Chapter – 8

Introduction to Hydraulic Machines

8.1 Introduction

The function of a hydraulic machine is to effect an exchange of energy between a mechanical and a fluid system. In civil engineering the only classes of hydraulic machine with which we are directly concerned are ***pumps*** & ***turbines***.

Pumps are a means of adding energy to water. They convert mechanical energy (imparted by rotation) in to water (hydraulic) energy used in lifting water to higher elevations. The mechanical energy is provided by an electric motor.

Turbines are a means of taking energy out of water. They convert water (hydraulic) energy in to mechanical energy (shaft power). The shaft power developed is used in running an electric generator directly coupled to the shaft of the turbine, thus producing electrical power.

8.2 Pump Types

There are two main categories of pumps:

- Positive displacement pumps
- Roto-dynamic pumps

8.2.1 Positive displacement pumps

Positive displacement pumps usually deliver only small discharges irrespective of the head pumped against. Typical examples of this type of pumps include:

- Reciprocating pump
- Rotary pump

8.2.1.1 Reciprocating pump

This type of pump is often used for domestic water supplies in developing countries for lifting ground water. In its usual form it consists of a ram, piston, and valve arrangement.

The piston moves up & down in a cylinder *(see figure 6.1)*. When the lever is pushed downwards the piston rises, lifting water above it through the outlet. At the same time it sucks water up the well through the non-return valve & fills the cylinder. When the lever is raised the non-return valve close & the piston

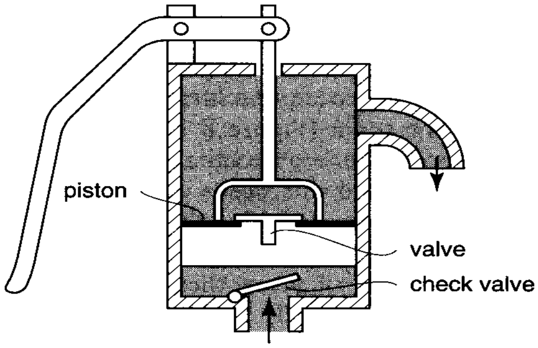


Fig. 8.1 Reciprocating pump

descends allowing water to flow through another valve in to the upper part of the cylinder. The process is then repeated.

8.2.1.2 Rotary Pump

Rotary pump contains two gears or rotors, which mesh together as they rotate in opposite directions *(see fig 6.2)*. Pressure is generated by the intermeshing gears, which operate with minimum clearance. Water becomes trapped between the gears and forced in to the delivery pipe.

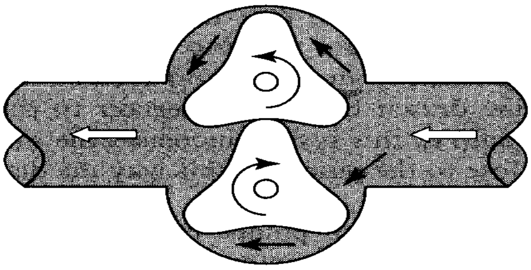


Fig. 8.2 Rotary Pump

This form of pump is eminently suited to handling small discharges (<30 l/s) and viscous liquids.

8.2.2 Roto-dynamic Pumps

Roto-dynamic pumps rely on rotational movement for their pumping action. A rotating element, known as *impeller*, imparts velocity to a liquid and generates pressure. An outer fixed casing, shaft, & diving motor complete the pump unit.

Roto-dynamic pumps are the most widely used types of pumps in civil Engineering. Its field of employment ranges from public water supply, drainage, & irrigation to the very special requirements of suction dredging & the transport of concrete or sludge.

There are three main categories of roto-dynamic pumps based on the way water flows through them:

- Centrifugal (radial flow) pumps
- Axial flow pumps
- Mixed them pumps

8.2.2.1 Centrifugal pumps

Centrifugal pumps are the most widely used of all the roto-dynamic pumps. They are named because of the fact that the pressure head created is largely attributable to centrifugal action. They may be designed to handle up to a head of 120m.

Water is drawn in to the pump from a source of supply through a short length of pipe called the *suction (see fig. 6.3).* Water enters at the center or eye of the impeller, is picked up by the vanes, and forced outwards in a radial direction. The water is collected by the pump casing & guided towards the outlet called the *delivery.*

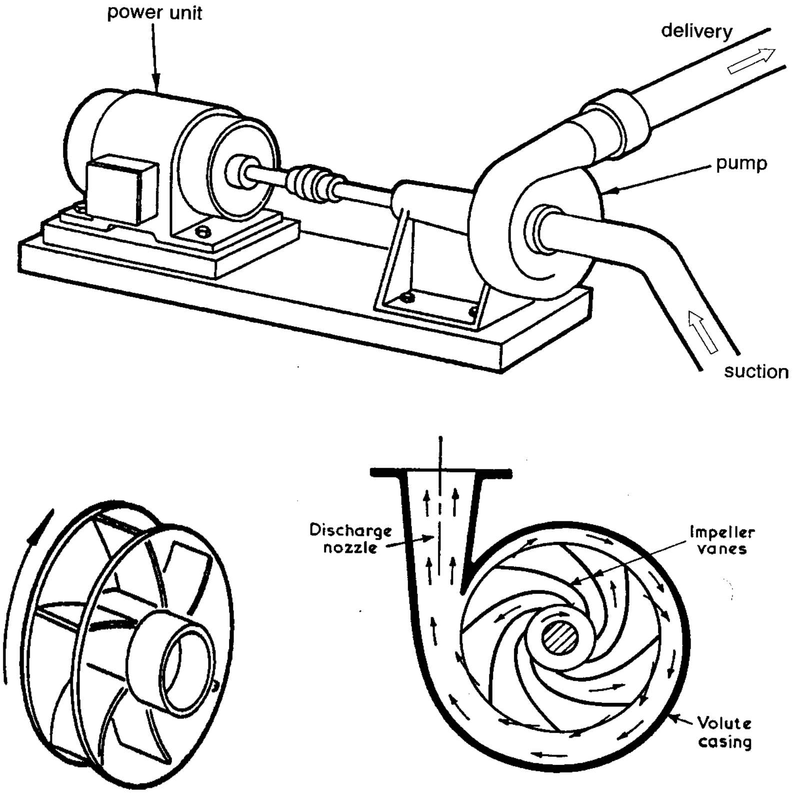


Fig. 8.3 Centrifugal pump

In order that energy shall, not be wasted and efficiency there by lowered, it is essential to convert as much as possible of the considerable velocity head at exit from the impeller in to useful pressure head. Normally, this is achieved by shaping the outer casing in spiral form so that the sectional area of flow around the periphery of the impeller is gradually expanded.

8.2.2.2 Axial flow pumps

This type of pump is well suited to situations where a large discharge is required to be delivered against a low head. The maximum operating head is between 9 and 12m.

Axial flow pumps consist of a propeller housed inside a tube that acts as a discharge pipe *(see fig 6.4).* The power unit turns the propeller by means of a long shaft running down the middle of the pipe & this lifts the water up the pipe.

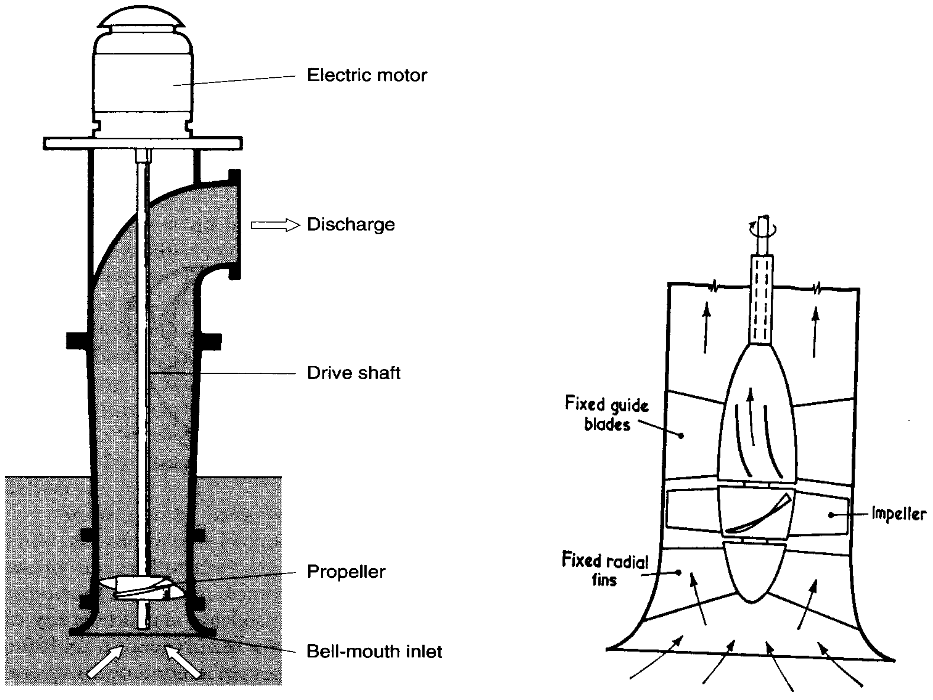


Fig. 8.4 Axial flow pump

Water enters axially and the impeller imparts a rotational component, the actual path followed by a particle being that of a helix on a cylinder. Head is developed by the propelling action of the vanes, centrifugal effects playing no part.

8.2.2.3 Mixed flow pumps

Mixed flow pumps occupy an intermediate position between the centrifugal & axial flow types and so combines the best features of both pump types. Flow is part radial & part axial, the impeller being shaped accordingly. The path traced by a fluid particle is that of a helix on a cone. The head range is up to about 25m.

Mixed flow pumps are efficient at pumping larger quantities of water than centrifugal pumps and are more efficient at pumping to higher pressures than axial flow pumps.

8.2 Turbine types

The possible combination of head and discharge at hydroelectric sites is extremely varied and is reflected in a corresponding diversity of turbine design. There are two main categories of turbine:

- Impulse turbines
- Reaction turbines

8.2.1 Impulse turbines

An impulse turbine is one in which the pressure energy of the water is converted to velocity energy before it impinges on a rotational element over a limited portion only of the periphery, their being no subsequent change in pressure. Impulse machines today are of the *Pelton wheel turbines*, also called tangential flow turbines, and are suitable for high heads in excess of 300 m.

A typical Pelton turbine arrangement is shown in *fig 6.5*. The nozzle discharges in to the atmosphere a high velocity jet which impinges on a series of buckets mounted on the periphery of a wheel, also called *runner*. The torque exerted by the impact and deviation of the jet causes the wheel to rotate. Its energy usefully expended, water leaves the buckets at a relatively low velocity and is directed towards the discharge channel.

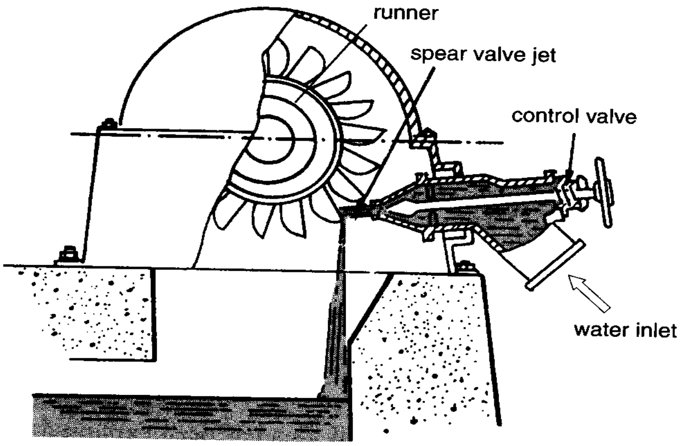


Fig8.5 Pelton turbine

The turbine must be set a sufficient height above the maximum tail water level if free discharge is to be ensured.

8.2.2 Reaction turbines

In a reaction turbine, the initial pressure-velocity conversion is only partial, so that water enters the rotating element throughout the entire periphery and all the flow passages run full. Modern reaction turbines are of two types: *Francis & Propeller (Kaplan)*, catering for medium and low heads respectively.

8.2.2.1 Francis turbines

Francis turbines are like a centrifugal pump in reverse *(see fig 6.6)*. The runner was shaped like a centrifugal impeller, flow being predominantly radial with the radii at entry and exit the same for all flow paths.

Water is directed in to the runner by means of a spiral casing and a number of aerofoil-shaped blades, called *guide blades*, spaced evenly around the periphery. These guide blades are adjustable, the amount of opening being controlled by the turbine governor. The role of the guide blades is to guide the flow in to the runner with the minimum amount of turbulence, as well as to regulate the discharge and hence power output.

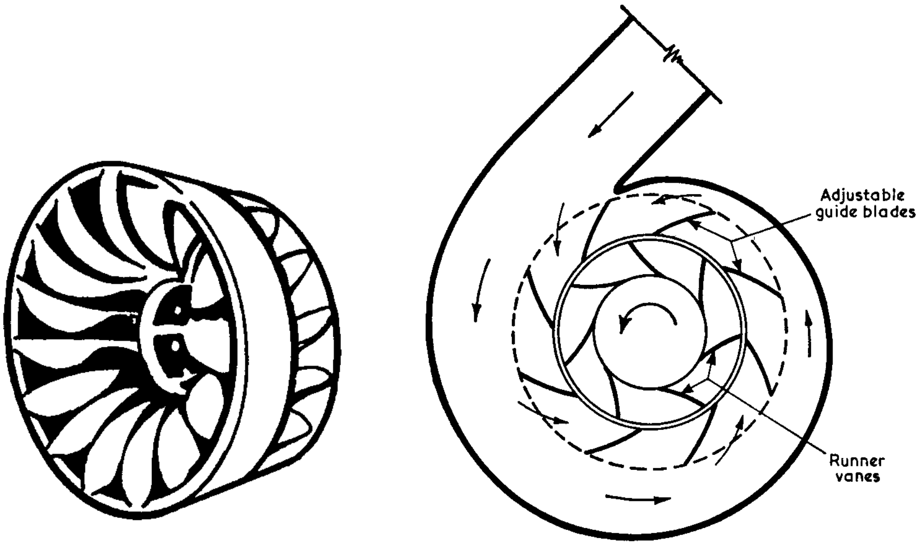


Fig. 8.6 Francis turbine

The head range for Francis turbine is from 30 m to about 450 m. As this is the most common head available, this type of turbine enjoys a great numerical superiority over other types.

The velocity head at discharge from the runner may amount to 20 %, or more, of the available head and as with centrifugal pumps it is clearly important to convert as much as possible of this otherwise wasted energy to useful pressure head. This can be accomplished by means of an expanding passage, called a *draft tube*, which finally discharges the water at a relatively low velocity to the tail water.

8.2.2.2 Kaplan turbines

Kaplan turbines are like axial flow pumps in reverse *(see fig 6.7)*. They operate at low heads, usually less than 60 m, and high discharges.

They have blades on their runners that can be twisted to different angles in order to work at high efficiency over a wide range of operating conditions.

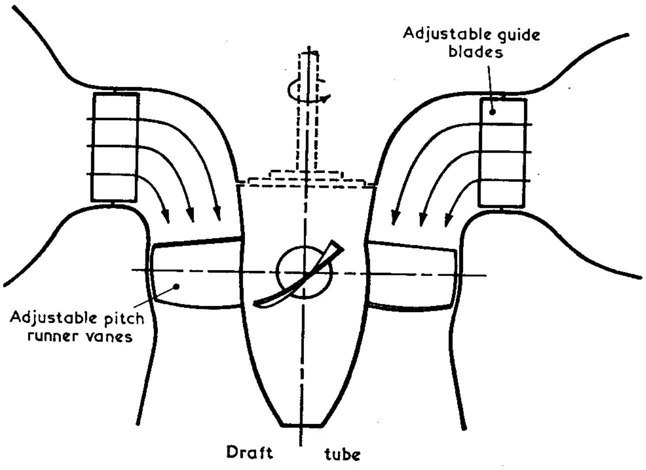


Fig. 7.7 Kaplan turbine

8.3 Head on pumps and turbines

8.3.1 Head on pumps

The total head on a pump is the excess of the outlet head over the inlet head. Each of these heads may be regarded as being composed of elevation head, pressure head, and velocity head.

Referring to *fig 6.8*, the total head on a pump may be expressed by:

$$H = H_s + H_d + H_{Ls} + H_{Ld} \qquad --------- \qquad (1)$$

Where, H_s & H_d are the static suction and delivery lifts respectively, and H_{Ls} & H_{Ld} are the energy head losses (friction + minor) in suction and delivery branches, respectively. If the pump is situated below the level of the water surface in the suction well, H_s is negative.

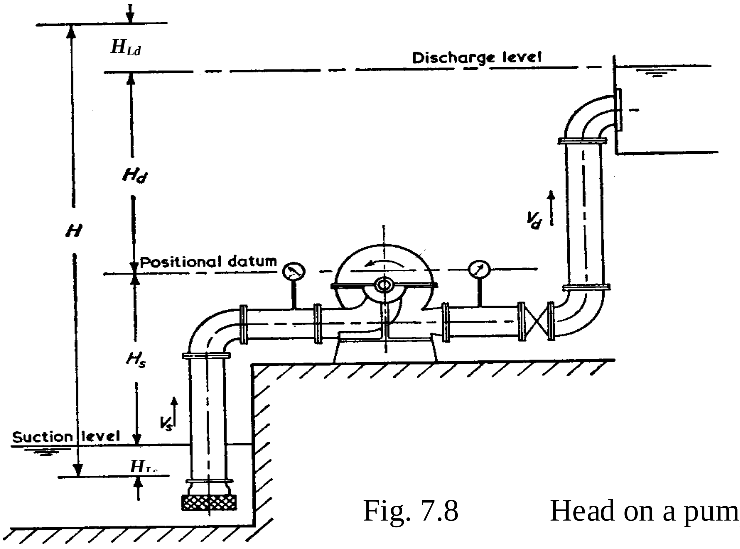


Fig. 7.8 Head on a pump

8.3.2 Head on turbines

The net head on a turbine is the head available for doing work, that is to say, the difference between the total head (elevation + pressure + velocity head) at inlet and outlet.

Referring to *fig 6.9*, the net head on a reaction turbine situated at some distance from the intake is given by:

$$H = H_G - H_L \qquad \text{-----------------} \qquad (.2)$$

Where, H_G is the gross head (intake surface level to tail water level) and H_L is the energy head loss in the supply pipeline.

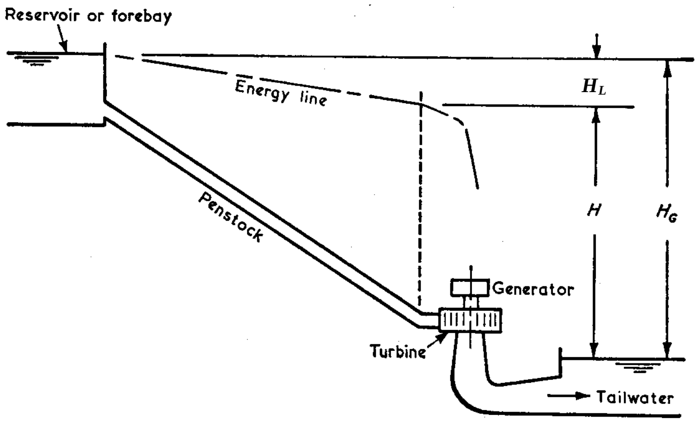


Fig. 8.9 Head on a reaction turbine

The same expression is applicable to impulse turbines. However, as these machines operate under atmospheric pressure, H_G is measured to an appropriate jet level.

8.4 Specific Speed

It is useful to have a common basis on which different types of pump or turbine design can be compared in respective of size. The parameter known as *specific speed* has been introduced for this purpose, and the respective definitions could be as follows:

☞ ***The specific speed of a pump*** is the speed in rev/min of a geometrically similar pump of such a size that it delivers 1 m³/s against 1 m head. It is expressed by:

$$n_s = \frac{nQ^{1/2}}{H^{3/4}} \qquad \text{------------------} \qquad (3)$$

Where, n_s is specific speed (rev/min), n is speed of rotation (rev/min), and Q & H are discharge (m³/s) and head (m) respectively.

☞ ***The specific speed of a turbine*** is the speed in rev/min of a geometrically similar turbine of such a size that it produces 1 kW under 1 m head. It is expressed by:

$$n_s = \frac{nP^{1/2}}{H^{5/4}} \qquad \text{------------------} \qquad (4)$$

Where, P is the power output in kW.

The above definitions of the specific speed have recognized the significant performance parameters. In the case of pumps it is the discharge that is important, while for turbines it is the power output.

The values of n, Q, H, & P in the expressions for the specific speed are those for normal operating condition (the design point), which would generally coincide with the optimum efficiency.

It can be noted that the specific speed is independent of the dimensions and therefore relates to shape rather than size. Thus, all pumps or turbines of the same shape have the same specific speed.

The valve of specific speed is mainly used for selection of a suitable type of pump or turbine for a particular site. The following table gives guidelines on this purpose.

Table: Specific speeds for different types of pumps and turbines.

Machine type		***n_s (rpm)***	***Comments***
Pumps	Centrifugal Mixed flow Axial flow	10 – 80 70 – 180 150 – 320	High head – small discharge Medium head - medium discharge Low head – large discharge
Turbines	Pelton Francis Kaplan	10 – 40 35 – 400 300 – 1000	High head – small discharge Medium head - medium discharge Low head – large discharge

8.5 Performance

8.5.1 Losses & efficiencies

The overall efficiency η of a pump or turbine is the ratio of the useful power output to the power input or available. Thus,

- For pumps;

$$\eta = \frac{\gamma QH}{P_i} \qquad \text{-----------------} \qquad (6.5)$$

- For turbines;

$$\eta = \frac{P}{\gamma QH} \qquad \text{-----------------} \qquad (6.6)$$

Where, P_i is the power input to a pump and P the corresponding output from a turbine.

Pump efficiencies are usually of the order of 80 %, whereas turbine efficiencies are rarely less than 90 %, the difference being largely accounted for by the generally greater size of turbines and the more efficient flow passages.

The energy losses that occur within a pump or turbine are attributable to volumetric, mechanical, and hydraulic losses.

☞ ***The volumetric loss*** arises from the slight leakage Q_L (from the high pressure side to the low pressure side) in the small clearances that must be provided between the rotating element and the casing. Thus, the impeller passages of a pump are handling more water than is actually delivered, while the runner passages of a turbine are handling less than is available. The volumetric efficiency η_v is given by:

- For pumps;

$$\eta_v = \frac{Q}{Q+Q_L} \qquad \text{------------------} \qquad (6.7)$$

- For reaction turbines;

$$\eta_v = \frac{Q-Q_L}{Q} \qquad \text{------------------} \qquad (6.8)$$

☞ ***The mechanical loss*** is a result of power loss due to mechanical friction at bearings and fluid shear in the clearances. Thus, the mechanical efficiency η_m is given by:

- For pumps;

$$\eta_m = \frac{\gamma(Q+Q_L)H_o}{P_i} \qquad \text{---------------} \qquad (6.9)$$

- For turbines;

$$\eta_m = \frac{P}{\gamma(Q-Q_L)H_o} \qquad \text{---------------} \qquad (6.10)$$

☞ ***The hydraulic loss*** arises from head loss in the flow passages due to friction and eddies. Thus, the hydraulic efficiency may be given by:

- For pumps;

$$\eta_h = \frac{H}{H+loss} = \frac{H}{H_o} \qquad \text{---------------} \qquad (6.11)$$

- For turbines;

$$\eta_h = \frac{H-loss}{H} = \frac{H_o}{H} \qquad \text{---------------} \qquad (6.12)$$

The overall efficiency expressed by equations (6.5 & 6.6) is the result of the product of the volumetric, mechanical, & hydraulic efficiencies. That is,

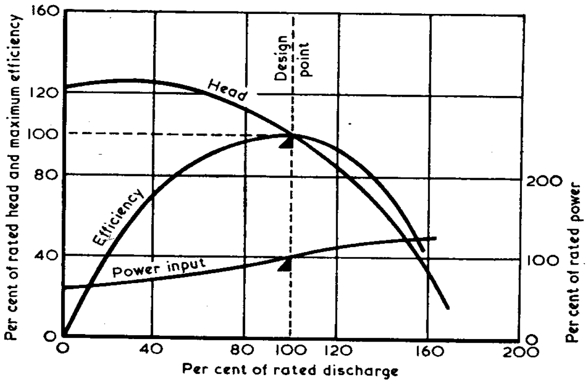
$$\eta = \eta_v \times \eta_m \times \eta_h \qquad \text{--------------} \qquad (6.13)$$

8.5.2 Characteristics

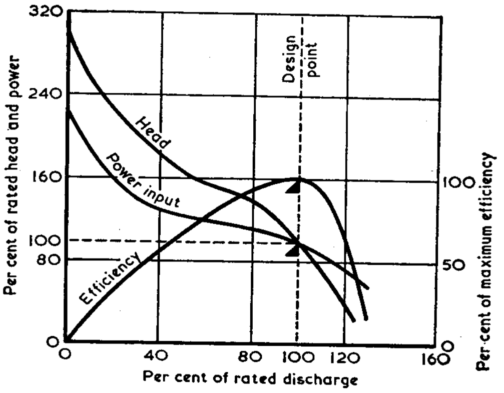
8.5.2.1 Pump characteristics

As the discharge is nearly the primary factor, it is customary for the performance curves to consist of the three curves of head, power input, and efficiency, drawn to common baseline of discharge.

Each design of pump has its own characteristic behavior. *Figure 6.10* shows the performance curves for the centrifugal and axial flow pumps. The curves are drawn for a particular operating speed.



(a) Centrifugal pump



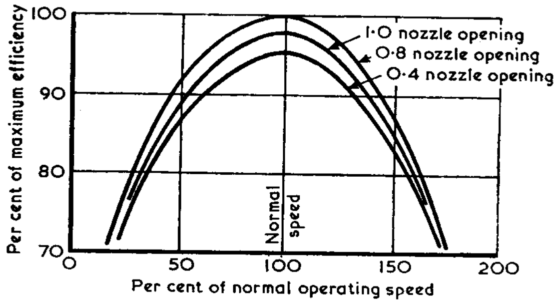
(b) Axial flow pump

Fig. 8.10 Pump characteristics

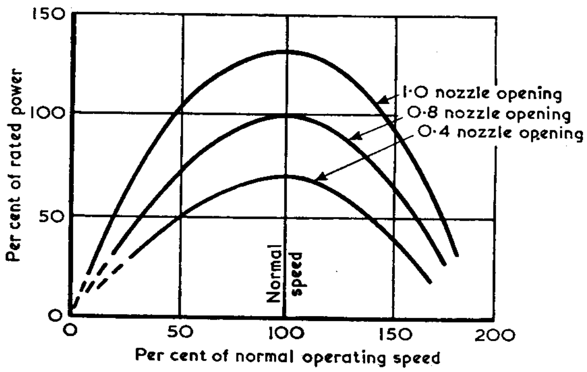
8.2.3 Turbine characteristics

As turbine output must be varied to suit the electrical demand it is customary to design the machine so that optimum efficiency occurs at about three-quarters of full load. Efficiency and power output are usually plotted against speed for a constant head.

Figure 8.11 shows typical performance curves for a Pelton turbine, while *figure 8.12* shows the corresponding curves for Francis turbine.



(a) Efficiency—speed curves



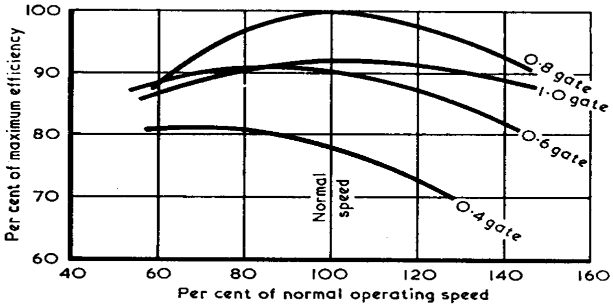
(b) Power—speed curves

Fig 8.11 Performance curves for a Pelton turbine

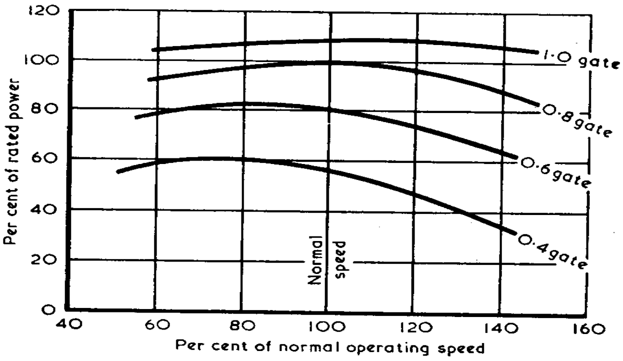
8.6 Cavitations

Cavitation occurs in both pumps and turbines. The primary cause of cavitation is a low pressure and this usually be brought about by a high local velocity. Cavitation is a harmful phenomenon and influences the design of the machines. It also imposes severe limitations on the machine setting, that is to say the permissible suction lift in the case of pumps and the height above the tail water in the case of turbines.

With pumps, the most vulnerable points for attack are the impeller vane tips at discharge. It is a result of high water velocities (low pressure) created near entry



(a) Efficiency—speed curves



(b) Power—speed curves

Fig. 8.12 Performance curves for a Francis turbine

in to an impeller. Here, vapour bubbles or cavities tend to form *(see figure 6.13)* which are then carried forward by the flow to a region of higher pressure near the exit where they collapse violently, causing pitting and severe damage to the impeller blades.

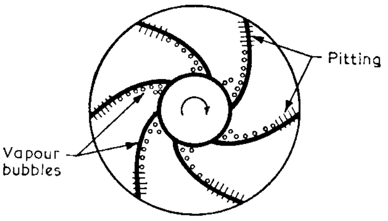


Fig. 8.13 Cavitation in a pump impeller

Cavitation also occurs in turbine runners in a similar manner. High velocities at the turbine inlet produce cavities which then collapse close to the runner blades near the exit.

Apart from the physical damage caused by cavitation, the reduction of the effective volume of the flow passages due to the presence of water vapour results in a smaller discharge and a sharp drop in efficiency. Additional evidence is the noise and vibration produced by the collapse of the vapour bubbles.