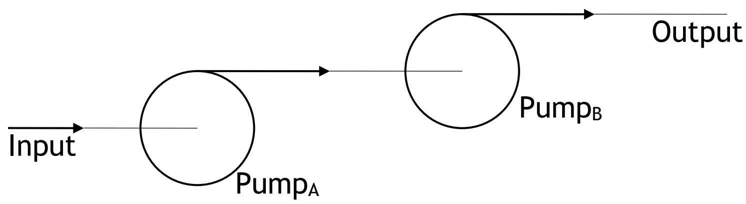
When two identical pumps are connected in parallel, the total head is the same as for one pump, but the total flow rate = $Q_A + Q_B$.



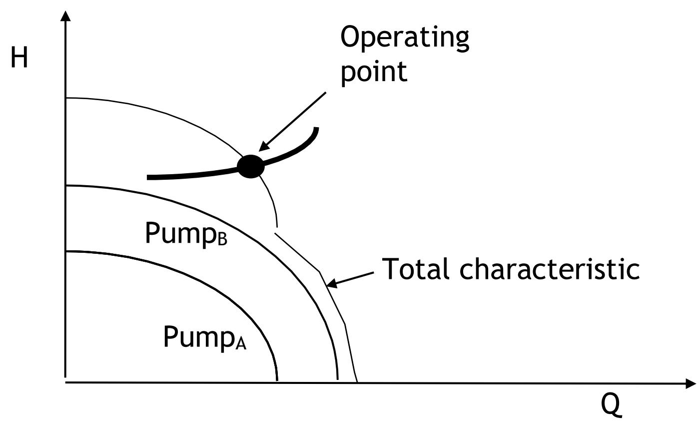
The total pump characteristic is determined as shown below.



When two identical pumps are connected in series, the total flow rate is the same as for one pump, but the total head $= H_A + H_B$.



The characteristic is shown below.



e) Problems with pumps

There are several problems which can become apparent with pumps. Some pumps (in particular displacement pumps) cannot endure pipe blockages - they generally have pressure release valves to counter this situation. Likewise, if a pump runs dry (in particular a rotodynamic pump) it may over-speed and mechanical damage may result, rotodynamic pumps are also generally not self-priming. The pump specifications must be carefully consulted to ensure that pumps are not over or under driven. Attention must also be paid to ensure that the pump is suitable for the fluid parameters.

There are two particular problems which merit special mention. The first of these is Cavitation.

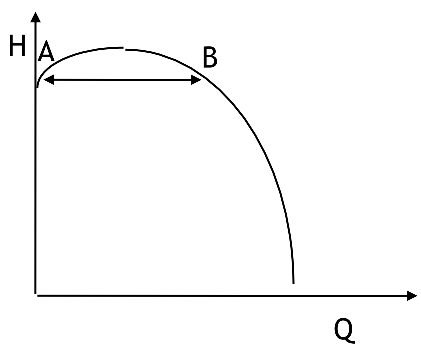
Cavitation is the local boiling of the fluid. It can cause damage to the pump through vibration and erosion of its structure. It also lowers the efficiency of the pump.

Pump manufacturers get over this problem by specifying the minimum allowed pressure at the pump inlet. This is known as *Net Positive Suction Head* or NPSH. Care must be taken to ensure that the pressure at the inlet is greater than this.

$$NPSH = \frac{p_o - p_v}{\rho g}$$

Where p_o is the stagnation pressure at the inlet ($p + \rho v^2/2$) and p_v is the vapour pressure.

Another potential problem occurs when two identical pumps are used in parallel. If the pump's maximum head does not occur at zero flow, the pumps can *shuttle* between points A and B as shown below. This results in the pumps rapidly switching load and potentially causing damage.



TOPIC 3 – GENERAL POWER IN FLUIDS

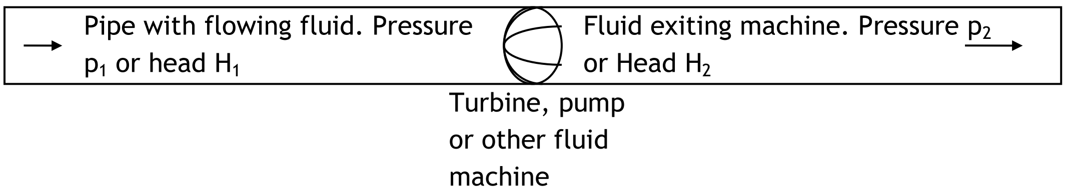
The power lost or gained in a fluid passing through a system or machine (which might add power to it like a pump, or remove power like a turbine) is:

$$P = Q\rho g\Delta H = Q\Delta p$$

Where P is the power gained or lost by the fluid; Q is the volumetric flow rate (m^3/s) though the system; ρ is the fluid density (Kg/m^3); g is gravitational acceleration (9.81 m/s); ΔH is "head" (pressure measured as a height, m) and Δp is pressure difference (Pa) between one side of the system and the other.

This is a very useful equation because the quantities are easy to measure and it applies to all machines. Of course the actual amount of power delivered to the output of the machine also depends on its efficiency as well (there will be losses in the bearings and other mechanical parts).

If a machine like a turbine is placed in the path of the fluid as shown below, then the power consumed by it can be calculated if the pressure drop across it is known (and, as stated above, this also applies to a compressor or pump supplying power *to* the fluid).



In such cases the pressure (or head) rise or drop is:

$$\Delta p = p_1 - p_2$$
$$\Delta H = H_1 - H_2$$

TASK 6

a) How much power is being absorbed by a machine powered by water if it has a pressure difference of 10000 Pa across it and the water enters through a circular pipe of diameter 1 cm at a velocity of 5 m/s?

b) How much power is required to pump water at 1 m^3/s up a slope of vertical height 1m assuming the pump is only 50% efficient.

TOPIC 4 – EULER'S TURBINE/PUMP EQUATION

If you need to know what is going on inside the machine in greater detail, this can be obtained using Euler's turbine equation. The idea behind this is that angular momentum is conserved be fluid travelling through the system and so any change in a rotating machine must be due to torque added or subtracted from the machine shaft.

We can develop the basic equation very simply by noting that torque is force times distance:

$$\Gamma = f \times d$$

But the force of a jet of fluid is its mass flow-rate times its velocity (see second-year notes on fluid momentum):

$$\Gamma = \dot{m}v \times d$$

In a pump or turbine with a circular configuration, the distance d at which the fluid enters or leaves the machine is the radius from the center r. So if the fluid enters at the input i and leaves at the output o, then:

$$\Gamma = \dot{m}_o v_o r_o - \dot{m}_i v_i r_i$$

Since the mass-flow rate through the system is a constant then:

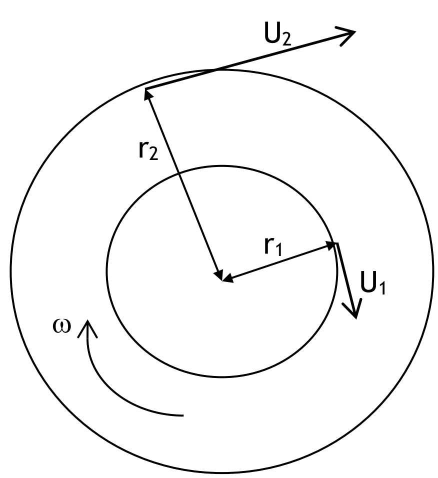
$$\Gamma = \dot{m}(v_o r_o - v_i r_i)$$

Where the velocities v used in the equations are those tangential to the machine rotors. Euler's equation therefore works out torque and power from the magnitude and direction of the input and output fluid velocities.

TASK 7

A rotating machine has water enter through a circular duct of diameter 2cm at 10cm/s this water enters the blades of a machine tangentially at a radius of 10cm from the axis of the machine. The water leaves the end of the blades with a tangential velocity of 1m/s at a radius of 40cm from the axis. Calculated the torque being delivered by the shaft. It the machine is consuming 2W of power calculate the rpm of the shaft.

The equation above can be used as the basis for deriving more complex expressions for real machines by combining it with expressions for the rotor speeds and other parameters. As an example of this, let us consider a centrifugal compressor where fluid enters in the centre (at right angles to the blades) and exits at the rim. This treatment is a simplified one - textbooks will contain a full derivation (and with modification it can be applied to any rotating machine). This situation is shown in the diagram below.



U_1 is the tangential velocity of the (end of the) machine blade at the input of the turbine or pump (= $r_1\omega$) and U_2 is the tangential velocity of the (tip of the) machine blade at the output of the turbine or pump (= $r_2\omega$).

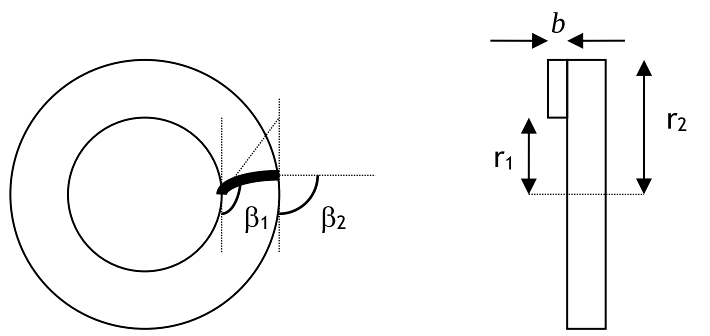
If the fluid velocity in the direction of U_1 at the input of the machine is F_1 and the fluid velocity, at the output, in the direction of U_2 is F_2, then shaft torque is:

$$T = \rho Q(r_2 F_2 - r_1 F_1) = \dot{m}(r_2 F_2 - r_1 F_1)$$

and power is:

$$P = T\omega = \rho Q(U_2 F_2 - U_1 F_1)$$

The relative fluid velocities and other useful information can be found from simple geometry and vector diagrams. For example, take a centrifugal pump with the geometry shown in the diagram below. β_1 is angle of machine blade at the input side of the pump (with respect to a tangent to the input pipe) and β_2 is angle of machine blade at the output of the pump. The width of the blade is b.



Front view Side view

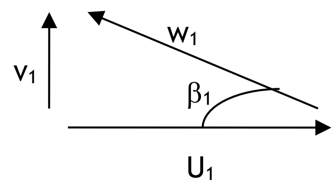
At the input F_1 = 0, this is because the fluid is arriving through a pipe at right angles to the blades (from a pipe flowing into the page in the front-view diagram above). The volumetric flow-rate is:

$$Q = 2\pi r_1 b v_1$$

Where v_1 is the component of flow directly into the machine (normal to U_1):

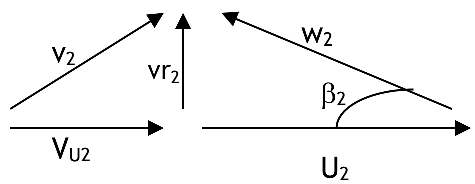
$$v_1 = U_1 \tan \beta_1$$

As can be seen from this vector diagram representing the various velocities at the input:



The velocity w_1 represents the component of velocity along the surface of the blade.

A similar (though slightly more complex) diagram can be drawn for the output:



$$U_2 = r_2\omega$$
$$Q_2 = Q_1 = Q = 2\pi r_2 b v_{r_2}$$
$$v_{r_2} = \frac{Q}{2\pi r_2 b}$$

and finally

$$F_2 = U_2 - v_{r_2}\cot\beta_2$$

SUMMARY

- Flow through even a frictionless pipe is not perfect – it has a flow profile and may be laminar or turbulent.
- Pressure drops between the ends of a real pipe – this is not predicted by Bernoulli's equation.
- The pressure drop can be predicted using a simple equation and a parameter called friction factor.
- The friction-factor of the pipe depends on whether the flow is laminar or turbulent.
- Turbulent flow friction-factor may be obtained using the Moody Chart.
- Additional losses due to fittings must be added to the straight pipe loss.
- Networks of pipes can be simulated using equivalent electrical circuits.
- Most pumps can be classified broadly into displacement or rotodynamic
- Displacement pumps are useful for viscous fluids, but have several disadvantages.
- Rotodynamic pumps give steady, high-volume flow.
- Two common rotodynamic pumps are centrifugal and axial types.
- Pumps come with characteristic specification curves.
- Pipe performance can be plotted on these curves to obtain the operating point of the system.
- Pumps can be operated in series for higher pressure gains and parallel for high flow rates.

- There are several non-ideal aspects to pumps which must be considered in practical situations.
- Power gained or lost by a fluid can be calculated using a simple equation which applied to any machine or system.
- An in-depth understanding of fluid machine operation can be obtained using Euler's fluid machinery equations.