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CHAPTER (7)

Hydro-Water Power Plant

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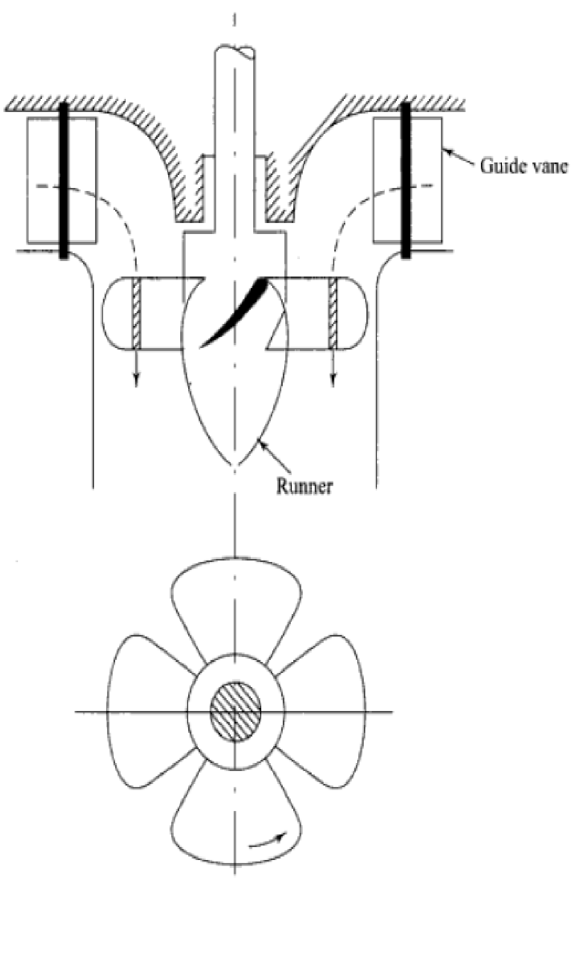

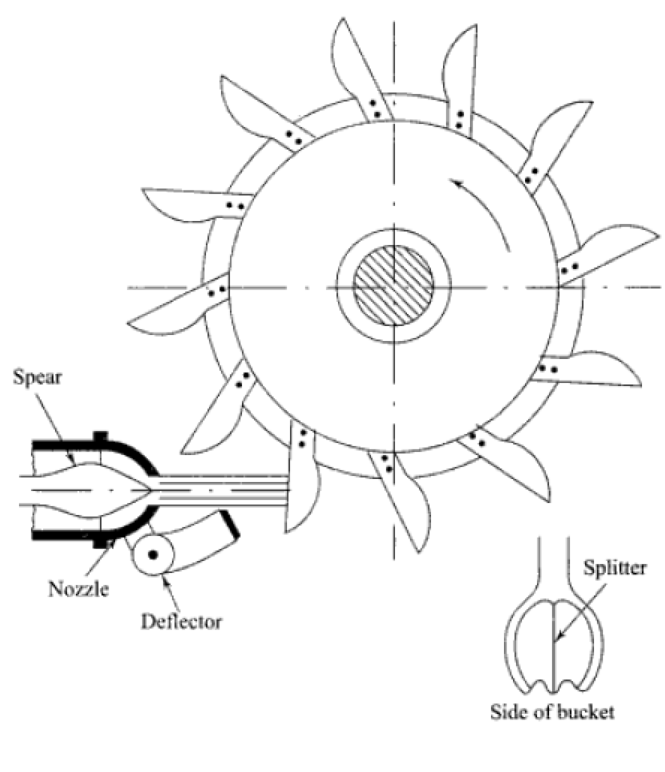

Chapter 7

Hydro-Water Power Plant

Definition of Hydraulic Turbines, Overall Efficiency, Specific Speed, Classification of Hydraulic Turbines: Kaplin Turbines, Francis Turbines, Pelton Turbines, Flow through Kaplin Turbine, Performance parameters of Kaplan Turbine, Tangential Force, Axial Force, Power Produced (Diagram Power), Hydraulic Efficiency, Example (1), Example (2), Flow through Francis Turbine, Performance parameters of Francis Turbine, Tangential Force, Power Produced (Diagram Power), Hydraulic Efficiency, Example (3), Example (4), Example (5), Example (6), Flow through Pelton Turbine, Performance parameters of Pelton Turbine, Tangential Force, Power Produced (Diagram Power), Hydraulic Efficiency, Example (7), Example (8), Example (9), Example (10), Governing of Impulse Turbine, Governing of Reaction Turbine, Tutorial Sheet 7.

10
Hydroelectric Power Plant

10.1 INTRODUCTION

In hydroelectric power plants the energy of water is utilized to drive the turbine which, in turn, runs the generator to produce electricity. Rain falling upon the earth's surface has potential energy relative to the oceans towards which it flows. This energy is converted to shaft work where the water falls through an appreciable vertical distance. The hydraulic power is thus a naturally available renewable energy source given by Eq. (10.1).

$$P = g\rho QH \tag{10.1}$$

Here P is the hydraulic power in Watts, g is 9.81 m/s^2 (the acceleration due to gravity), ρ is the water density, 1000 kg/m^3, Q is the flow or discharge, m^3/s and H is the height of fall of water or head, m. The electrical energy produced in kWh can then be written in the form of Eq. (10.2).

$$\begin{aligned} W &= 9.81 \times 1000 \times Q \times H \times \eta \times t \\ &= 9.81\ QH\ \eta t \text{ kWh} \end{aligned} \tag{10.2}$$

where t is the operating time in hours (8760 h/year) and η is the efficiency of the turbine-generator assembly, which varies between 0.5 and 0.9. The power developed thus depends on quantity (Q) and head (H) of water.

Hydro or water power is important only next to thermal power. Nearly 20 per cent of the total power of the world is met by hydropower stations. There are some countries like Norway and Switzerland where the hydropower forms almost the total installed capacity.

Hydroelectric power was initiated in India in 1897 with a run-of-river unit near Darjeeling. However, the first major plant was the Sivasamudram Scheme in Mysore of 4.5 MW capacity commissioned in 1902. Khopoli project of 50 MW in Maharashtra was put into operation in 1914 to supply power to Mumbai city. Since independence a substantial growth in hydropower has occurred with the commissioning of large multipurpose projects like

Damodar Valley Corporation (DVC), Bhakra Nangal, Hirakud, Nagarjunsagar, Mettur, Koyna, Rihand and so on.

10.2 ADVANTAGES AND DISADVANTAGES OF WATER POWER

These have been stated point by point as below.

10.2.1 Advantages of Water Power

Hydropower have some inherent advantages which make it very attractive.

1. Water source is perennially available. No fuel is required to be burnt to generate electricity. It is aptly termed as 'the white coal'. Water passes through turbines to produce work and downstream its utility remains undiminished for irrigation of farms and quenching the thirst of people in the vicinity.
2. The running costs of hydropower installations are very low as compared to thermal or nuclear power stations. In thermal stations, besides the cost of fuel, one has to take into account the transportation cost of the fuel also.
3. There is no problem with regards to the disposal of ash as in a thermal station. The problem of emission of polluting gases and particulates to the atmosphere also does not exist. Hydropower does not produce any greenhouse effect, cause the pernicious acid rain and emit obnoxious NO.
4. The hydraulic turbine can be switched on and off in a very short time. In a thermal or nuclear power plant the steam turbine is put on turning gear for about two days during start-up and shut-down.
5. The hydraulic power plant is relatively simple in concept and self-contained in operation. Its system reliability is much greater than that of other power plants.
6. Modern hydropower equipment has a greater life expectancy and can easily last 50 years or more. This can be compared with the effective life of about 30 years of a thermal or nuclear station.
7. Due to its great ease of taking up and throwing off the load, the hydro-power can be used as the ideal spinning reserve in a system mix of thermal, hydro and nuclear power stations.
8. Modern hydro-generators give high efficiency over a considerable range of load. This helps in improving the system efficiency.
9. Hydro-plants provide ancillary benefits like irrigation, flood control, afforestation, navigation and aqua-culture.
10. Being simple in design and operation, the hydro-plants do not require highly skilled workers. Manpower requirement is also low.

10.2.2 Disadvantages of Water Power

Major disadvantages of water power are the following:

1. Hydro-power projects are capital-intensive with a low rate of return. The annual interest of this capital cost is a large part of the annual cost of hydro-power installations.
2. The gestation period of hydro projects is quite large. The gap between the foundation and completion of a project may extend from ten to fifteen years.
3. Power generation is dependent on the quantity of water available, which may vary from season to season and year to year. If the rainfall is in time and adequate, then only the satisfactory operation of the plant can be expected.
4. Such plants are often far way from the load centre and require long transmission lines to deliver power. Thus the cost of transmission lines and losses in them are more.
5. Large hydro-plants disturb the ecology of the area, by way of deforestation, destroying vegetation and uprooting people. Strong public opinion against erection of such plants is a deterrent factor. The emphasis is now more on small, mini and micro hydel stations.

10.3 OPTIMIZATION OF HYDRO-THERMAL MIX

A hydroelectric power plant was earlier used as an exclusive source of power. However, it suffers seasonal variation of output proportional to the variation of water flow. To meet the variable load demand, large amount of water requires to be stored. At the times of low water flow rates the hydro plants cannot meet the maximum load. Again, if the maximum capacity of the station is based on the minimum water flow, this will prove uneconomical. There will be a great wastage of water over the dam for greater part of the year. Hence, the present trend is to use hydroelectric power in conjunction with thermal power in an interconnected system. This hydro-thermal mix is optimized to achieve minimum cost of power generation, which may be 30 per cent hydro-70 per cent thermal or 35 per cent hydro-65 per cent thermal. Load sharing by hydro is maximum when the available flow of water is maximum, say during the monsoon months. As long as there is planety of water stored in the reservoir the hydro part of the system carries the base load, with thermal plants taking the peaks. When water availability is low, say during the dry months of winter and spring, the steam plants take the base load and hydro plants meet the peak load (Fig. 10.1). By interconnecting hydropower with steam, a great deal of saving in cost can be effected by way of the following.

(i) Reduction in necessary reserve capacity.
(ii) Diversity of construction programmes.
(iii) Higher utilization factors of hydro-plants.
(iv) Higher capacity factors of thermal plants.

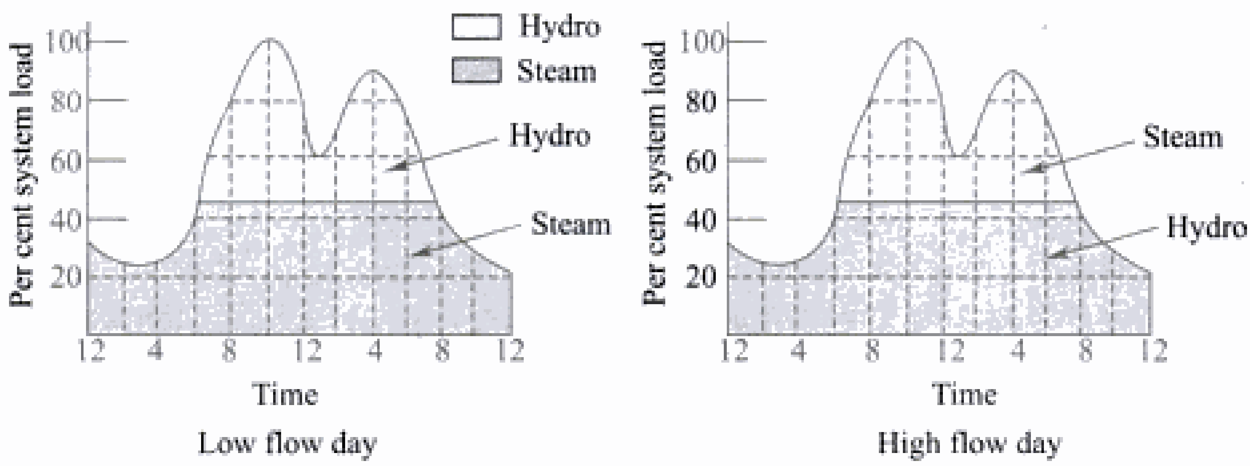

Fig. 10.1 *Typical division of load on a hydro-steam system*

10.4 SELECTION OF SITE FOR A HYDROELECTRIC PLANT

The following factors should be considered while selecting the site for hydroelectric power plant.

1. Availability of water
2. Water storage capacity
3. Available water head
4. Accessibility of the site
5. Distance from the load centre
6. Type of land of site

1. *Availability of water* The design and capacity of the hydro-plant greatly depends on the amount of water available at the site. The run-off data along with precipitation at the proposed site with maximum and minimum quantity of water available in a year should be made available to

(a) decide the capacity of the plant,
(b) set up the peak load plant such as steam, diesel or gas turbine plant,
(c) provide adequate spillways or gate relief during flood period.

2. *Water storage capacity* Since there is a wide variation in rainfall all round the year, it is always necessary to store the water for continuous generation of power. The storage capacity can be estimated with the help of mass curve.

3. *Available water head* In order to generate the desired quantity of power it is necessary that a large quantity of water at a sufficient head should be available. An increase in effective head, for a given output, reduces the quantity of water required to be supplied to the turbines.

4. *Accessibility of the site* The site should be easily accessible by rail and road. An inaccessible terrain will jeopardize the movement of men and material.

5. ***Distance from the load centre*** If the site is close to the load centre, the cost of transmission lines and the transmission losses will be reduced.

6. ***Type of the land of the site*** The land of the site should be cheap and rocky. The dam constructed at the site should have large catchment area to store water at high head. The foundation rocks of the masonry dam should be strong enough to withstand the stresses in the structure and the thrust of water when the reservoir is full.

10.5 HYDROLOGICAL CYCLE

Hydrology is the science that deals with the processes governing depletion and replenishment of water resources over and within the earth's surface. With the knowledge of hydrology at a certain site it is possible to design the irrigation and flood control works, power projects, water supply schemes, navigation works, etc.

As water vapour in atmospheric air goes up it cools, condenses and falls as rain, hail, snow or sleet. When this precipitation falls on hills and mountains and converges to form streams and rivers, it can be used for power generation. Intensity of rainfall, season and topography largely determine the usefulness of rainfall for power purposes. Light falls aid the growth of vegetation but do not contribute to stream flow. When total monthly precipitation concentrates in one or more storms, the *runoff* will increase greatly though vegetation may suffer. Distribution of precipitation may be classified as (i) direct evaporation (ii) absorption and transpiration by vegetation. (iii) seepage and storage; and (iv) direct surface runoff, eventually forming rivers (Fig. 10.2).

(i) A major part of precipitation on land areas that reaches the soil re-evaporates to the atmosphere, the rate being large from surfaces of lakes, ponds and swamps. A rise in temperature and drop in humidity increase the evaporation rate with the wind aiding it.

(ii) Plants absorb water through their roots and *transpire* it as vapour through their leaves to the atmosphere.

(iii) Precipitation absorbed by the soil *seeps* or percolates into the ground, forming bodies of water called the *water table* or *ground storage*. It is also called "infiltration" which is a process by which water enters the surface strata of the soil and makes its way downwards to the water table. The amount of seepage or infiltration depends on the geological character of the surface and subsoil.

(iv) The remaining water flows over the ground surface as direct runoff to form brooks and rivers (Fig. 10.2). The amount of runoff from a given rainfall depends on the nature of precipitation. Short, hard showers may produce relatively little runoff, whereas long rainfall saturates the soil lowering seepage rate and slows down evaporation by increased humidity and thus produces more runoff.

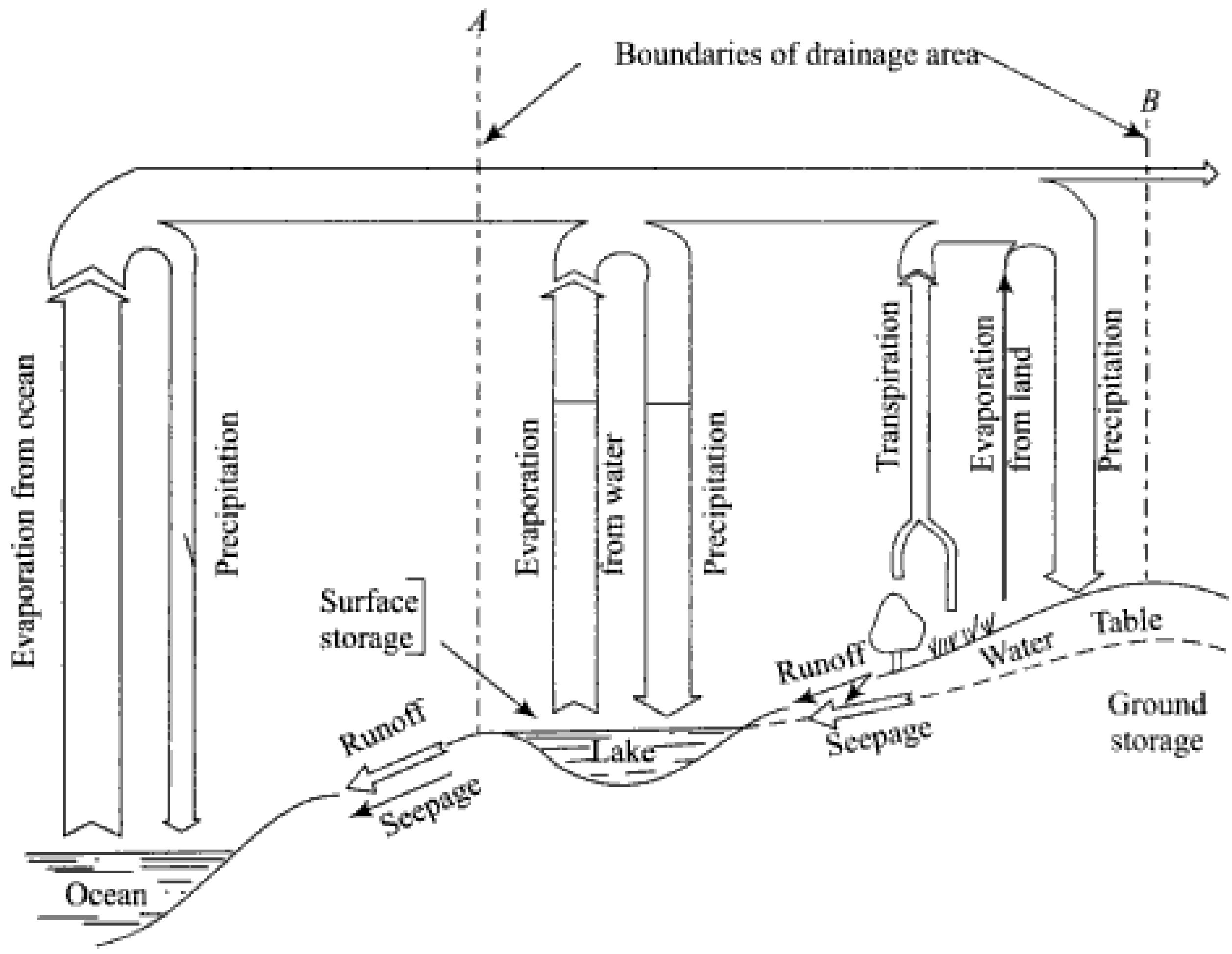

Fig. 10.2 *Graphic portrayal of the water cycle*

The water equation summarizing the disposal of the rainfall over a certain area during a given period is given by

Runoff + Seepage + Evaporation + Transpiration = Precipitation ± Change in storage

The best way to study the rainfall pattern is with the help of graphical plots. The *hyetographs* are the rainfall intensity-time curves which indicate the variation of the rate of rainfall with respect to time. The cumulative value of rainfall plotted against time represents the mass curve of rainfall.

10.6 HYDROGRAPHS

The variation of stream flow at a given site depends on the geographical, geological and topographical features of the drainage area feeding the river as well as the magnitude of the area rainfall. *Hydrographs* show the variation of river flow (discharge) with time. Runoff may be plotted as *flow duration curves* (Fig. 10.3 a), which show the time when a stream flow rate is equalled or exceeded in any period (daily, weekly or monthly basis). The area under the flow duration curve represents the average yield from the stream. By changing the ordinate to power (kW) instead of discharge (m^3/s) in Fig. 10.3 (a), the *power duration curve* is obtained and the area under the curve would then represent the average yield of power from the hydro-power project. It can be noted in Fig. 10.3. (b) that Q_m is the minimum flow rate that would be available

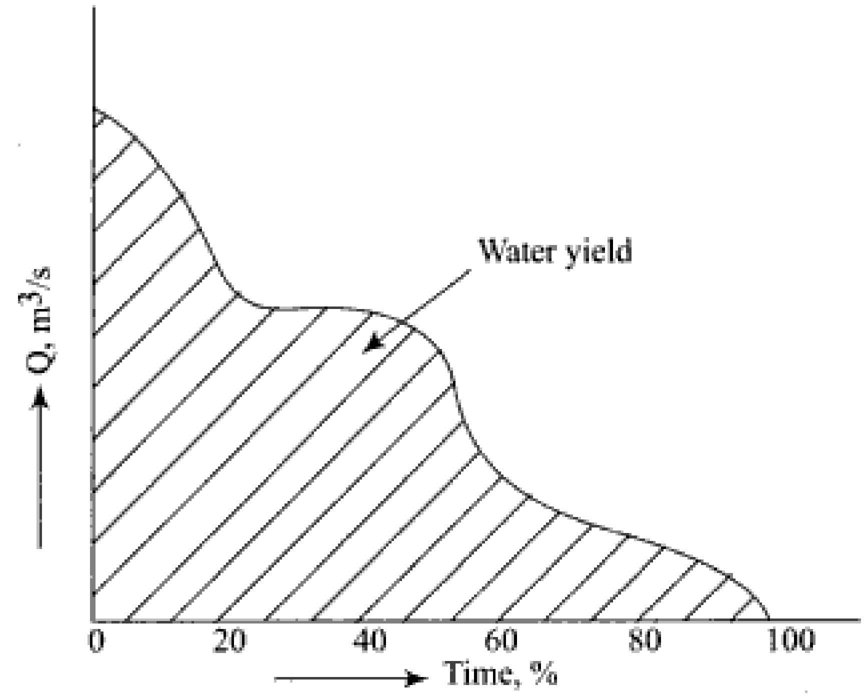

Fig. 10.3 (a) *Flow duration curve with % time on x-axis and Run-off on y-axis*

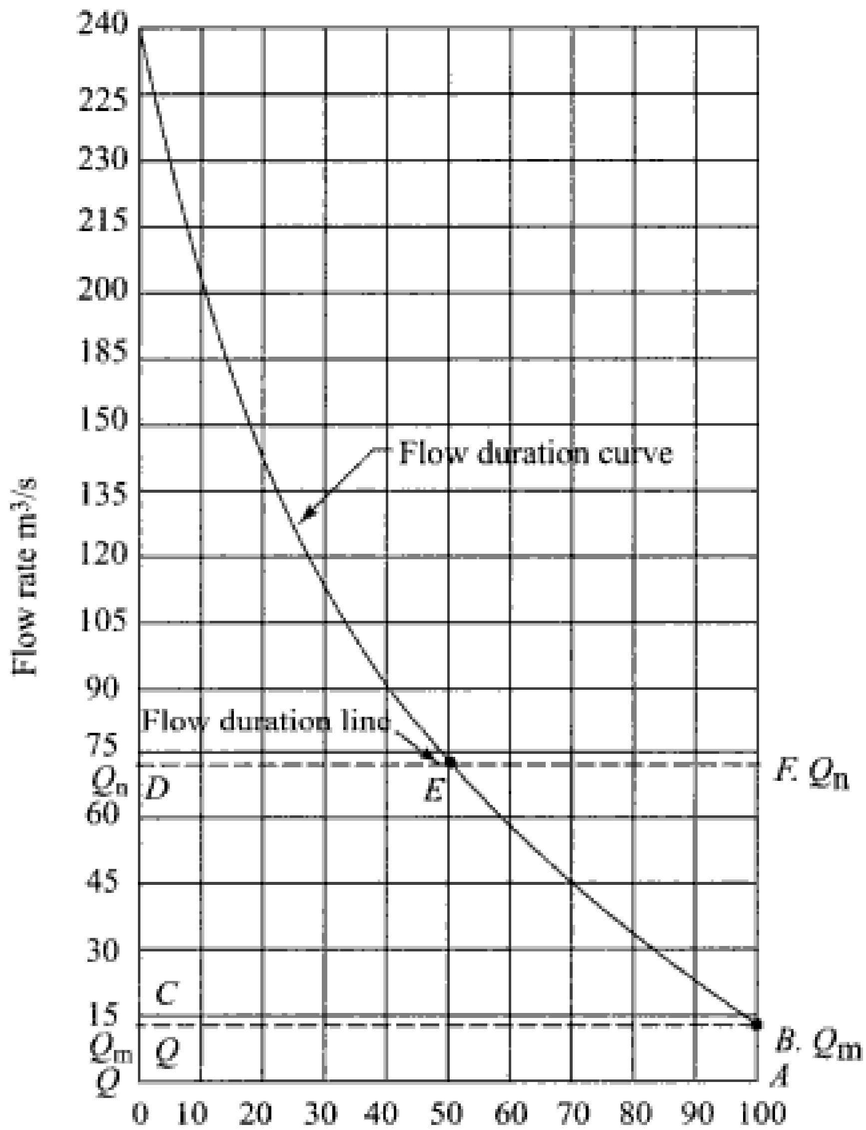

Fig. 10.3 (b) *Flow duration curve of a typical river having a low flow*

for all the times (i.e. for 100 per cent of time) and the area *OABC* would represent the firm yield of water or power, often termed as *primary power*. The additional output available at higher water flows is called *secondary power*. If a flow rate of Q_n is required for all the times as indicated by the area under the flow demand line *DEF*, then it would be possible to meet this uniform demand of flow rate (or power) for all the times only if storage equal to area *BEF* is provided. An alternative to this is to install a thermal power unit of *BF* capacity to work as a supplement to the hydro-power unit. The curve also shows that natural flow sufficient to meet the flow demand Q_n is available for 53.5 per cent of time or 195 days in the year of the lowest flow of the record. In the absence of any storage, area *BCDE* represents the secondary power that would be available from the river.

In order to facilitate the storage computation, mass curves are commonly used. A *mass curve* is a plot of accumulated flow (in hectare-metre) against time, made from the records of mean monthly flows of a stream (Fig. 10.4). The slope of the curve at any point indicates the rate of flow at that particular time. If the curve is horizontal, the flow is zero and if there is a high rate of flow the curve rises steeply. Relatively dry periods are indicated as concave depressions on the mass curve.

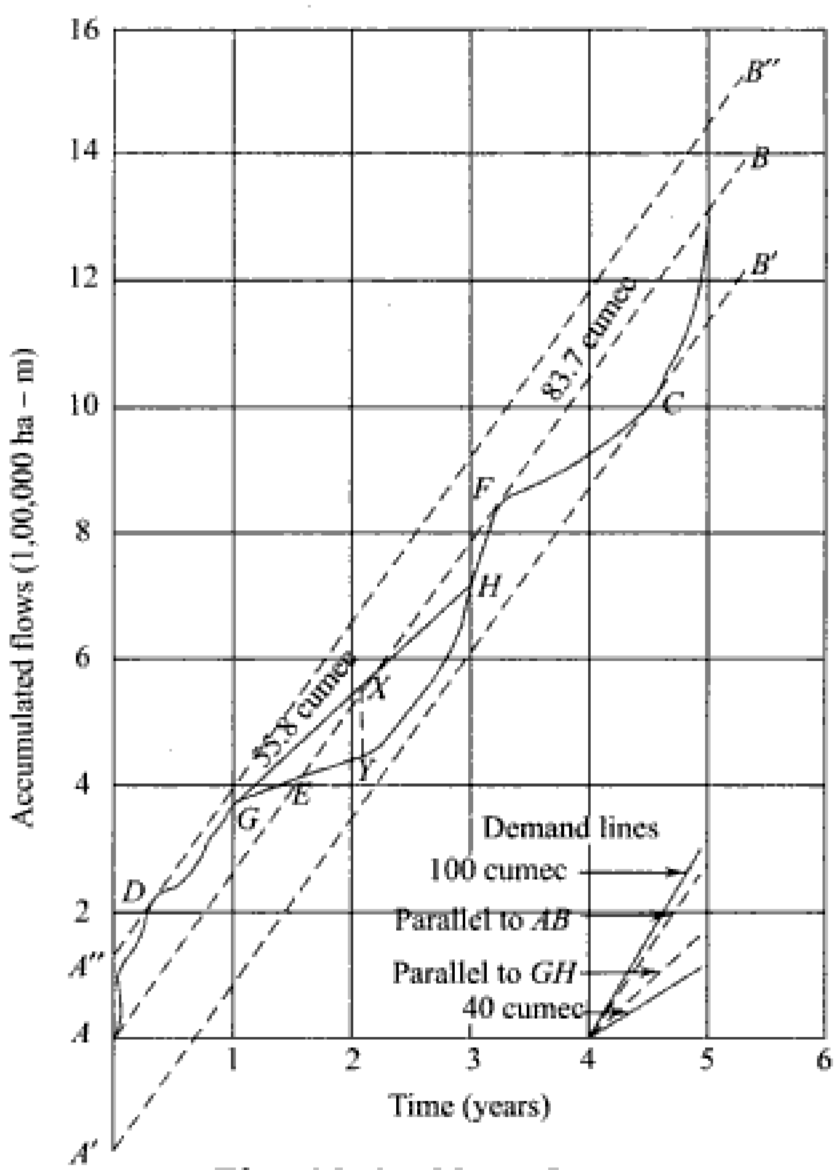

Fig. 10.4 *Mass Curve*

10.7 STORAGE AND PONDAGE

As stated earlier the flow rate of a stream varies considerably with time. For example, during rainy season when the stream is in floods it carries a huge quantity of water as compared to other times of the year when the quantity of water carried by it is considerably less. However, the demands for power ordinarily do not correspond to such variations of the natural flow of the stream. As such some arrangement in the form of storage and pondage of water is required for the regulation of the flow of water so as it make it available in requisite quantity to meet the power demand at a given time.

Storage may be defined as impounding of a considerable amount of excess run off during seasons of surplus flow for use in dry seasons. This is accomplished by constructing a dam across the stream at a suitable site and building a storage reservoir on the upstream side of the dam.

Pondage may be defined as a regulating body of water in the form of a relatively small pond or reservoir provided at the plant. The pondage is used to

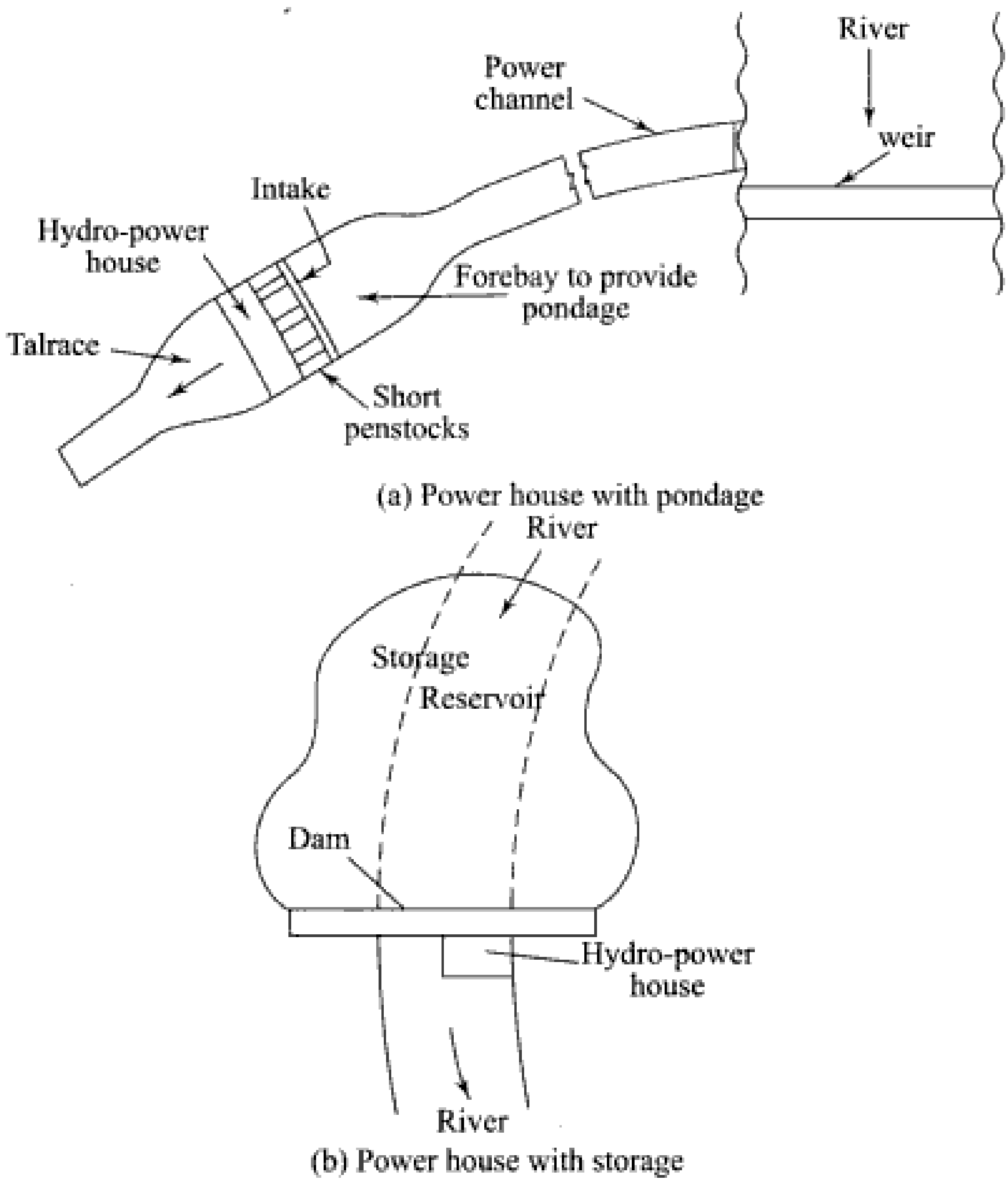

Fig. 10.5 *Hydro-power units with pondage and storage*

regulate the variable water flow to meet power demand. It caters for short-term fluctuations which may occur due to (a) sudden increase or decrease of load on the turbine (b) sudden changes in the inflow of water, say by breaches in the conveyance channel (c) change of water demand by turbines and the natural flow (supply) of water from time to time. The turbines are often required to meet the power demand higher than the average load when the pondage supplies the excess quantity of water required during that period. Figure 10.5 shows the locations of power houses with storage and pondage. Pondage increases the capacity of a river over a short-time, such as a week. Storage, however, increases the capacity of a river over an extended period of 6 months to as much as 2 years.

10.8 ESSENTIAL ELEMENTS OF A HYDROELECTRIC POWER PLANT

Figure 10.6 gives the flow diagram of a typical hydroelectric power plant. The essential elements of such a plant are the following.

1. Catchment area
2. Reservoir
3. Dam
4. Spillways
5. Conduits
6. Surge tanks
7. Draft tubes
8. Powerhouse
9. Switch yard for transmission of power.

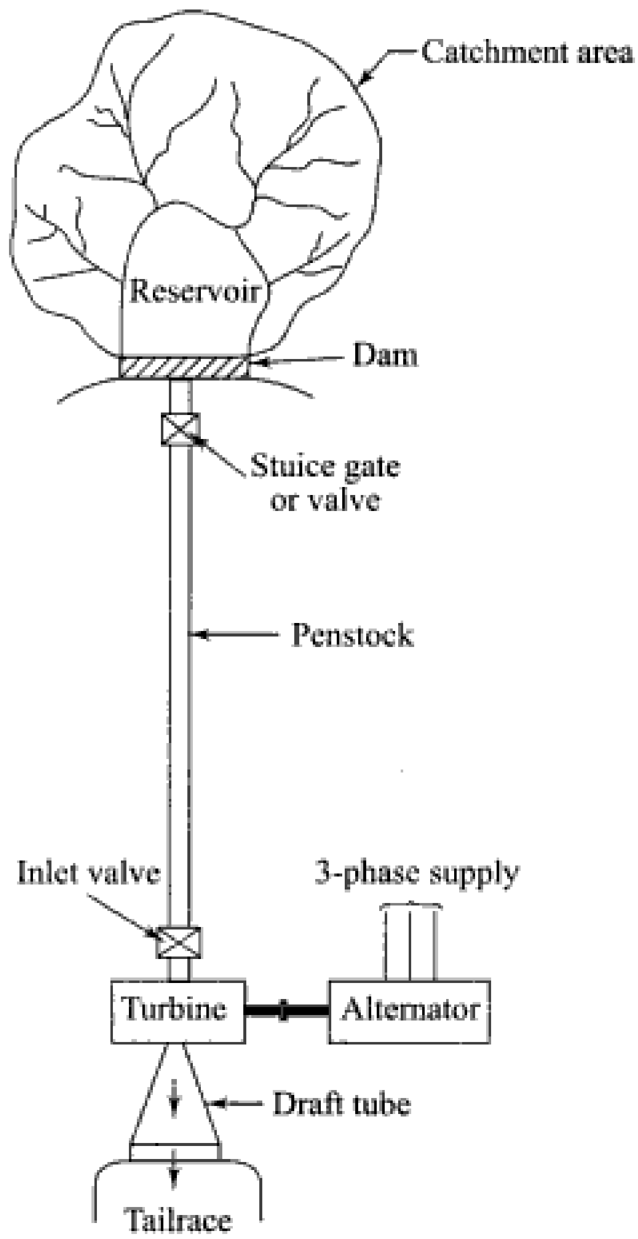

Fig. 10.6 *Flow sheet of a hydroelectric power plant*

10.8.1 Catchment Area

The whole area behind the dam draining into a stream or river across which the dam has been constructed is called the catchment area. The characteristics of the catchment include its size, shape, surface, orientation, altitude, topography and geology. The bigger the catchment, steeper is the slope, higher is the altitude, and greater is the total runoff of water.

10.8.2 Reservoir

Storage during times of plenty for subsequent use in times of scarcity is fundamental to the efficient use of water resources. The management of

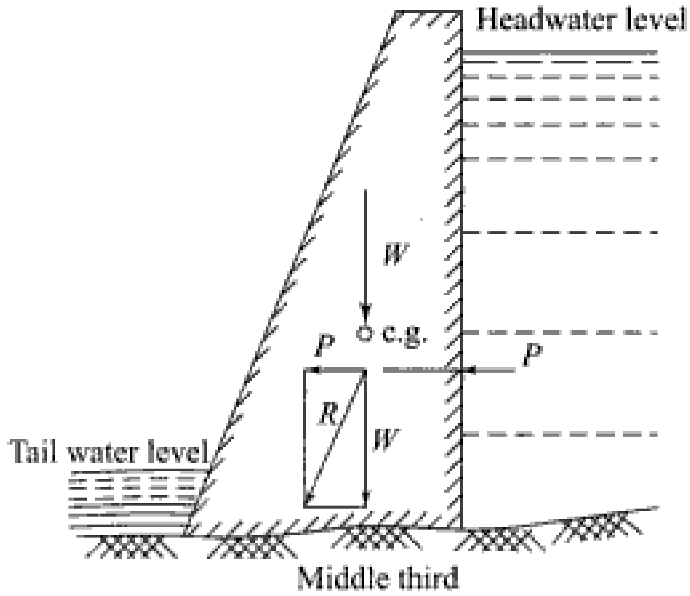

Fig. 10.8 *Cross-section of solid gravity type of masonry dam*

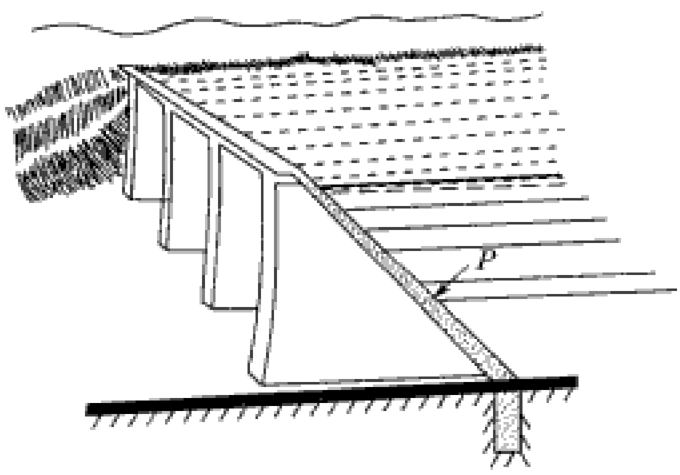

Fig. 10.9 *Buttress or hollow gravity type of masonry dam with flat deck*

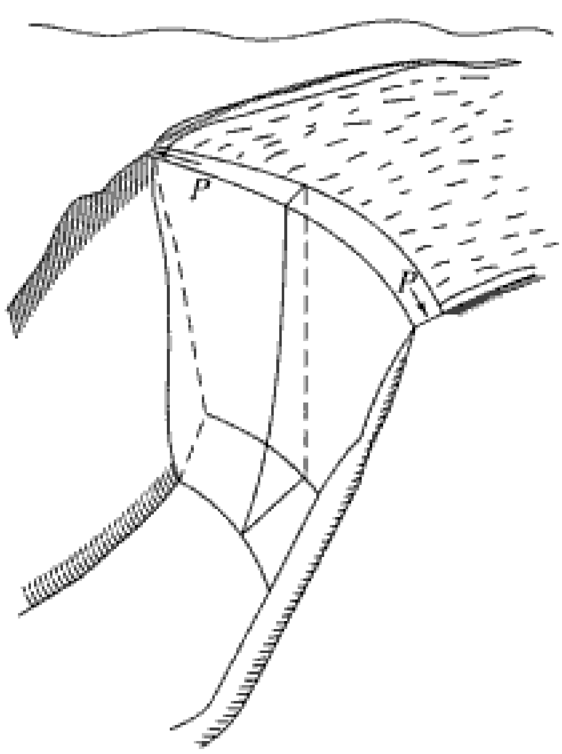

Fig. 10.10 *Arch type of masonry dam*

Earth dams For a small project of up to 70 m in height, dams constructed of earth fill or embankment are used. A large volume of material is required and it should be available in the vicinity. The dam construction varies with the height and the side slopes are flatter (Fig. 10.11). It is cheaper than masonry dam, but has more seepage losses. There may be serious damage from erosion by water overtopping the dam or seeping through it.

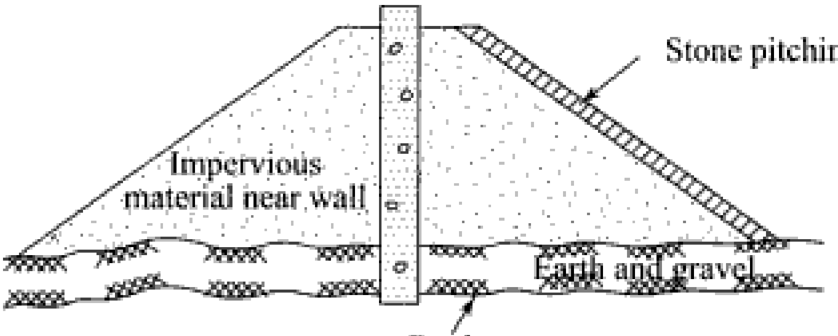

Fig. 10.11 *Earth dam*

Rock-fill dams It is made of loose rock of all sizes and has a trapezoidal shape with a wide base, having a watertight section to reduce seepage. It is used in mountainous region where rock is available.

10.8.4 Spillways

When the water level in the reservoir basin rises, the stability of the dam structure is endangered. To relieve the reservoir of this excess water, a structure is provided in the body of a dam or close to it. This safeguarding structure is called a *spillway*. It provides structural stability to the dam under conditions of floods without raising reservoir level above H.F.L. (high flood level). Following are the various types of spillways.

1. ***Overall spillway*** It is also called solid gravity spillway. It is provided in concrete and masonry dams (Fig. 10.12). Water spills and flows over the crest in the form of a rolling sheet of water. The bucket at the lower end changes the direction of the fast moving water, destroying its excess energy.

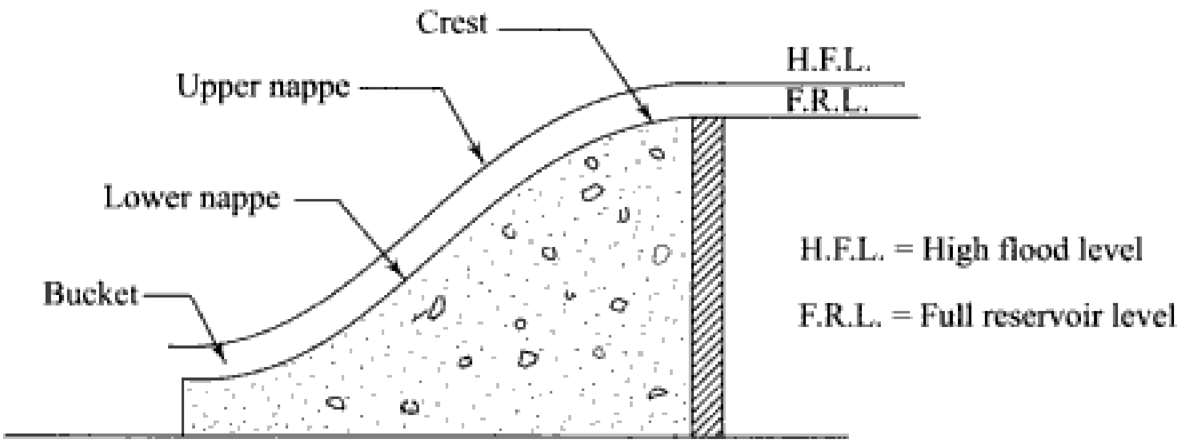

Fig. 10.12 *Overall spillway*

2. ***Chute or trough spillway*** It is suitable when the valley is too narrow to accommodate the solid gravity spillway in the body of the dam. After crossing over the crest the water shoots down a channel or trough to meet the river downstream of the dam.

3. ***Side channel spillway*** This is used when the valley is too narrow and in non-rigid dams where the flood water is not desired to flow over the dam. When there is no room to provide chute spillway, the side channel spillway (Fig. 10.13) is used. Here, after crossing the crest water flows parallel to it.

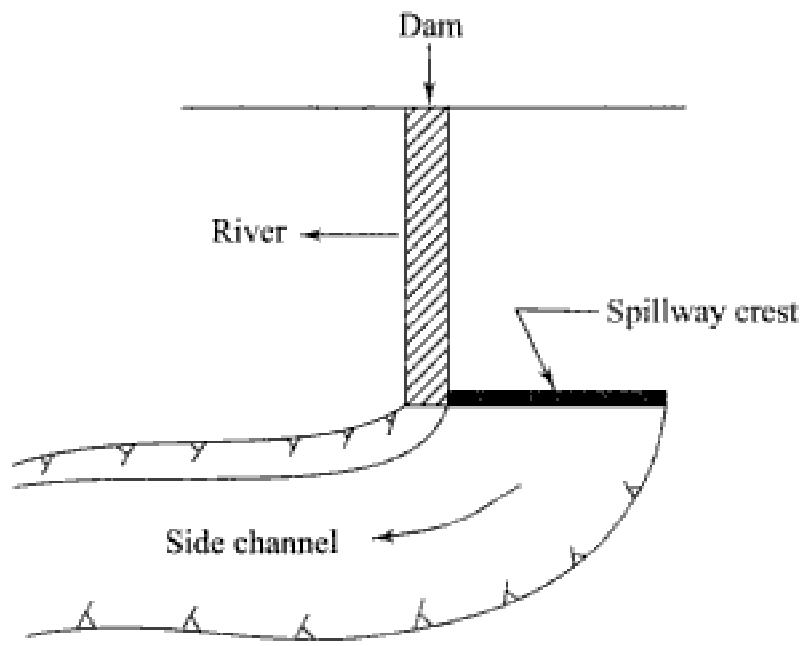

Fig. 10.13 *Side channel spillway*

4. ***Saddle spillway*** When conditions are not favourable for any of the above types of spillway, a saddle spillway is used. Some natural depression or saddle on the periphery of the reservoir basin away from the dam is used as the spillway, with the bottom of the depression being at the full reservoir level (Fig. 10.14).

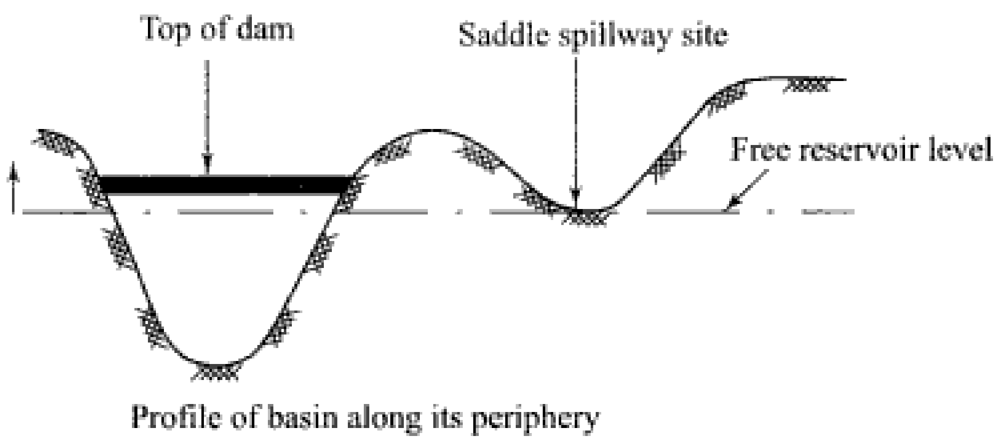

Fig. 10.14 *Saddle spillway*

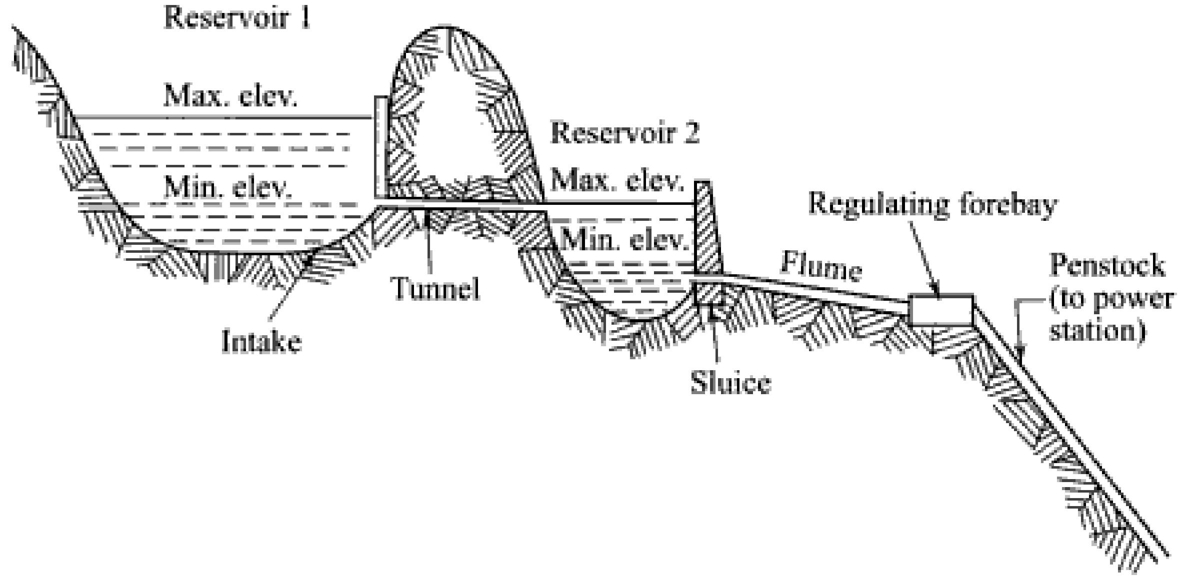

Fig. 10.16 *Combination of tunnel, flume and penstocks at a high head power plant*

10.8.6 Surge Tanks

A surge tank is a small reservoir in which the water level rises or falls to reduce the pressure swings so that they are not transmitted to the closed conduit. If the power house is located within a short distance of the headworks, surge tanks are not necessary. Thus for run off plants and medium head schemes no surge tank is needed. Surge tanks are required for high head plants where water is taken to the power house through tunnels and penstocks. A typical arrangement is shown in Fig. 10.17, where the surge tank is a vertical standpipe connected to the penstock

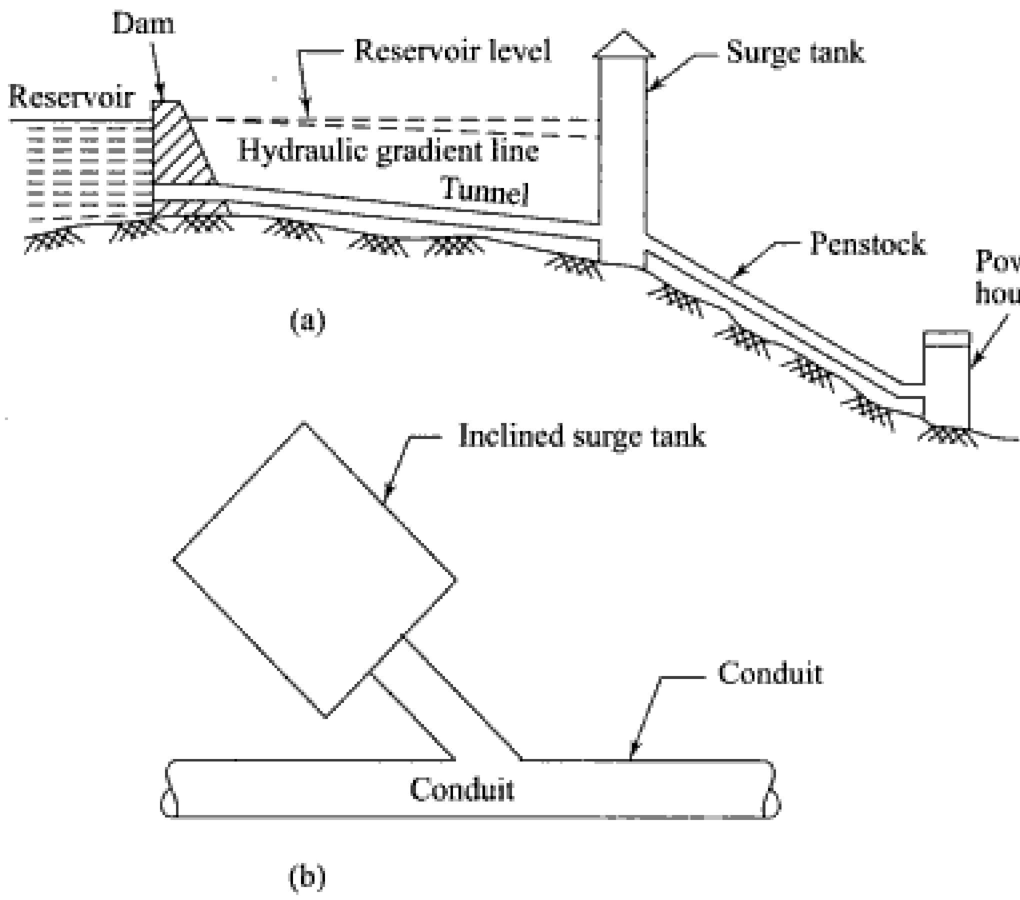

Fig. 10.17 *(a) Surge tank on ground level, (b) Inclined surge tank*

with no overflow of water. A further discussion on surge tanks has been made in Section 10.23.

10.8.7 Draft Tubes

The draft tube allows the turbine to be set above the tailrace to facilitate inspection and maintenance and by diffuser action regains the major portion of the kinetic energy or velocity head at runner outlet, which would otherwise go waste as an exit loss. The draft tube can be a straight conical tube (Fig. 10.18 a) or an elbow tube (Fig. 10.18 b). The conical type is used for low power units, while the elbow type is more common. In the elbow type energy is regained in the vertical portion which flattens in the elbow section to discharge water horizontally to the tailrace.

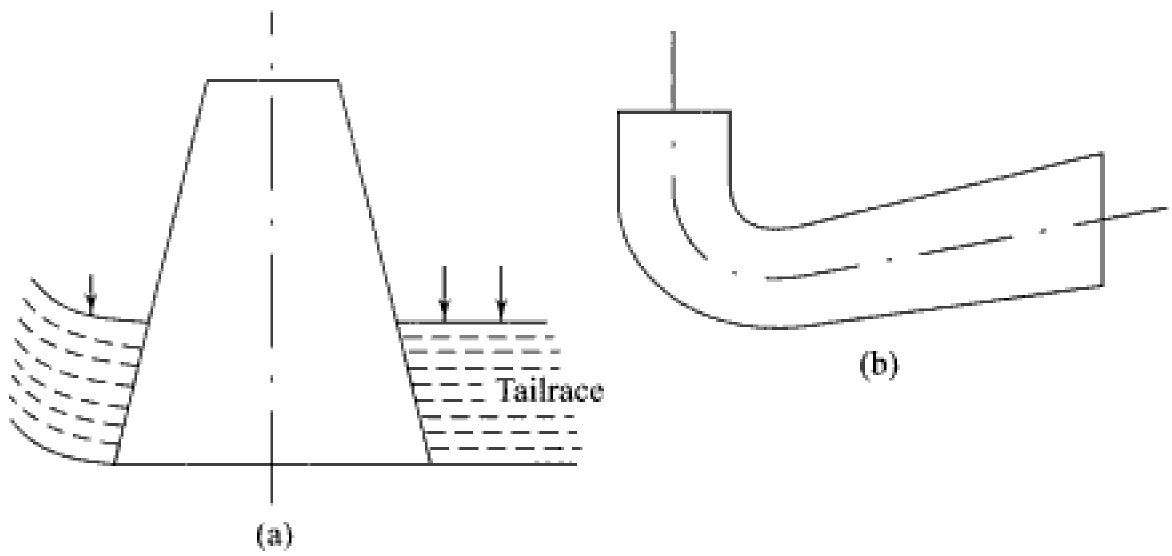

Fig. 10.18 *(a) Straight conical draft tube (b) Elbow type draft tube*

10.8.8 Powerhouse

A powerhouse should have a stable structure and its layout should be such that adequate space is provided around the equipment for convenient dismantling and repair. The equipment provided in the powerhouse includes the following.

(i) Hydraulic turbines
(ii) Electric generators
(iii) Governors
(iv) Gate valves
(v) Relief valves
(vi) Water circulation pumps
(vii) Air duct
(viii) Switch board and instruments
(ix) Storage batteries
(x) Cranes

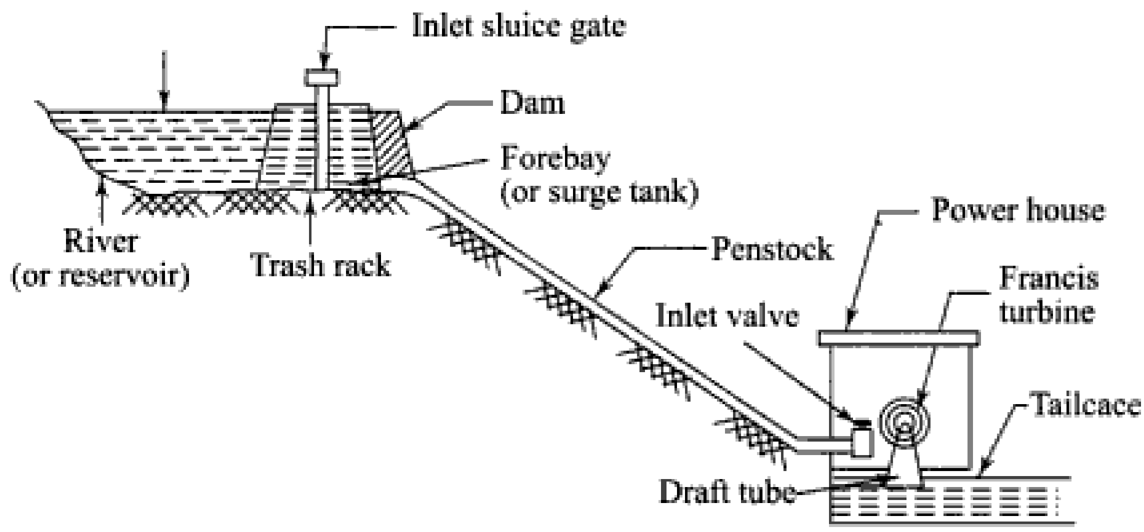

Fig. 10.20 *Medium head power plant*

10.9.3 Low Head Power Plants

A dam is constructed across a river and a sideway stream diverges from the river at the dam. Later this channel joins the river further downstream (Fig. 10.21). Francis turbine or Kaplan turbine is used for power generation.

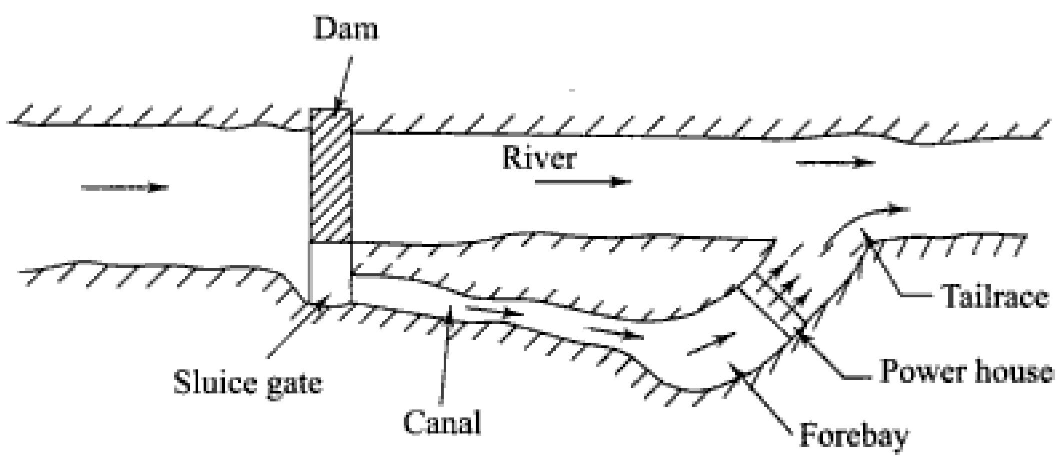

Fig. 10.21 *Low head power plant*

10.9.4 Base Load Plants

These plants are required to supply constant power to the grid. They run continuously without any interruption and are mostly remote controlled.

10.9.5 Peak Load Plants

They only work during certain hours of a day when the load is more than the average. Thermal stations work with hydel plants in tandem to meet the base load and peak load during various seasons.

10.9.6 Run-Of-River Plants with or Without Pondage

Such a plant works daily according to the nature and limit to the flow in the river. Power generated depends on the quantity of flow. Sometimes, a small storage reservoir or pond is built, which can store a few hours' supply of water to the plant, when the river flow exceeds the amount required by the plant. Such a scheme is called a run-of-river plant with pondage. The pondage or stored water is used in generating power during the hours when the demand is in excess of the flow of the river at the moment.

10.9.7 Hydroelectric Plants with Storage Reservoir

These plants are most common in India. During the rainy season water is stored in reservoirs so that it can be utilized during other seasons to supplement the flow of the river whenever the flow in the river falls below a specified minimum. Power can be generated directly from the reservoir. Sometimes canals are constructed to convey water from the reservoir for irrigation purposes.

10.9.8 Pumped Storage Plants

Water after working in turbines is stored in the tailrace reservoir. During *low load*, say night time, the water is pumped back from the tail to the head reservoir drawing excess electricity from the grid or from the nearby steam plant. During *peak load*, this water is used to work on turbines to produce electricity (Fig. 10.22). It is always economical to run the steam power plants all the time at full plant capacity factor. Whenever the load demand is less than the full plant capacity, the surplus energy instead of being wasted is transmitted to a pump is installed at the tailrace of the hydroelectric plant. The advantages of such a plant can be summarized as follows:

(a) Substantial increase in peak load capacity at low cost (b) High operating efficiency (c) Better load factor (d) Independence of stream flow conditions.

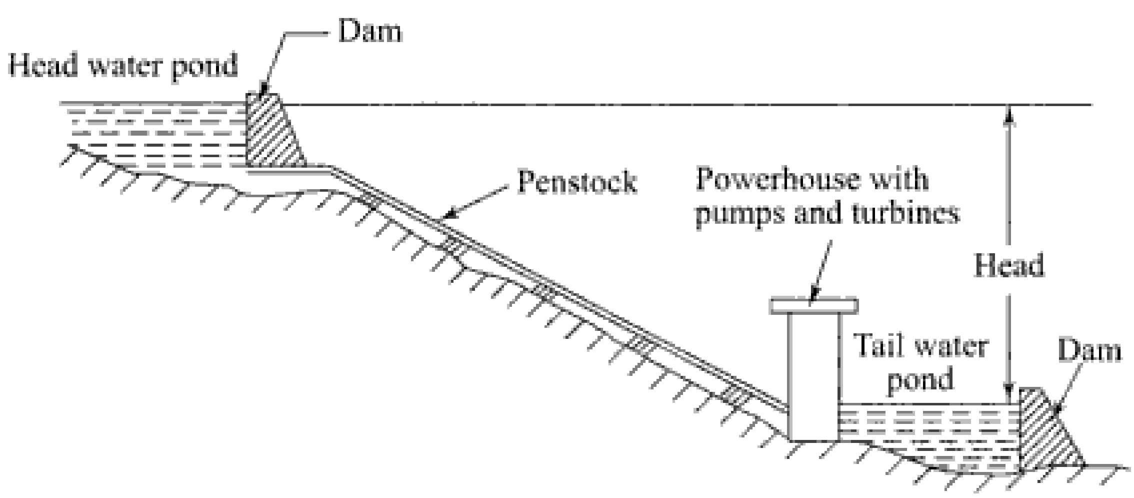

Fig. 10.22 *Pumped storage plant*

About 70 per cent of power used in pumping is recovered. A motor-generator set and a reversible turbine-pump unit can be profitably used as the same machine can be operated as a motor or generator and similarly as a pump or a turbine (Deriaz turbine).

10.9.9 Mini and Micro-Hydel Plants

More emphasis is now being given on such plants. The natural water source in hilly terrain can be utilized for power generation with low-head standardized turbo-generator units. Its adverse effect on ecology is negligible. The mini-plants operate with 5 m–20 m head producing about 1 MW to 5 MW of power, while micro-plants are still smaller and work under a head of less than 5 m and generate electricity between 0.1 MW to 1 MW. The potential energy source in India in this category is around 20,000 MW.

10.10 HYDRAULIC TURBINES

Hydraulic turbines convert the potential energy of water into shaft work, which, in turn, rotates the electric generator coupled to it in producing electric power. Historically, hydraulic turbines of today are derived from the waterwheels of the middle ages used for flour mills (to grind wheat) and ore-crushing. One such waterwheel (pan-chakki) can still be seen at Aurangabad, which is, at least, four hundred years old. Modern turbines have undergone many technological advances in diverse areas like fluid mechanics, metallurgy and mechanical engineering.

10.10.1 Classification of Hydraulic Turbines

The hydraulic turbines can be classified according to the (a) head and quantity of water available (b) name of the originator (c) nature of working on the blades (d) direction of flow of water (e) axis of the turbine shaft (f) specific speed.

1. ***According to the head and quantity of water available*** The difference in elevation of water surface between upstream and downstream of the turbine is the head under which the turbine acts (Fig. 10.23). The turbines work under a wide range of heads varying from 2 to 2000 m. A classification of turbine based on head as follows.

Low head	2–15 m
Medium head	16–70 m
High head	71–500 m
Very high head	Above 500 m

For low heads, only Kaplan or propeller turbines are used. For medium heads either Kaplan or Francis turbines are used. For high heads either Francis or Pelton turbines are used. For very high heads, invariably Pelton turbines are used. Deriaz turbines are used up to a head of 300 m. Their use is, however, restricted under reversible flow conditions (i.e. pumped-storage plants where the turbine also works as a pump).

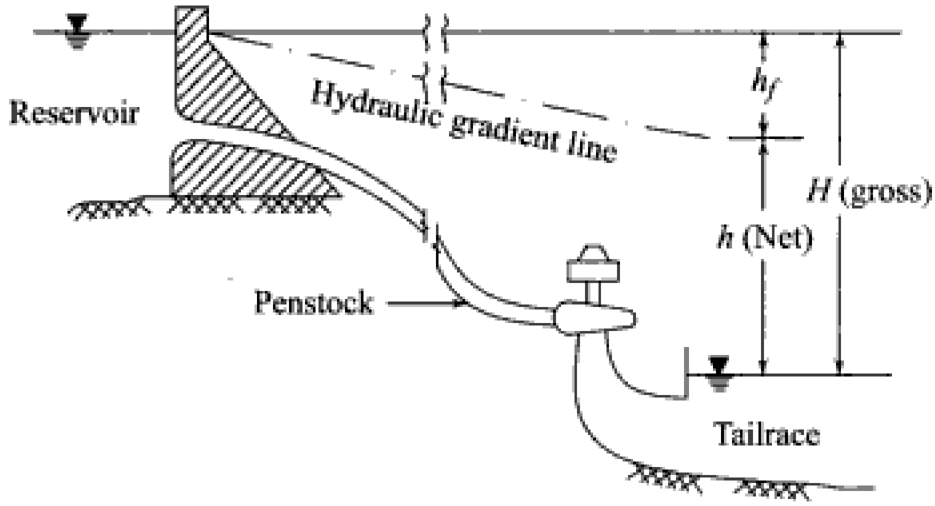

Fig. 10.23 *Head on the turbine*

Turbines can also be classified as low discharge, medium discharge and high discharge turbines, depending on the flow available. Pelton turbines are relatively low discharge turbines. Kaplan turbines are high discharge turbines, while Francis turbines occupy an intermediate position in this regard.

2. ***According to the name of the originator*** (i) Pelton turbine-named after Lester Allen Pelton of the USA, an impulse turbine used for high head and low discharge.
   (ii) Francis turbine—named after James B. Francis, a reaction turbine used for medium head and medium discharge.
   (iii) Kaplan turbine—named after Dr. Victor Kaplan, a reaction turbine used for low head and large discharge.
   (iv) Deriaz turbine—named after the Swiss engineer Deriaz, a reversible turbine-pump used up to a head of 300 m.

3. ***According to the nature of working on the blades*** Turbines are classified as impulse and reaction turbines depending on the mode of energy conversion of potential energy of water into shaft work. In an impulse turbine all the available head of water is converted into kinetic energy in a nozzle. The water shoots out of the nozzle in a free jet into a bucket which revolves round a shaft. During this action, the water is in contact with air all the time and the water discharged from bucket falls freely through the discharge passage into the tailwater. The free jet is at atmospheric pressure before and after striking the vanes. These are pressureless or impulse turbines. Pelton wheel belongs to this category.

   In reaction turbines, the entire flow from the headwater to the tailwater takes place in a closed conduit system which is not open to the atmosphere at any point in its passage. At the entrance to the runner, only a part of P.E. is converted into K.E. and the remaining into pressure energy. The runner converts both K.E. and pressure energy into mechanical energy. Such turbines are called reaction or pressure

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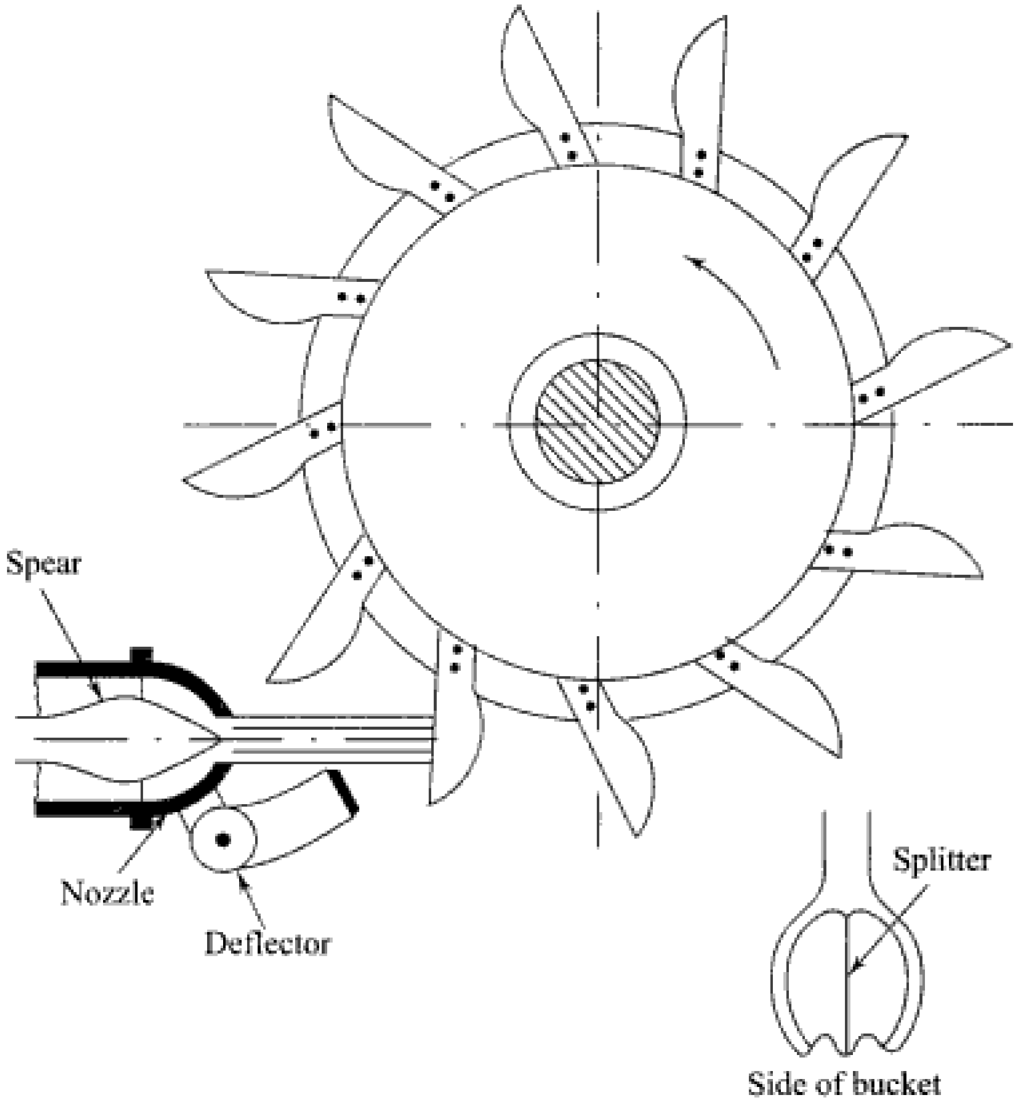


Fig. 10.24 *(a) Pelton wheel*

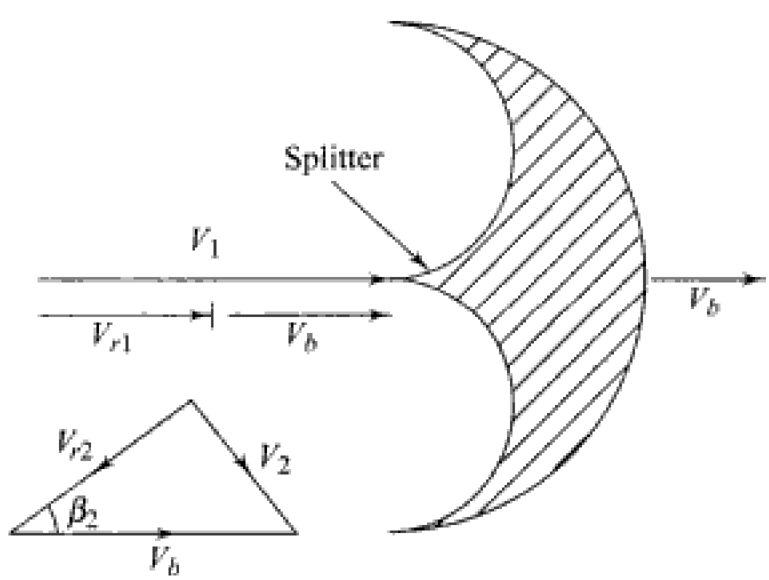


Fig. 10.24 *(b) Velocity diagrams and section through a bucket*

The nozzle directs the flow on the wheel. It also governs the quantity of flow with the help of a *spear valve* controlled by the governor action. In the simple arrangement there is a single nozzle feeding water to the turbine. However, for larger discharge there are turbines having up to six jets, all symmetrically arranged and causing rotation in the same direction. Figure 10.25 shows a multi-jet arrangement with four jets. Multi-jet machines usually have vertical shafts.

The specific speed of a multi-jet machine, i.e., N_{smj} is given by the following reaction.

$$N_{S_{MJ}} = \sqrt{n}\, N_{S_{SJ}} \tag{10.5}$$

where $N_{S_{SJ}}$ is the specific speed for a single-jet machine and n is the number of jets. Thus the specific speed of a given wheel can be increased by using multi-jet arrangement. The maximum number of jets used so far is six and the maximum speed for a single jet is of the order of 30. Thus the maximum specific speed for multi-jet machines is about 70.

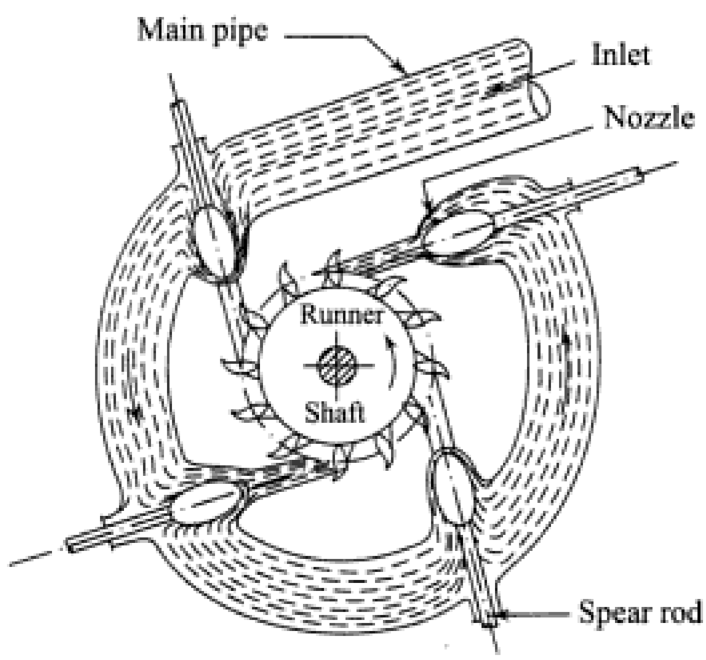


Fig. 10.25 *Multi-jet Pelton wheel arrangement*

It may be noted that the bucket deflection angle is of the order of 165° (Fig. 10.26). The slightly oblique direction of existing water allows it to escape freely without hitting the back of the next bucket. The water after leaving the bucket drops freely into the tailrace.

The jet moves in a tangential plane before and after striking the wheel and the bucket moves at a speed given by Eq. (10.6)

$$V = \omega r = \frac{\pi D N}{60} \tag{10.6}$$

where r and D are the bucket circle radius and diameter respectively and ω is the angular velocity given by $\frac{2\pi N}{60}$, N being the rpm.

With the nozzle diameter d, D/d is a size parameter for the turbine. This is known as *jet ratio*, *m*, having a value in the range of 10 to 24. The net head available at the nozzle is equal to the gross head less losses in the pipeline. If it is equal to H, the velocity of jet issuing from the nozzle is as follows.

$$V_1 = C_v [2gH]^{1/2} \qquad (10.7)$$

where C_v is the coefficient of velocity (0.97 – 0.99).

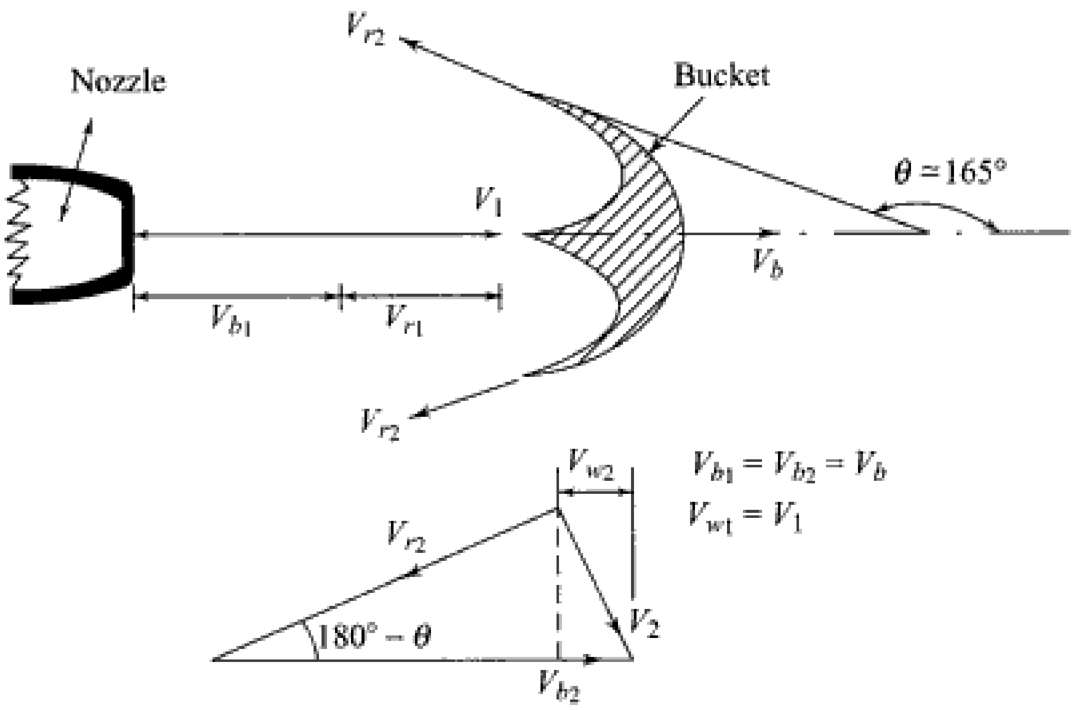


Fig. 10.26 *Bucket deflection angle and velocity diagrams*

Total energy transferred to the wheel (Fig. 10.26) is given by Euler equation (Eq. 10.8).

$$E = (V_{w_1} V_{b_1} - V_{w_2} V_{b_2})/g = \frac{V_b}{g} (V_{w_1} - V_{w_2}) \qquad (10.8)$$

where subscript 1 represents the condition at inlet and subscript 2 the condition of water at outlet of the bucket, V_w is the velocity of whirl (tangential component) and V_b is the bucket velocity given by $V_{b1} = V_{b2} = \frac{\pi DN}{60}$

Now, from the exit velocity diagram (Fig. 10.26), we get

$$V_{w2} = V_b - V_{r2} \cos (180 - \theta) = V_b + V_{r2} \cos \theta \qquad (10.9)$$

where θ is the bucket deflection angle (~ 165°).

Now, V_{r2} = relative velocity of water at exit

$$= kV_{r1} = k(V_1 - V_b) \qquad (10.10)$$

where k is the blade friction coefficient, V_1 is the absolute velocity of water from the jet, and V_{r1} is the relative velocity of water at inlet.

$$V_{w2} = V_b + k(V_1 - V_b) \cos \theta \qquad (10.11)$$

Substituting V_{w2} from Eq. (10.11) in Eq. (10.8) and since $V_{w1} = V_1$, we get

$$E = \frac{V_b}{g} [V_1 - V_b - k (V_1 - V_b) \cos \theta]$$

$$= \frac{V_b}{g}(V_1 - V_b)(1 - k\cos\theta)$$

$$= \frac{1-k\cos\theta}{g}\left(V_1 V_b - V_b^2\right) \qquad (10.12)$$

For given values of V_1, k and θ, there is a certain value of V_b for which E is maximum. Differentiating E with respect to V_b and putting it equal to zero, we get

$$\frac{dE}{dV_b} = \frac{1-k\cos\theta}{g}(V_1 - 2V_b) = 0$$

$$V_b = \frac{V_1}{2} \qquad (10.13)$$

Therefore, the optimum bucket velocity for maximum work output is half the jet velocity. On its substitution, we get

$$E_{max} = \frac{1-k\cos\theta}{g}\frac{V_1^2}{4} \qquad (10.14)$$

The kinetic energy of the input jet $= \dfrac{V_1^2}{2g}$.

Therefore, the blading or diagram or hydraulic efficiency of the wheel is given by

$$\eta_D = \frac{E}{V_1^2/2g} = \frac{1-k\cos\theta}{g}\left(V_1 V_b - V^2_b\right)\frac{2g}{V_1^2}$$

$$= 2(1 - k\cos q)(\rho - \rho^2) \qquad (10.15)$$

where ρ is the velocity ratio, $\dfrac{V_b}{V_1}$ (Fig. 10.27)

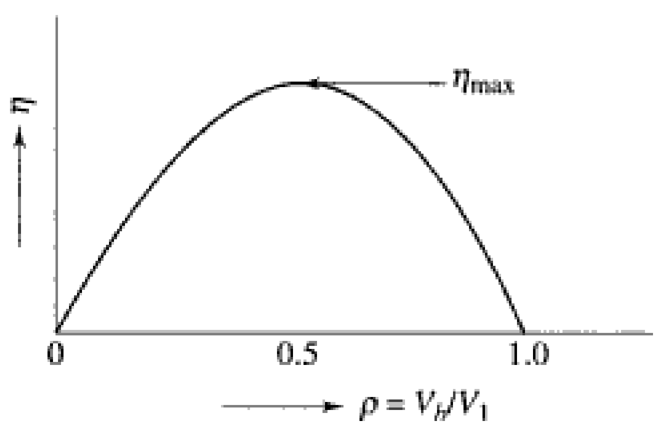


Fig. 10.27 *Variation of x with velocity ratio*

Making $$\frac{d\eta_D}{d\rho} = 0_1, \text{ we get}$$

$$2(1 - k\cos q)(1 - 2\rho) = 0$$

or $$(\rho_{opt})_{max\ effy} = 1/2 \tag{10.16}$$

Thus, for maximum hydraulic efficiency also, the wheel velocity is half the jet velocity.

Substituting results of Eq. (10.16) in Eq. (10.15), we get

$$(\eta_D)_{max} = 2(1 - k\cos\theta)(1/2 - 1/4)$$

$$= \frac{1 - k\cos\theta}{2} \tag{10.17}$$

Again, the diagram efficiency for maximum work is given by

$$(\eta_D)_{max\ work} = \frac{E_{max}}{V_1^2/2g} = \frac{1 - k\cos\theta}{g}\frac{V_1^2}{4} \times \frac{2g}{V_1^2}$$

$$= \frac{1 - k\cos\theta}{2} = (\eta_D)_{max} = \eta_{max} \tag{10.18}$$

If $k = 1$, i.e. there is no energy loss due to friction, then

$$\eta_{max} = \frac{1 - \cos\theta}{2}$$

If $\theta = 180°$, $\eta_{max} = 1$ or 100%.

However, k lies between 0.8 and 0.85 and $\theta \cong 165°$, so that the exiting water does not hit the following bucket.

$$\eta_{max} = \frac{1 - 0.8\cos 165°}{2} \cong 0.886$$

In practice, $\rho_{opt} \cong 0.46$, instead of 0.5.

If we plot η *vs* ρ, we get Fig. 10.27. Now,

V_b is constant and $V_1 = c_v\,[2gH]^{1/2}$, which depends on net head H. Then the discharge is given by the following equation.

$$Q = A\,c_v\,[2gH]^{1/2}\ \text{m}^3/\text{s} \tag{10.19}$$

where the flow area A is controlled by the spear to regulate Q.

The velocity of wheel V_b is given by $V_b = \phi\,\sqrt{2gH}$

where ϕ = speed ratio, which varies from 0.43 to 0.48.

The minimum number of buckets in the wheel is approximately given by

$$Z = \frac{m}{2} + 15 \tag{10.19 a}$$

where m is equal to the jet ratio, D/d.

The erosion of the Pelton wheel occurs (i) on the buckets due to erosive effect of flow and (ii) at the nozzle due to cavitation effect (discussed later). To protect the buckets from wear and tear, chrome alloy steel or stainless steel is used. In India, many Pelton turbines are in operation such as at Koyna (475 m head, 4 jets), Sharavathi (570 m head, 4 jets), Kundah I (360 m head, 5 jets) and Kundah II (690 m, 3 jets). The world's largest Pelton turbine is a 6-jet, 840 m head turbine at Aurland-2 in Brazil producing 243 MWe.

10.13 DEGREE OF REACTION

By applying Bernoulli's equation to the inlet and outlet of a turbine, we get

$$\frac{p_1}{\rho g} + \frac{V_1^2}{2g} = E + \frac{p_2}{\rho g} + \frac{V_2^2}{2g} \tag{10.20}$$

where E is the energy transferred from fluid to the rotor. Therefore,

$$E = \frac{p_1 - p_2}{\rho g} + \frac{V_1^2 - V_2^2}{2g} \tag{10.21}$$

The first term on the R.H.S. is the energy transfer due to drop in static pressure and the second term represents the energy transfer due to drop in velocity head.

If $p_1 = p_2$, i.e. if pressure is constant,

$$E = \frac{V_1^2 - V_2^2}{2g} \tag{10.22}$$

This happens in the case of impulse turbine, i.e. Pelton wheel where the pressure is atmospheric.

If $V_1 = V_2$,

$$E = \frac{p_1 - p_2}{\rho g} \tag{10.23}$$

This holds good for a pure reaction turbine, where the wheel rotates only due to pressure drop across it exerting a reaction by Newton's third law of motion.

Degree of reaction, R, is defined in the following manner.

$$R = \frac{\text{Energy transfer due to pressure drop}}{\text{Total energy transfer}}$$

$$= \frac{(p_1 - p_2)/\rho g}{E}$$

$$= \frac{E - \dfrac{V_1^2 - V_2^2}{2g}}{E}$$

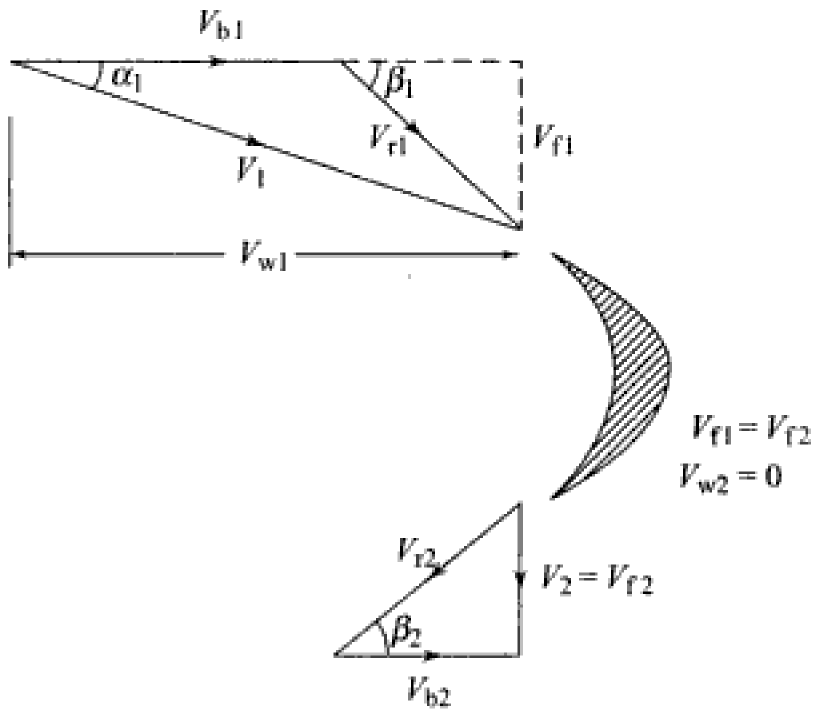


Fig. 10.29 *Velocity triangles of a Francis turbine blade*

Blading or diagram efficiency, $\eta_D = \dfrac{E}{E + E_l} = 1 - \dfrac{E_l}{E + E_l}$

$$= 1 - \frac{1}{1 + 2\cot\alpha_1(\cot\alpha_1 - \cot\beta_1)} \qquad (10.27)$$

Degree of reaction, $$R = 1 - \frac{V_{f_1}^2 \cot^2\alpha_1}{E}$$

Hydraulic efficiency, $$\eta_h = \frac{E}{H} = \frac{V_{w1}V_{b1}}{gH} \qquad (10.28)$$

Overall efficiency, $$\eta_0 = \frac{P}{\rho QgH} \qquad (10.29)$$

where P is the total power output.

10.15 PROPELLER AND KAPLAN TURBINES

The propeller turbine is a reaction turbine used for low heads (4 m – 80 m) and high specific speeds (300 – 1000). It is an axial flow device providing large flow area utilizing a large volume flow of water with low flow velocity. It consists of an axial-flow runner usually with four to six blades of airfoil shape (Fig. 10.30). The spiral casing and guide blades are similar to those in Francis turbines. In propeller turbines as in Francis turbines the runner blades are fixed and non-adjustable.

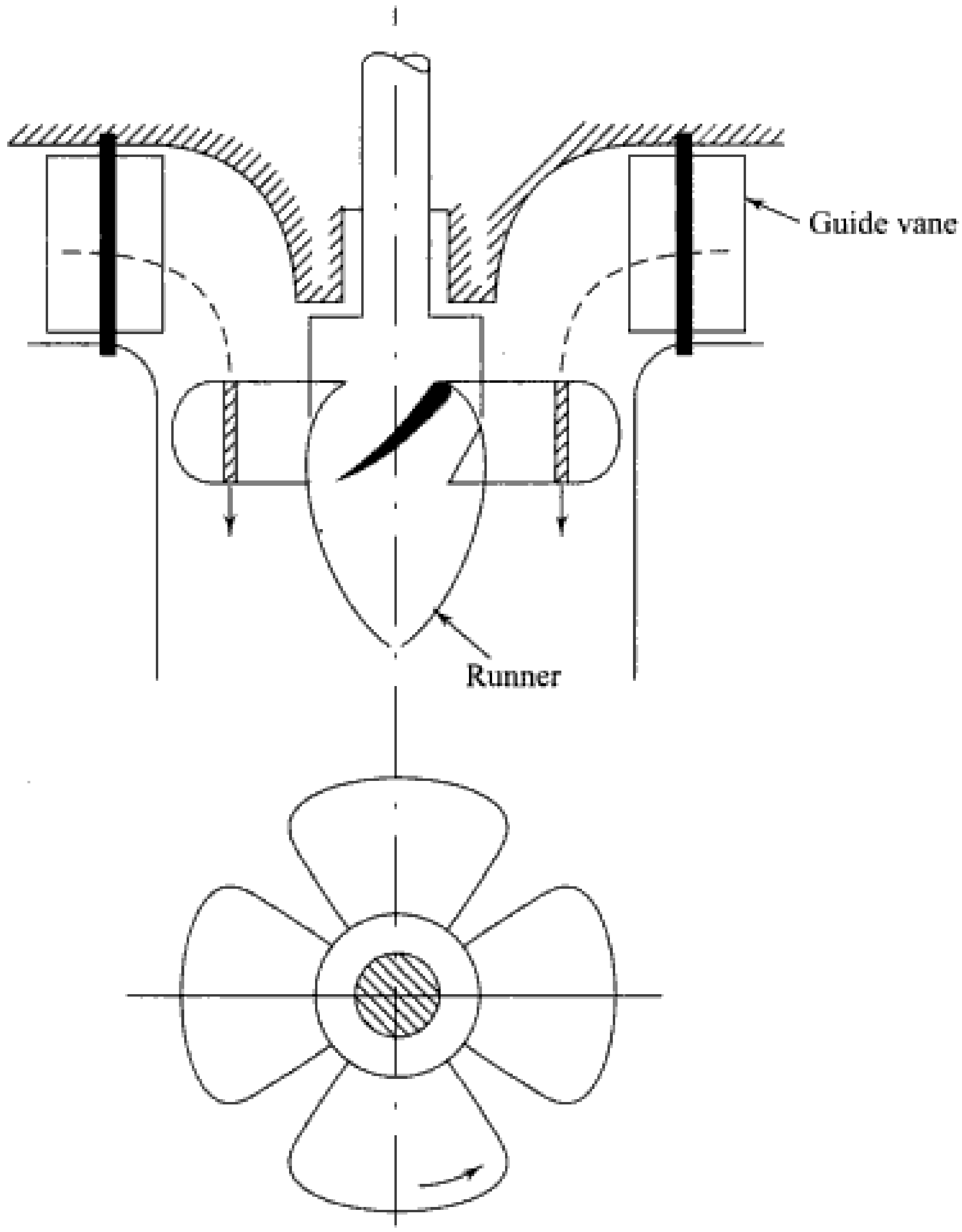


Fig. 10.30 *Schematic view of propeller turbine*

A special type of a propeller turbine is the Kaplan turbine in which the individual runner blades are pivoted on the hub (Fig. 10.31) so that their inclination may be adjusted during operation responding to changes in load. The blades are adjusted automatically rotating about pivots with the help of a governor servo-mechanism. The efficiency of a reaction turbine depends on the inlet blade angle. In fixed blade runners, it is not possible to vary the inlet blade angle for varying demands of power (load). So such turbines are designed for maximum efficiency only for a particular load. At all other loads their efficiency is less than this. In the Kaplan turbine, because of the arrangement for automatic variation of inlet blade angle with variation in load, the turbine can be run at maximum efficiency at all loads.

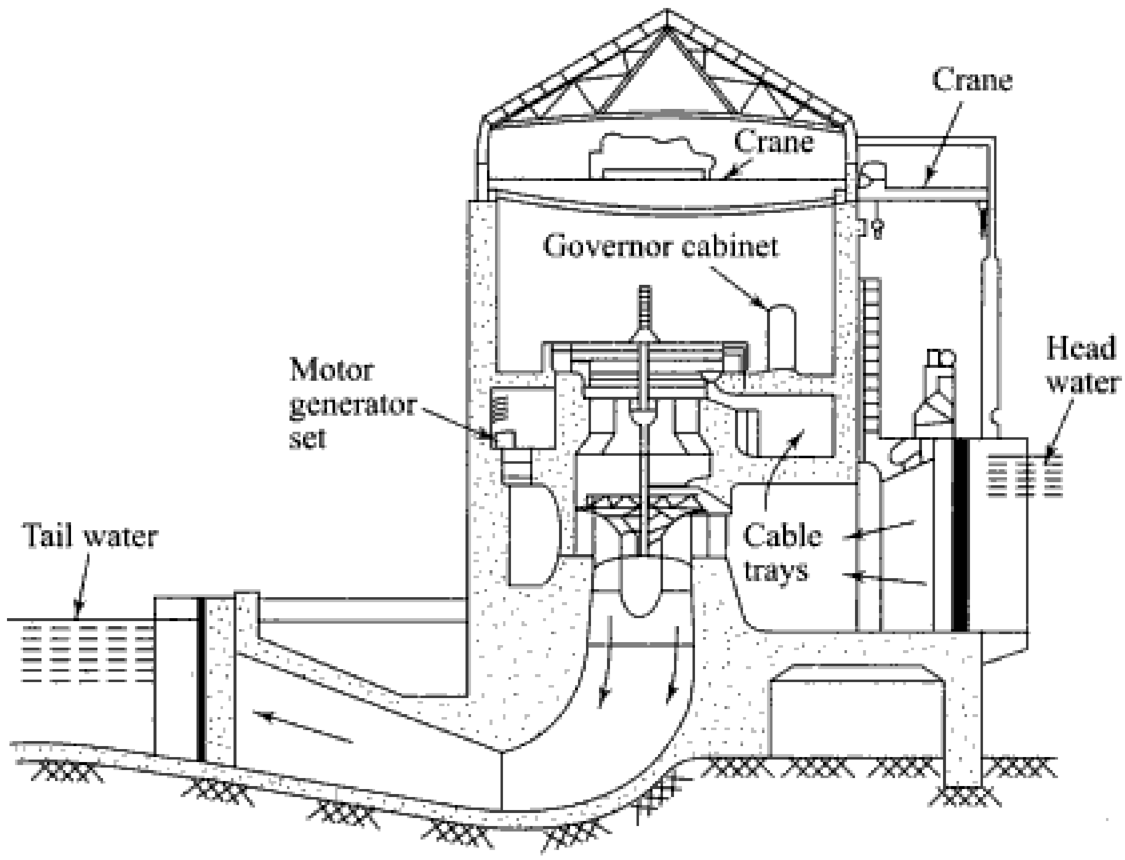


Fig. 10.31 *(c) Cross-section of typical low head concrete spiral-case setting with Kaplan turbine*

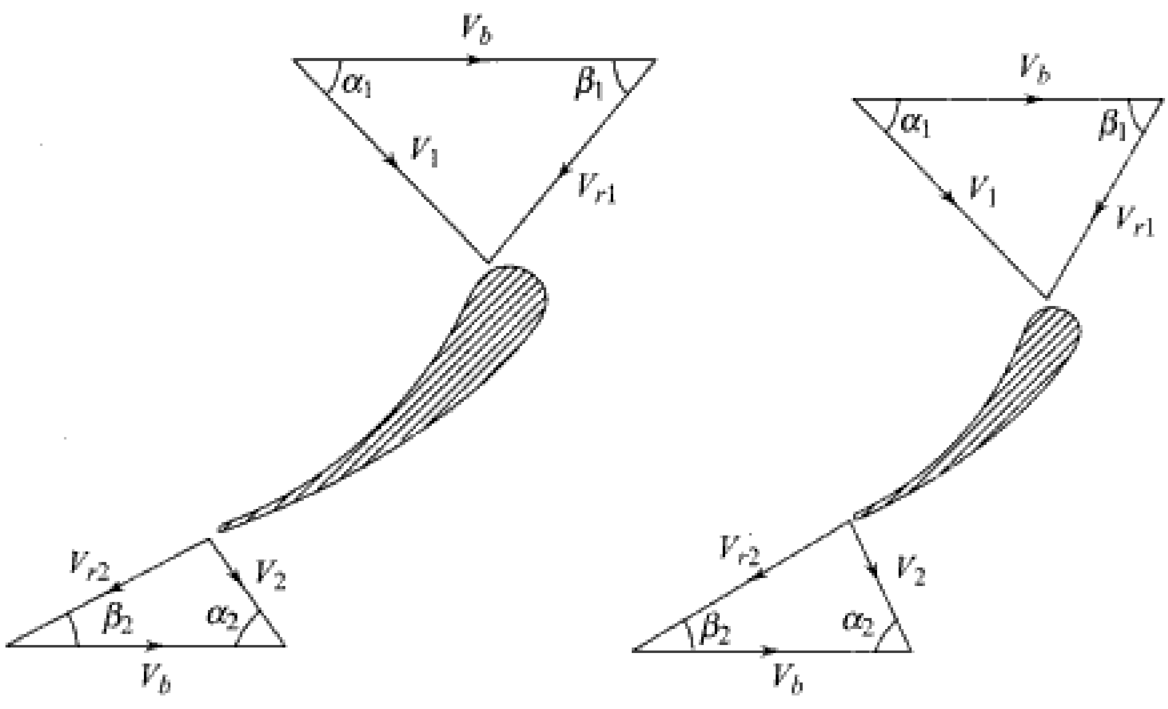


Fig. 10.32 *Velocity triangles for propeller or Kaplan runner blade*

10.16 DERIAZ TURBINE

The Deriaz turbine is also known as the 'diagonal turbine'. The flow over the runner is at an angle of 45° to the axis (Fig. 10.33). It has adjustable blades like

Kaplan turbines. At the same time the flow is diagonal or mixed as in Francis turbines. It can be described as a cross between the two turbines (Kaplan and Francis) and can be used for heads up to 200 m. The number of blades varies from 10 to 12. The guide blades and the stay vanes are also inclined.

The Deriaz runner is particularly suited for reversible flow conditions when the turbine also has to work as a pump as in pumped-storage power plants.

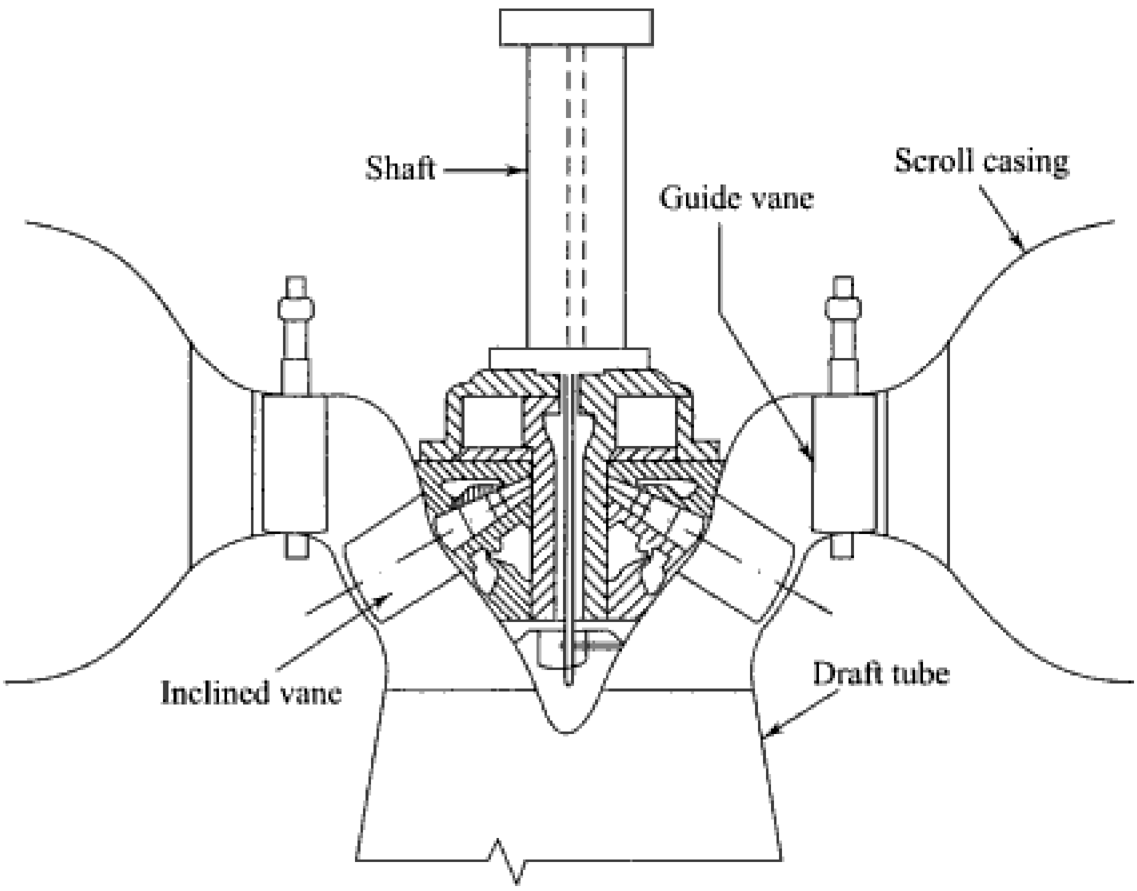


Fig. 10.33 *Deriaz turbine*

10.17 BULB TURBINE

Tubular or bulb turbines are small fixed axial flow propeller turbines operating under low heads. The turbo-generator is housed in an enclosed bulb-shaped casing, which is installed right in the middle of the flow passage. The bulb and the propeller form an integral unit followed by a straight conical flaring draft tube (Fig. 10.34). Bulb turbines are suitable for tidal power plants.

10.18 SPECIFIC SPEED

To analyse hydroelectric schemes it is economical to make a scale model and perform necessary hydraulic tests on it in order to predict what will happen in the prototype or full-sized system under similar operating conditions. The suitability of a turbine for a particular application depends on (a) head of water (b) rotational speed (c) power developed, which together fix a parameter called 'specific speed'.

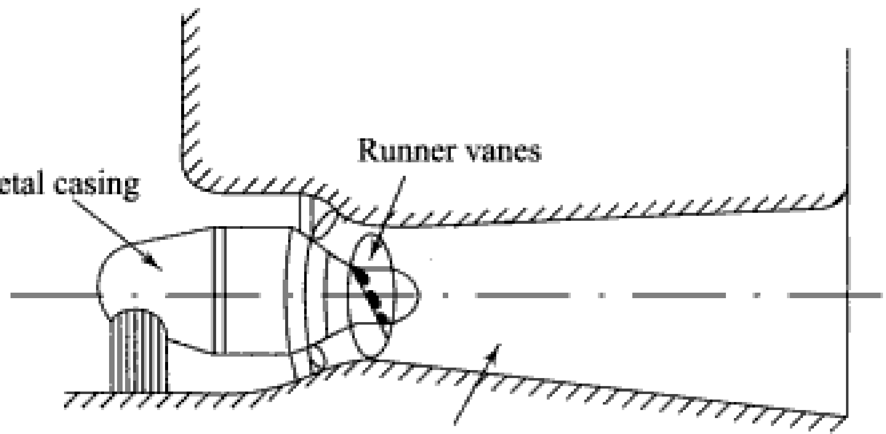


Fig. 10.34 *Bulb turbine*

The specific speed of a turbine is defined as the speed of operation of a *geometrically similar model* of the turbine which is so proportioned that it produces 1 kW *power* when operating under 1m *head*.

We know,

$$\text{Power, } P = \rho QgH$$

Therefore, $P \propto QH$, since the density of water and acceleration due to gravity g are constant,

or $$P \propto (AV)\, H \tag{10.30}$$

where A is the cross-sectional flow area and V is the water velocity.

or $$P \propto D^2\, [2gH]^{1/2}\, H$$

or $$P \propto D^2 \times H^{3/2} \tag{10.31}$$

where D is the wheel diameter through which the water flows axially and H is the net head of water. Again, the blade velocity is

$$V_b \propto V \tag{10.32}$$

or $$DN \propto [2gH]^{1/2}$$

or $$D \propto \frac{H^{1/2}}{N} \tag{10.33}$$

Substituting Eq. (10.33) in Eq. (10.31), we get

$$P \propto \frac{H}{N^2} \cdot H^{3/2}$$

$\therefore$ $$N^2 \propto \frac{H^{5/2}}{P}$$

$\therefore$ $$N \propto \frac{H^{5/4}}{P^{1/2}} \tag{10.34}$$

or $$N = \frac{KH^{5/4}}{P^{1/2}}$$

or $$K = \frac{NP^{1/2}}{H^{5/4}}$$

If $P = 1$ kW, $H = 1$ m, then K is equal to N, called the specific speed, N_s.

$$\therefore \quad N_s = \frac{NP^{1/2}}{H^{5/4}} \tag{10.35}$$

The classification of turbines on the basis of specific speeds has been discussed in Section 10.10. The ranges of specific speeds of different turbines are given in Table 10.2. The non-dimensional form of specific speed is given by the following equation which is Eq. (10.4) derived earlier.

$$N_s' = \frac{NP^{1/2}}{\rho^{1/2}(gH)^{5/4}}$$

It is also known as the shape number of the turbine.

10.18.1 Scale Ratio

The model of a turbine and its prototype are in definite geometric ratio depending on their respective heads and the rotative speeds. The ratio of blade velocity V_b and the water velocity V is called the speed ratio, which has a definite value for a particular turbine (0.42 to 0.47 for a Pelton turbine, from 0.55 to 1.00 or more for a Francis turbine and 1.5 to 3.00 or more for a propeller turbine).

$$\therefore \quad V_b \propto V$$

$$\therefore \quad DN \propto \sqrt{H}$$

Using subscript m for the model turbine and p for the prototype, we get

$$\frac{D_m N_m}{D_p N_p} = \sqrt{\frac{H_m}{H_p}}$$

$$\frac{D_m}{D_p} = \sqrt{\frac{H_m}{H_p}}\frac{N_p}{N_m} \tag{10.36}$$

This is called the *scale ratio* which represents the ratio of the diameters of the model turbine and the prototype turbine.

10.18.2 Unit Speed, Unit Power and Unit Discharge

The terms 'unit speed', 'unit power' and 'unit discharge' are frequently used to express the operational characteristics of hydraulic turbines.

The unit speed N_u is defined as the speed of a geometrically similar turbine working under a head of 1 m,

$$V_b \propto V$$

$$DN \propto [2gH]^{1/2}$$

$$N \propto \sqrt{H}$$

$\therefore$ $$N = K\sqrt{H}$$

where K is a constant.

When $H = 1$ m, $N = N_u = K$

$\therefore$ $$N_u = \frac{N}{\sqrt{H}} \quad (10.37)$$

The *unit power* P_u is the kW of power generated by a geometrically similar turbine working under a head of 1 m.

$$P = \rho QgH = \rho(AV)gH$$

$$= \rho A[2gH]^{1/2} \cdot gH$$

or $$P \propto H^{3/2}$$

When $H = 1$ m, $P = P_u = K$

$\therefore$ $$P_u = \frac{P}{H^{3/2}} \quad (10.38)$$

The unit discharge, Q_u is the flow rate the turbine would have under a head of 1 m.

$$Q = AV = A\,[2gH]^{1/2}$$

$$Q = K\sqrt{H}$$

where K is a constant.

When $H = 1$ m, $Q = Q_u = K$

$\therefore$ $$Q_u = \frac{Q}{H^{1/2}} \quad (10.39)$$

10.19 COMPARISON OF TURBINES

The characteristic features of common types of turbine are summarized in Table 10.4.

10.20 CAVITATION

When the velocity of a fluid increases its pressure falls. In any turbine part if the pressure drops below the vapour pressure at that temperature some of the liquid flashes into vapour. The bubbles formed during vaporization are carried by the water stream to higher pressure zones, where the bubbles condense into liquid forming a cavity or vacuum. The surrounding liquid rushes towards the cavity giving rise to a very high local pressure which may be as high as 7000 atm. The formation of such a cavity and high pressure occurs repeatedly hundreds of times in a second. This phenomenon is known as cavitation, which causes pitting on the metallic surface of runner blades and draft tube. It is accompanied by considerable vibration and noise.

Table 10.4 *Comparison of common turbines*

		Pelton wheel	*Francis turbine*	*Kaplan/ Propeller turbine*
1.	Flow	Tangential, single stage, impulse	Inward radial flow, single stage, reaction	Axial flow, single stage, reaction
2.	Maximum capacity	250 MW	720 MW	225 MW
3.	Number of jets/kind of blades	1 to 6 Maximum 2 for horizontal and 6 for vertical shaft	Fixed blades	Propeller turbines have fixed blades, while Kaplan turbines have adjustable blades
4.	Head	100–1750 m	30–550 m	1.3–77.5 m
5.	RPM	75–1000	93.8–1000	72–600
6.	Hydraulic efficiency	Single jet 85–90%	90–94%	85–93%
7.	Specific speed	6–60	50–400	280–1100
8.	Regulation mechanism	Spear nozzle and deflector plate	Guide vanes	Blade stagger

This table can be used in selecting a turbine for a specific application.

Cavitation should be minimised or avoided by selecting proper material like stainless steel or alloy steel, by adequate polishing of the surface, by selecting a runner of low specific speed or by keeping the runner under water.

10.21 GOVERNING OF HYDRAULIC TURBINES

Hydraulic turbines are directly coupled to the electric generators. The generators are always required to run at a constant speed irrespective of the variations in the load. This constant speed (rpm) of the generator is given by

$$N = \frac{120f}{p} \tag{10.40}$$

where f is the frequency for power generated in cycles per second and p is the number of poles for the generator. The speed of the generator can be maintained at a constant level only if the speed of the turbine runner is constant as given by Eq. (10.40). It is known as the synchronous speed of the turbine runner for which it is designed.

If the load on the generator goes on varying and if the input for the turbine remains the same, then the speed of the runner tends to increase if the load goes down or it tends to decrease if the load on the generator goes up. Therefore, the speed of the generator and hence, the frequency will vary accordingly, which is not desired. Therefore, the speed of the runner is always required to be maintained at a constant level at all loads.

It is done automatically by a governor which regulates the quantity of water flowing through the runner in proportion to the load.

10.21.1 Governing of Impulse Turbine

In a Pelton turbine, water flow to the runner is regulated by the combined action of the spear and the deflector plate. There is a centrifugal governor, as in the case of a steam turbine, where its sensitivity to load variation is augmented by an oil-operated servo-mechanism (Fig. 10.35). When the load on the generator drops, the speed of turbine runner increases. The flyballs of the centrifugal governor fly outward due to more centrifugal force (due to higher rpm). The sleeve moves up, the portion of the lever to the right of the fulcrum moves down pushing the piston rod of the control valve downwards.

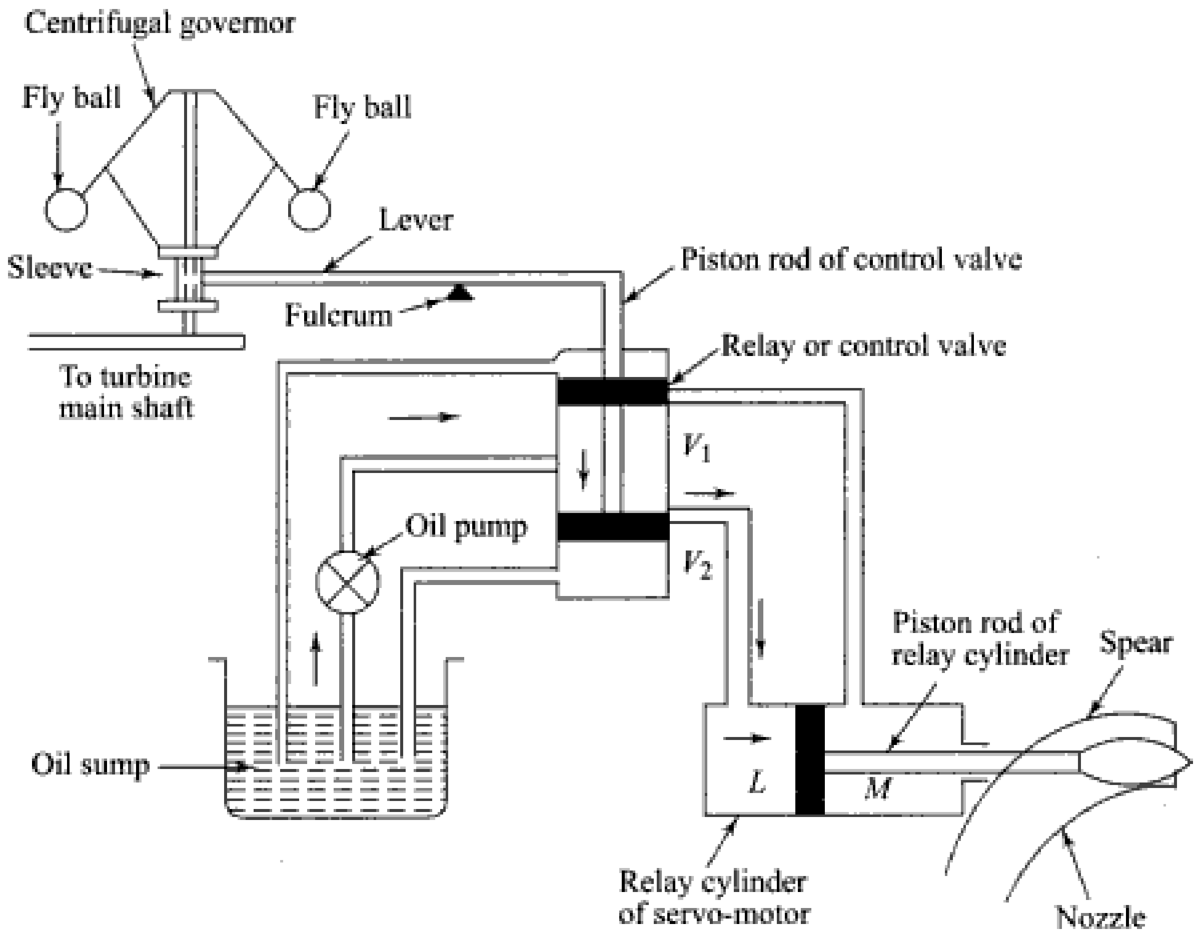


Fig. 10.35 *Governing of Pelton turbine*

With the downward motion of the piston rod, valve V_1 closes and valve V_2 opens as shown in Fig. 10.35. A gear pump pumps oil from the oil sump to the control or relay valve. Oil flows through valve V_2 and exerts force on the face L of the piston of the relay cylinder. The piston (or spear) rod along with the spear moves to the right, thus decreasing the flow area and hence, the rate of water flow to the turbine. The speed of the turbine falls till it becomes normal when the flyballs, sleeve, lever, etc. also come to normal position. The reverse happens when the load on the generator increases, speed decreases, flyballs fly inward with less centrifugal force (due to less rpm), the sleeve moves down, the piston rod of control valve goes up, valve V_1 opens and valve V_2 closes, the oil under pressure flows through valve V_1 and exerts a force on the face M of the piston. The piston rod and the spear move to the left as a result of which more water flows to the turbine to take up more load and the speed becomes normal, i.e. attains its rated value.

The spear or needle valve is used normally for small load fluctuations. When there is a sudden fall of load, the spear has to move rapidly to close the nozzle. This rapid closing may cause water hammer. It is quite serious in large capacity plants with long penstocks. To avoid the water hammer effects during a sudden fall of load, a deflector is introduced in the system, which is not shown in Fig. 10.35. The function of the deflector is to deflect some water from the jet advancing to the turbine runner when the load on the turbine suddenly decreases. The quantity of water flowing through the nozzle remains the same, but a certain part of water coming out from the nozzle is deflected and is not allowed to strike the buckets. The deflected water goes waste into the tailrace level.

10.22 GOVERNING OF REACTION TURBINES

The governing of Francis turbine is similar to the governing of Pelton wheel except that the motion of the piston in servo-motor is used to partially close or open the guide vanes gate through which the water is supplied to the turbine (instead of the spear in the nozzle of the Pelton turbine). The working diagram of the governor is shown in Fig. 10.36. The position of the control valve and the servo-motor correspond to the design load on the turbine and operate in the same way as in the case of Pelton wheel. A compensating device is, however, added to prevent the governor from overshooting. When the servo-motor piston moves to the right, the bell-crank lever EFG is rotated downward about F and the arm G is lowered. This pulls down the pivot A, which, in turn, lowers the fulcrum B. Thus the relay port 'b' is partially or fully closed, restricting the piston motion to the right.

The governor is always operated with a pressure relief valve (not shown). A sudden closure of wicket gates will open the relief valve due to the sudden increase in pressure and protect the conduit from inertia effects of speeding water. The relief valve consists of a spear and is held by fluid pressure to close the bypass of water from the spiral casing to the tailrace at design load. When the load decreases suddenly, a bell-crank lever opens the pilot valve of the pressure chamber so that the pressure on the spear is reduced, thereby permitting the spear to be lifted up and allowing a portion of water to flow directly from the spiral casing to the tailrace though the bypass without striking the turbine runner. Thus, both the deflector of Pelton wheel and the relief valve of Francis or Kaplan turbine perform the same function of protecting the system from water hammer effects when the load suddenly decreases.

In the case of Kaplan turbine, in addition to guide vanes the runner vanes are also adjustable and hence the governor is required to operate both sets of vanes simultaneously. The runner vanes are also operated by a separate servo-motor and a control valve which are interconnected with those of the guide vanes to ensure that for a given guide vane opening there is a definite runner vane inclination.

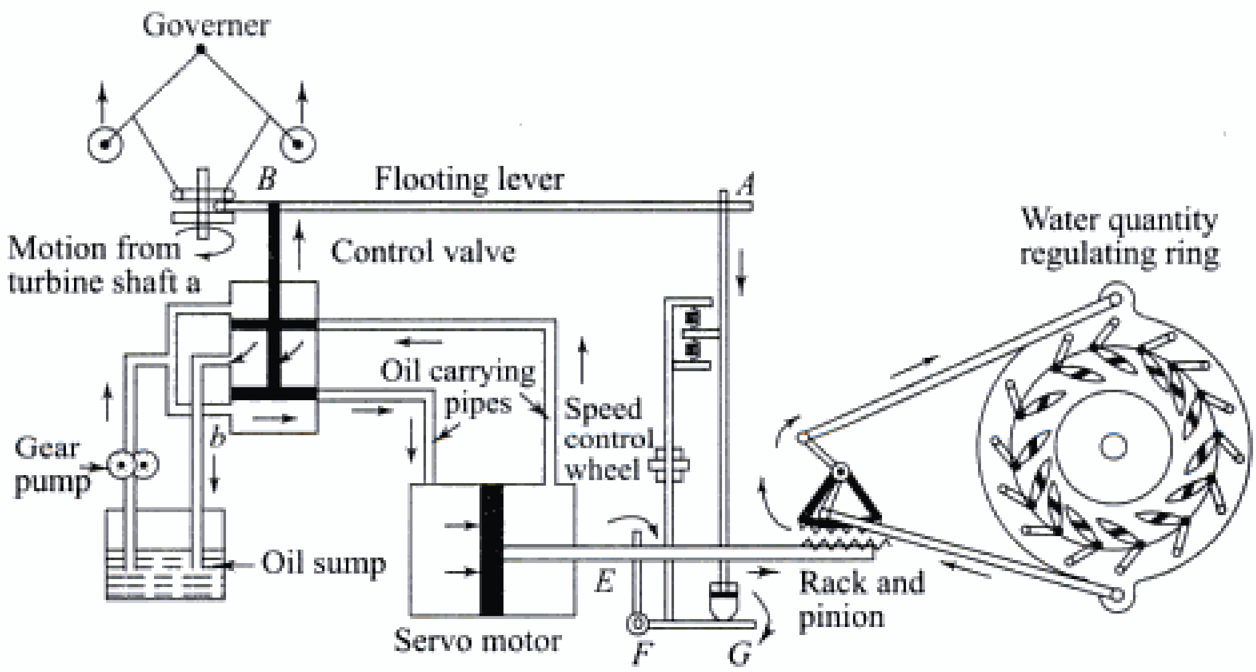


Fig. 10.36 *Governing of Francis turbine*

10.23 SURGE TANKS

A preliminary discourse on surge tanks was made in Section 10.8.6. When the load on the generator decreases the governor reduces the rate of flow of water striking the runner in order to maintain the constant speed of the runner. But the sudden reduction of the rate of flow in the penstock may build a water hammer in the pipe, which may cause excessive inertia pressure in the pipeline due to which the pipe may burst. Two devices, viz. the deflector and the relief valve, as described earlier, are provided to avoid the sudden reduction of the rate of flow in the penstock. But neither of these devices is of any help when the load on the generator increases and the turbine is in need of more water. Thus, in order to fulfil both the above objectives, in addition to the deflector or the relief valve, certain other devices such as *surge tank* and *forebay* are provided. Surge tanks are employed in the case of high head and medium head power plants where the penstock is very long and forebays are suitable for medium head and low head power plants where the length of the penstock is short.

An ordinary surge tank is a cylindrical open-topped storage reservoir, as shown in Fig. 10.37, which is connected to the penstock at a point as close as possible to the turbine. The upper lip of the tank is kept well above the maximum water level in the supply reservoir.

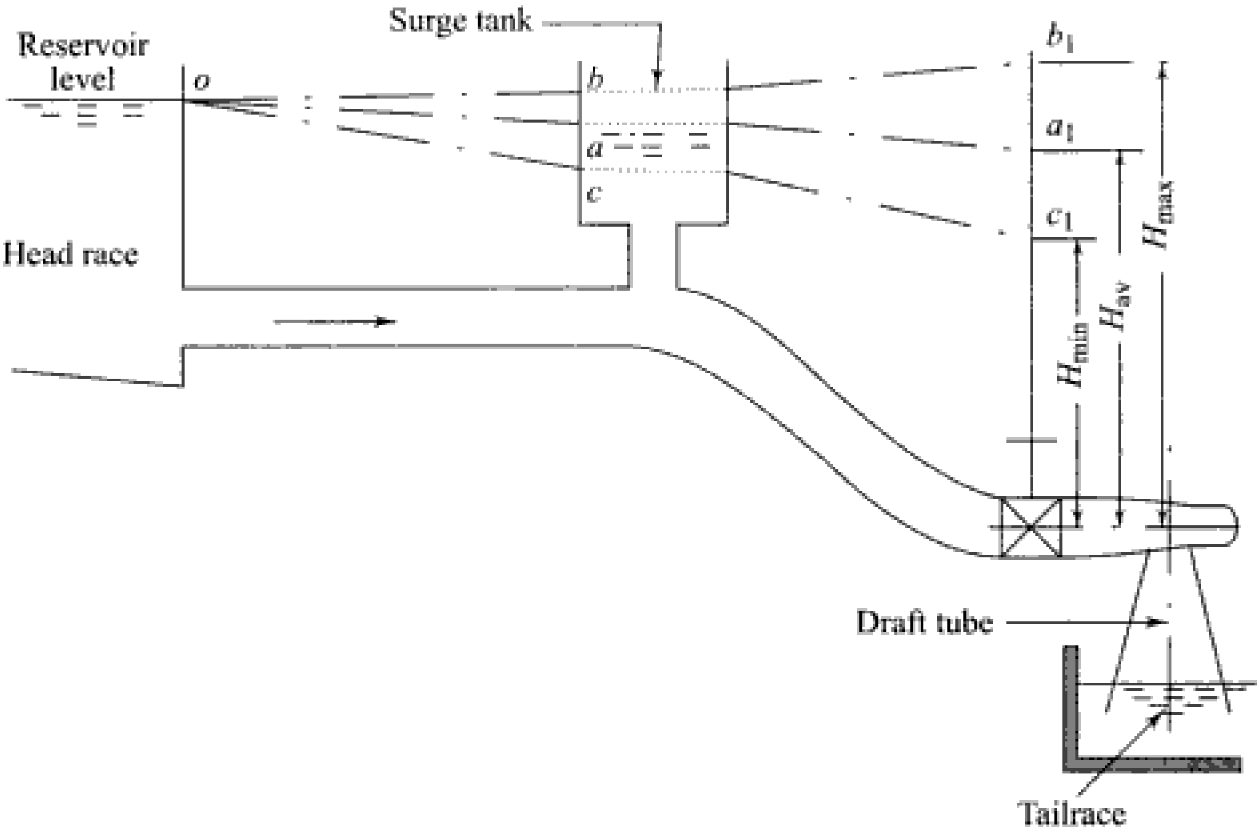
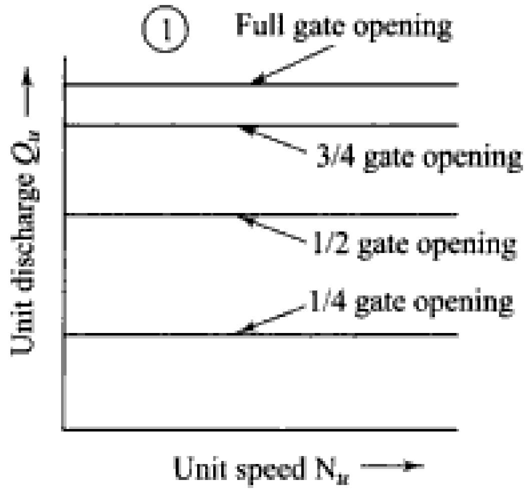


Fig. 10.37 *Ordinary surge tank*

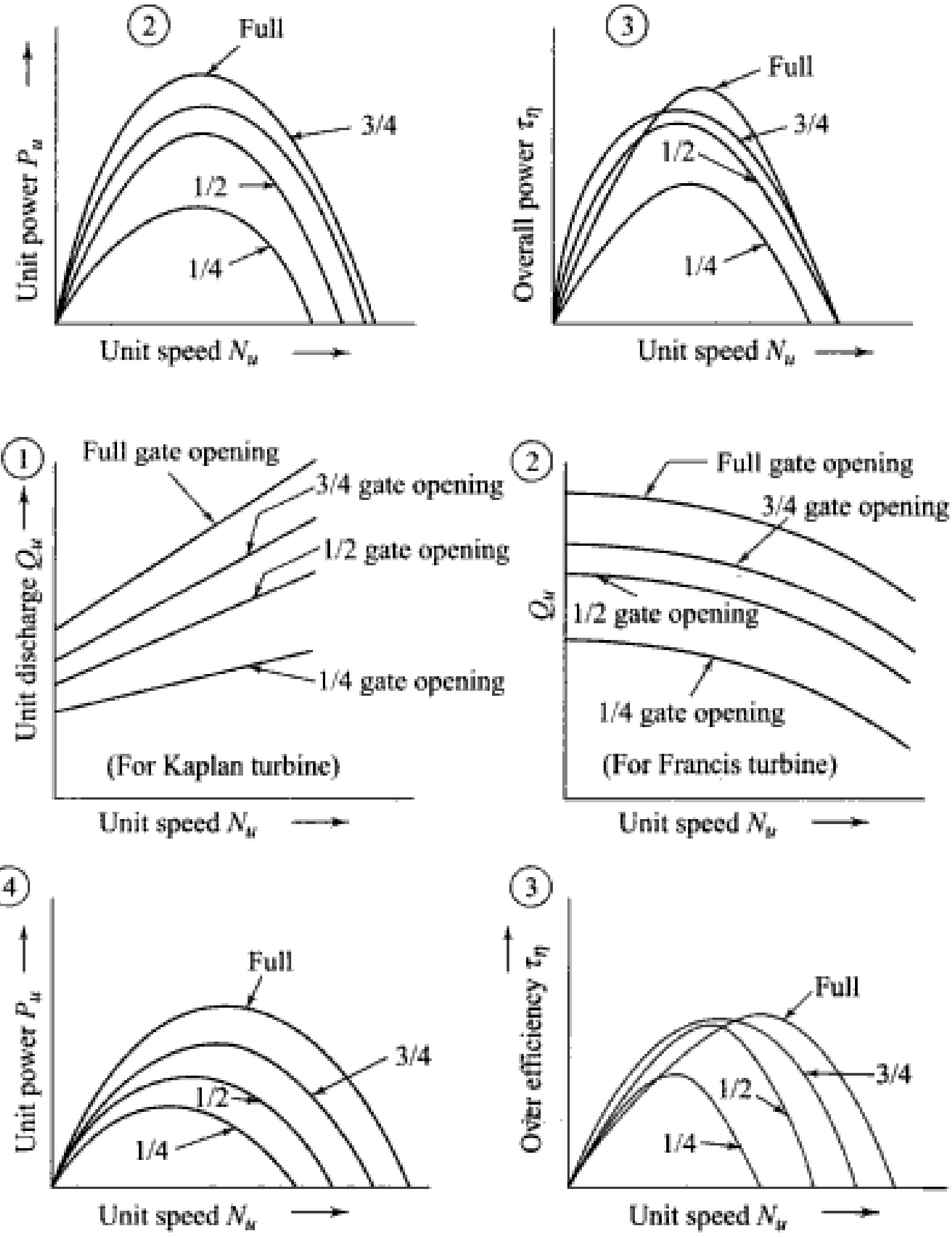
When the load on the turbine is normal and steady, there are no velocity variations in the pipeline and the normal pressure gradient is oaa$_1$ (Fig. 10.37). The water surface in the surge tank is lower than the reservoir surface by an amount equal to the friction head loss in the pipe connecting the reservoir and the surge tank. When the load on the generator decreases, the turbine gates are partially closed and the excess water moving towards the turbine is stored in the surge tank in the space between the levels *a* and *b* and a rising pressure gradient abb$_1$ develops. The resulting retarding head reduces the velocity of flow in the pipeline corresponding to the reduced discharge required by the turbine.

When the load on the generator increases, the governor opens the turbine gates to increase the rate of flow entering the runner. The increased demand of water by the turbine is partly met by the water stored between levels *a* and *c* in the surge tank (Fig. 10.37). As such the water level in the surge tank falls and a falling pressure gradient Occ$_1$ is developed. The surge tank thus provides an accelerating head which increases the velocity of flow in the pipeline corresponding to the increased demand by the turbine.

Various other types of surge tanks are also shown in Fig. 10.38. Type (a) is a conical type surge tank. Type (b) has an internal bell-mouth spillway which permits the overflow to be easily disposed of Type (c) is a differential surge tank, which has a central riser pipe having small ports at its lower end. It



(a) For Pelton wheel



(b) For reaction turbine

Fig. 10.39 *Constant head characteristics of Pelton wheel and reaction turbines*

(b) Constant speed characteristic curves In these tests the constant speed is attained by regulating the gate opening (i.e. discharge) as the load varies. The head may or may not remain constant. The characteristic curves of efficiency against load for different turbines are shown in Figs 10.40 (a) and (b). The efficiency increases with load and reaches the maximum at the full or rated load. It is observed that the Kaplan turbine and the Pelton wheel maintain a high efficiency over a longer range of part load as compared with either the Francis or the fixed blade propeller turbine.

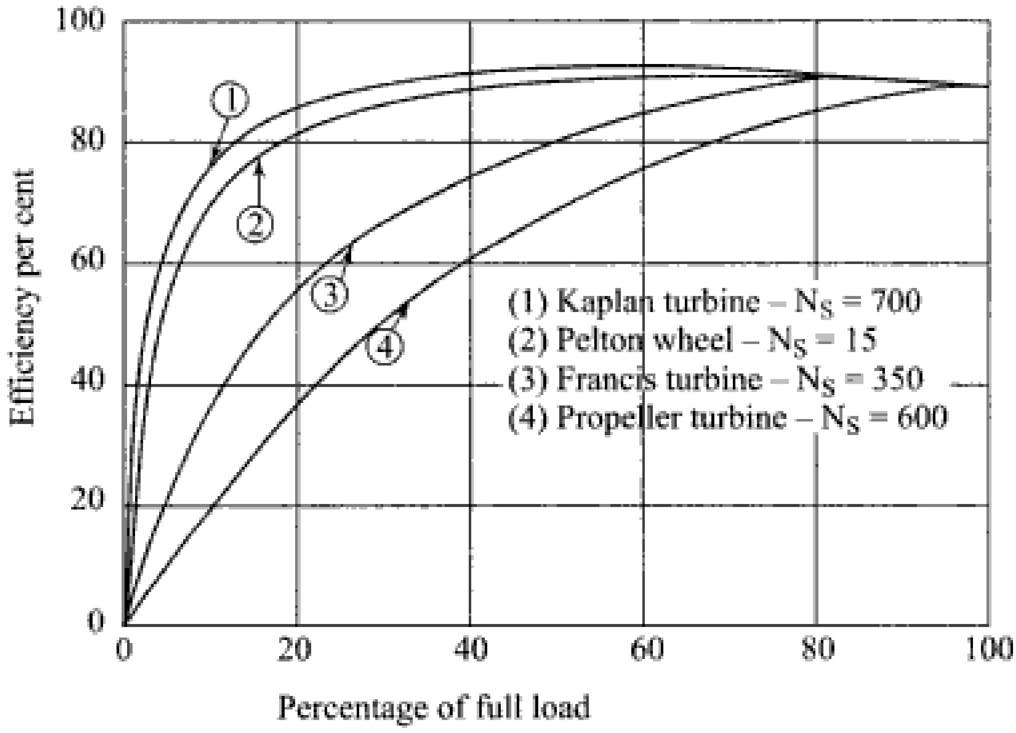


Fig. 10.40 *(a) Overall efficiency variation with load for various turbines*

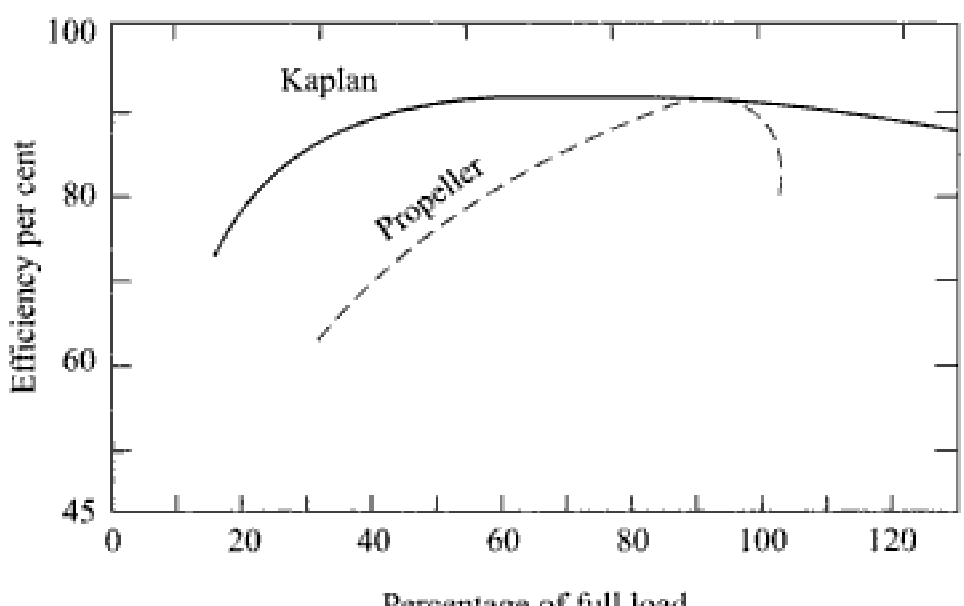


Fig. 10.40 *(b) Efficiency variation with load for propeller and Kaplan turbines*

Figure 10.41 shows the plots of efficiency and power varying with discharge, where Q_0 is the maximum discharge required to initiate the motion of the turbine runner from the state of rest. Since the power ($P = \rho QgH$) is directly proportional to discharge if the head is constant, the P vs Q plot is a straight line. However, the overall efficiency increases with discharge and becomes more or less constant beyond a certain value of discharge.

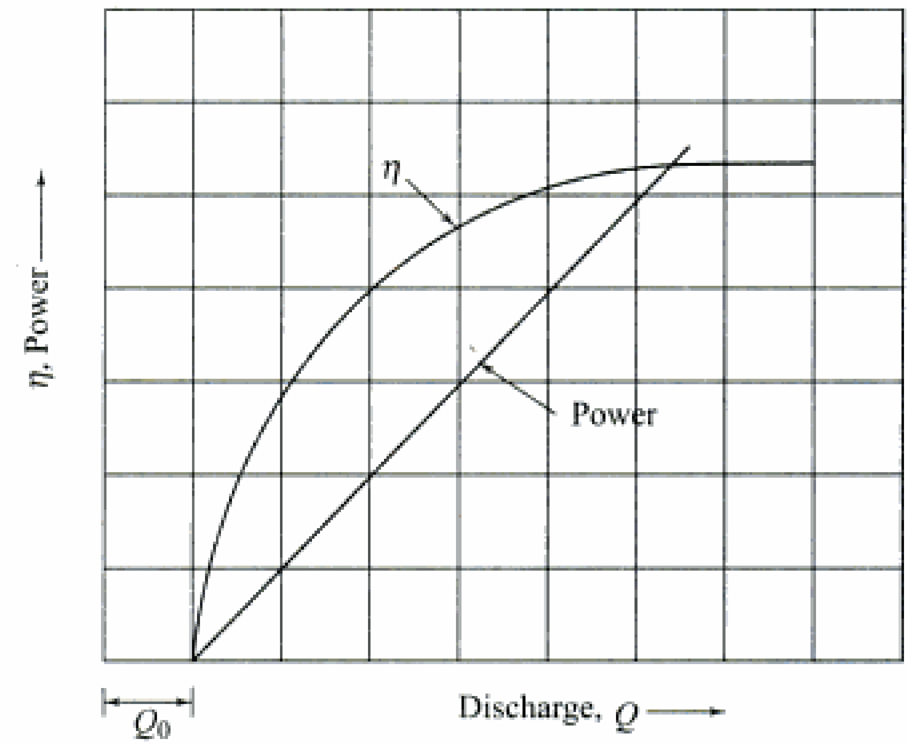


Fig. 10.41 *Overall efficiency and brake power varying with discharge*

(c) *Constant efficiency curves* Figure 10.42 shows the constant efficiency curves for all conditions of running, which are also called the universal characteristic curves of the turbine. The inner most curve represents the highest efficiency of the turbine and the outer curves represent lower efficiencies. If a vertical line is drawn at a certain Q_u, it will intersect an efficiency curve at two points and it will also touch some other efficiency curve of higher η at one point. Thus, for a unit discharge (or power) the vertical line touches the curve of maximum efficiency at only one point. Now, if these points are joined together by a smooth curve, we obtain the best performance curve for the turbine. By drawing a horizontal line for a given N_u (at certain H and N) which cuts this best performance curve, the point of maximum efficiency is known, corresponding to which Q_u or P_u can be obtained and hence Q and P can be estimated at which the turbine efficiency is maximum for the given H and N.

There is a term called '*runway speed*' which is the maximum speed of the turbine under no load and no governing action. The hydraulic design is for optimum speed, but it must also satisfy structurally the safety conditions at runaway speed, it is about 1.8 to 2.3 times the optimum speed.

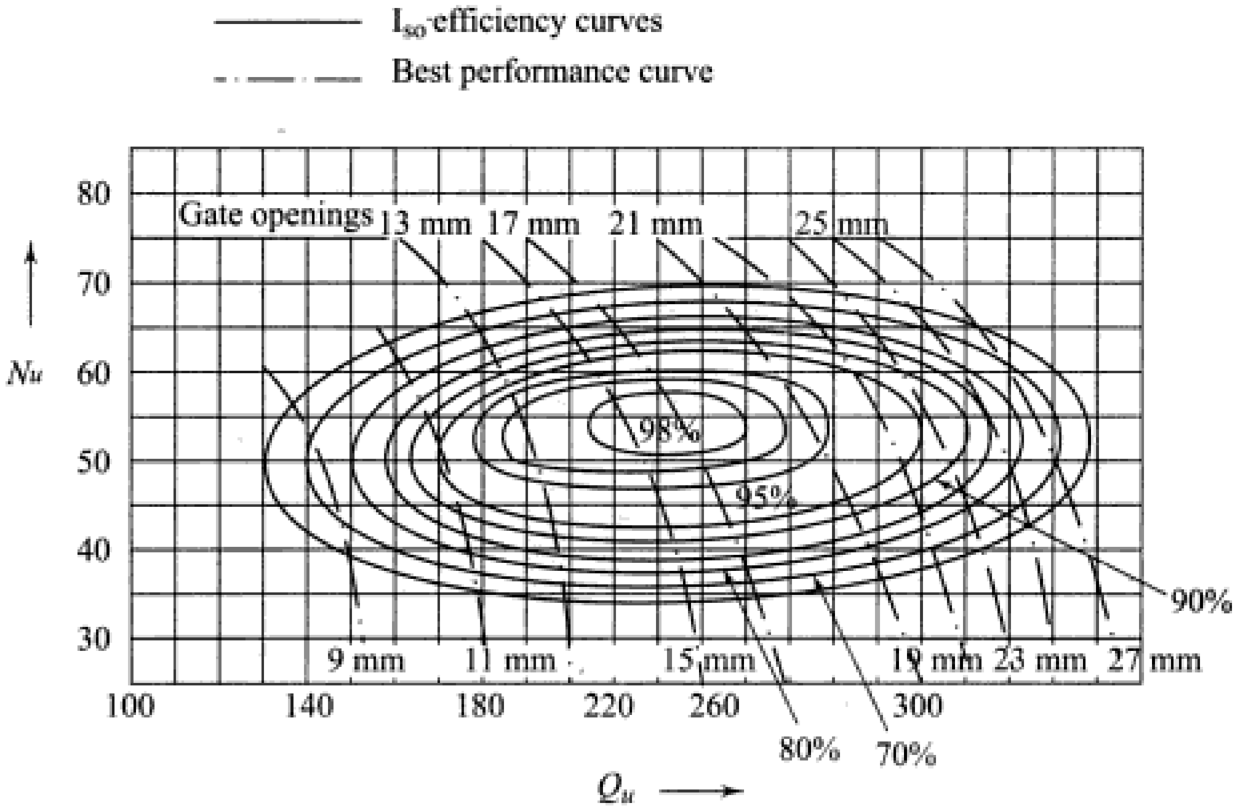


Fig. 10.42 *Constant efficiency and best performance curves (universal characteristics) of a Francis turbine*

10.25 SELECTION OF TURBINES

The hydraulic turbine is selected according to the specific conditions under which it has to operate and attain the maximum possible efficiency. The choice depends on the head available, power to be developed and the speed at which it has to run. The following factors basically govern the selection of a suitable type of turbine.

(a) *Operating head* The present practice is to use Kaplan and Propeller type of turbines for heads up to 50 m. For head from 50 to 400 m, Francis turbines are used. For heads greater than 400 m, impulse or Pelton turbines are used. The range of heads as mentioned is not rigid and may change if other conditions dominate to achieve economy.

(b) *Specific speed* It is better to choose turbines of high specific speeds. High speed turbines mean small sizes of turbines, generators, power house, etc. and are therefore, more economical. The range of specific speeds of the turbines should correspond to the synchronous speed of the generator, $N = \frac{120f}{p}$, where f is the frequency and p the number of poles.

(c) *Height of installation* It is better to install the turbines as high above the tail water level (TWL) as possible. This saves the cost of excavation for the draft tube. Care should be taken to ensure that cavitation does not occur.

(d) *Performance characteristics of turbine* The performance characteristics of turbines as discussed in Section 10.24 should be studied carefully before recommending the type of turbine to be used. A turbine has the maximum efficiency at a certain load. When a turbine has to operate mostly at part loads, only those turbines whose efficiencies do not fall appreciably with part loads should be selected. Kaplan and Pelton turbines are better than Francis and propeller turbines in this respect.

(e) *Size of turbine* It is better to go in for as large a size of turbine as possible since this results in economy of size of the power house, the number of penstocks, the generator, etc. Bigger size means less number of runners. However, the number of runners should not be less than two so that at least one unit is always available for service in the case of a plant breakdown.

Example 10.1 A Pelton wheel driven by two similar jets transmits 4000 kW to the shaft when running at 400 rpm. The head from the reservoir level to the nozzle is 200 m and the efficiency of power transmission through the pipelines and nozzles is 90 per cent. The jets are tangential to a 1.50 m diameter circle. The relative velocity decreases by 10 per cent as the water traverses the buckets, which are so shaped that they would, if stationary, deflect the jet by 165°. Neglecting windage losses, estimate (a) the efficiency of the runner and (b) the diameter of each jet.

Solution

Velocity of fluid at inlet to the bucket is,

$$V_1 = \sqrt{2gH.k_v} = \sqrt{2\times 9.81\times 200\times 0.9} = 59.43 \text{ m/s}$$

$$V_b = \frac{\pi DN}{60} = \frac{\pi\times 1.5\times 400}{60} = 31.42 \text{ m/s}$$

$$\eta = \frac{(1-k\cos\theta)(V_1-V_b)V_b}{g\times V_1^2/2g} = \frac{2(1-k\cos\theta)(V_1-V_b)V_b}{V_1^2}$$

$$= \frac{2(1-0.9\cos 165°)(59.43-31.42)\,31.42}{(59.43)^2}$$

$$= 0.9312 \text{ or } 93.12\%$$ *Ans.* (a)

Power developed, is

$$P = \frac{4000}{0.9312} = 4295.53 \text{ kW}$$

Power developed per jet is,

$$= \frac{4295.53}{2} = 2147.77 \text{ kW}$$

$$\therefore \quad 2147.77 = (\rho A_1 V_1)\frac{V_1^2}{2} = \rho\frac{\pi d^2}{4}\frac{V_1^3}{2}$$

$$= 1000 \times \frac{\pi}{8} d^2 \times (59.43)^3 \times 10^{-3}$$

$$\therefore \quad d = \sqrt{\frac{2147.77 \times 8}{\pi \times (59.43)^3}} = 0.1614 \text{ m}$$

= diameter of each jet *Ans.* (b)

Example 10.2 A Pelton wheel has to be designed for the following specifications. Power to be developed = 6000 kW. Net head available = 300 m. Speed = 550 rpm. Ratio of jet diameter to wheel diameter = 1/10. Hydraulic efficiency = 0.85. Assuming the velocity coefficient C_v = 0.98 and speed ratio f = 0.46, find (a) the number of jets (b) diameter of each jet (c) diameter of the wheel and (d) the quantity of water required.

Solution

$$V_1 = C_v \sqrt{2gH} = 0.98 \sqrt{2 \times 9.81 \times 300}$$

$$= 75.19 \text{ m/s}$$

$$V_b = 0.46\sqrt{2 \times 9.81 \times 300} = 35.29 \text{ m/s}$$

$$\eta_0 = \frac{P}{\rho QgH} = \frac{6000 \times 10^3}{1000 \times Q \times 9.81 \times 300} = 0.85$$

$$\therefore \quad Q = \frac{20}{9.81 \times 0.85} = 2.4 \text{ m}^3/\text{s}$$ *Ans.* (d)

$$V_b = 35.29 \text{ m/s} = \frac{\pi D \times 550}{60}$$

D = diameter of the wheel

$$= \frac{35.29 \times 6}{\pi \times 55} = 1.23 \text{ m}$$ *Ans.* (c)

$$\frac{d}{D} = \frac{1}{10}$$

$\therefore$ d = diameter of each jet = 0.123 m *Ans.* (b)

$$\text{Number of jet's} = \frac{Q}{V_1 \times \frac{\pi}{4} d^2}$$

$$= \frac{2.4 \times 4}{75.19 \times \pi \times (0.12)^2}$$

= 2.822, i.e. 3 jets *Ans.* (a)

Example 10.3 A single jet impulse turbine of 10 MW capacity is to work under a head of 500 m. If the specific speed of the turbine is 10, the overall efficiency is 80 per cent and the coefficient of velocity is 0.98, find the diameters of the jet and the bucket wheel. Assume the speed of the bucket wheel as 0.46 of the velocity of jet.

Solution

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$\therefore \quad N = \frac{N_s H^{5/4}}{\sqrt{P}} = \frac{10 \times (500)^{5/4}}{\sqrt{10.000}}$$

$$= 236.4 \text{ rpm}$$

Velocity of jet, $V = C_v\sqrt{2gH}$

$$= 0.98\sqrt{2 \times 9.81 \times 500}$$

$$= 97.06 \text{ m/s}$$

Speed of bucket wheel, $V_b = 0.46 \times 97.06$

$$= 44.65 \text{ m/s}$$

$$V_b = \frac{\pi DN}{60} = 44.65$$

$$\therefore \quad D = \frac{60 \times 44.65}{\pi \times 236.4} = 3.61 \text{ m} \qquad \textit{Ans.}$$

$$\eta_0 = \frac{P}{\rho QgH} = \frac{P}{\rho \times \frac{\pi}{4}d^2 \times V \times gH}$$

$$0.80 = \frac{10{,}000 \times 10^3}{1000 \times \frac{\pi}{4}d^2 \times 97.06 \times 9.81 \times 500}$$

$$d^2 = \frac{80}{0.8 \times \pi \times 97.06 \times 9.81} = 0.0334 \text{ m}^2$$

$\therefore$ Diameter of jet, $d = 0.183$ m *Ans.*

Example 10.4 Show that the specific speed of a single jet Pelton wheel is about 202 (d/D) where d and D represent the jet and bucket wheel diameters respectively. Take $C_v = 0.97$, $f = 0.45$ and $h = 0.85$.

Solution

$$Q = aV = n\frac{\pi}{4}d^2 C_v\sqrt{2gH}, \text{ where } n = \text{number of jets.}$$

$$P = \rho QgH\eta = \rho\eta\frac{\pi}{4}d^2C_v\sqrt{2gH}.gH\eta \times 10^{-3} \text{ where } P \text{ is in kW.}$$

The peripheral velocity V_b of the Pelton wheel of diameter D, is

$$V_b = \frac{\pi DN}{60} = \phi\sqrt{2gH}\text{, where } f \text{ is the speed ratio.}$$

$$N = \frac{60\phi\sqrt{2gH}}{\pi D}$$

$$N_s = N\frac{\sqrt{P}}{H^{5/4}}$$

$$= \frac{60\phi\sqrt{2gH}}{\pi D}\frac{\sqrt{\rho n\frac{\pi}{4}d^2C_v\sqrt{2gH}\cdot gH\eta\times 10^{-3}}}{H^{5/4}}$$

or, $$N_s = \frac{60}{\pi}\cdot\phi\sqrt{2g}\cdot\frac{d}{D}\sqrt{\rho n\frac{\pi}{4}C_v\sqrt{2g}\cdot g\eta\times 10^{-3}}$$

Substituting $f = 0.45$, $C_v = 0.97$, $h = 0.85$, $\rho = 1000$ kg/m^3, we get

$$N_s = \frac{60}{\pi}\times 0.45\sqrt{2g}\,\frac{d}{D}\sqrt{n\frac{\pi}{4}\times 0.97\sqrt{2g}\times g\times 0.85}$$

$$= 38.07\,\frac{d}{D}\cdot 5\cdot 30\cdot\sqrt{n}$$

$$= 202\,\sqrt{n}\,\frac{d}{D}$$

For a single jet turbine, $n = 1$. Thus the specific speed is given by

$$N_s = 202\,\frac{d}{D}\text{. Hence proved.}$$

Example 10.5 Four jets each of 60 mm diameter strike the buckets of an impulse wheel and each gets deflected by an angle of 165°. The speed of the bucket wheel is 45 m/s. Find the velocity of the jet for maximum efficiency, power developed and the hydraulic efficiency. Assume that the bucket moves linearly.

Solution

For maximum efficiency, the jet velocity is,

$$V_1 = 2\,V_b = 2\times 45 = 90 \text{ m/s} \qquad \textit{Ans.}$$

Flow through the jet,

$$Q = \frac{\pi}{4}\,d^2\times V_1$$

$$= \frac{\pi}{4} \times (0.06)^2 \times 90$$
$$= 0.2545 \text{ m}^3/\text{s}$$

Power developed, P, is given by Eq. (10.14), i.e. $= \frac{1-k\cos\theta}{g} \cdot \frac{V_1^2}{4}$ w.g.

Taking friction coefficient k to be unity,

$$P = (1 - \cos 165°) \cdot \frac{90^2}{4} \times 0.2545 \times 1000 \times 10^{-3} \text{ kW} = 1013 \text{ kW}$$

For four jets

$$P = 4 \times 1013 = 4052 \text{ kW} \quad \textit{Ans.}$$

The maximum efficiency is given by Eq. (10.17), i.e.

$$(\eta_D)_{max} = \frac{1-k\cos\theta}{2}$$
$$= \frac{1-\cos 165°}{2} = 0.983 \text{ or } 98.3\% \quad \textit{Ans.}$$

Example 10.6 The peripheral velocity of the wheel of an inward flow reaction turbine is 20 m/s. The velocity of whirl of the inflowing water is 17 m/s and the radial velocity of flow is 2 m/s. If the flow is 0.7 m^3/s and the hydraulic efficiency is 80 per cent, find the head on the wheel, the power generated by the turbine and the angles of the vanes. Assume radial discharge.

Solution

The velocity triangles for the moving vanes are shown in Fig. E10.6. Since the discharge is radial, Vw_2 is zero.

$\therefore$ Hydraulic efficiency, $\eta_h = \frac{Vw_1 \cdot V_b}{gH}$

$$\therefore \quad 0.8 = \frac{17 \times 20}{9.81 \times H}$$

$\therefore$ Head on the wheel, $H = 43.3$ m *Ans.*

Power generated is,

$$P = \rho Q g H \eta_h = 10^3 \times 0.7 \times 9.81 \times 43.3 \times 0.8 \times 10^{-3}$$
$$= 238 \text{ kW}$$

Exit angle of guide vanes = α_1.

Now,

$$\alpha_1 = \tan^{-1} \frac{Vf_1}{Vw_1} = \tan^{-1} \frac{2}{17}$$
$$= 180 - 6.71 = 173.29° \quad \textit{Ans.}$$

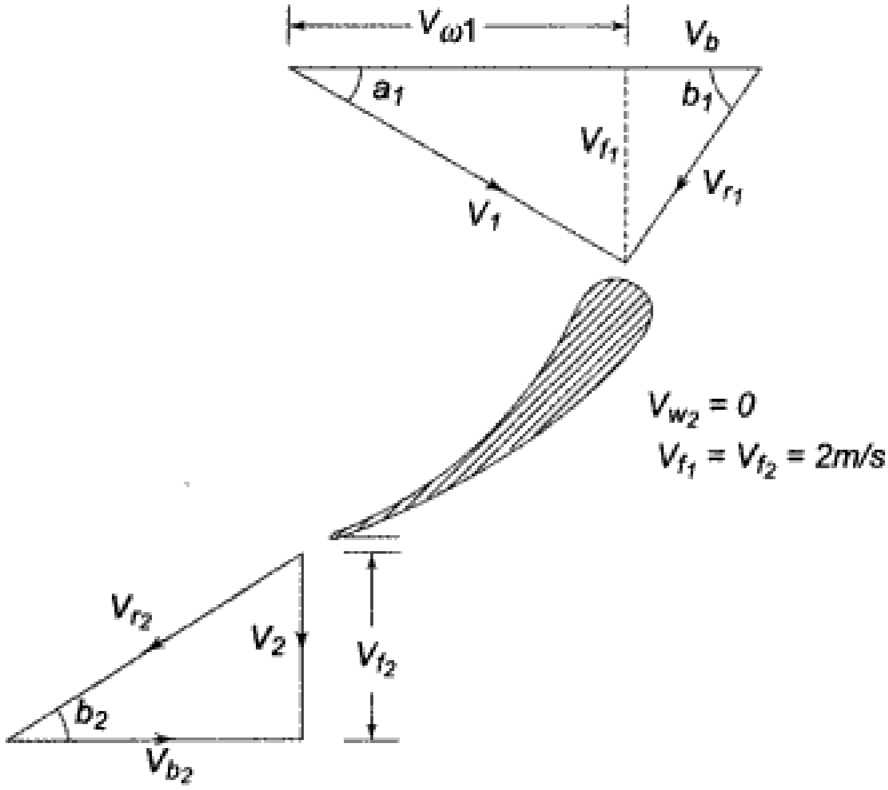


Fig. E10.6

Inlet blade angle = β_1

$$\beta_1 = \tan^{-1} \frac{V_{f1}}{V_{b_1} - V_{\omega_1}} = \tan^{-1} \frac{2}{20-17}$$

$$= \tan^{-1} 0.6667 = 33.7° \qquad \textit{Ans.}$$

Example 10.7 A runner of a Francis turbine having 1.50 m outer diameter and 0.75 m inner diameter operates under a head of 150 m with a specific speed of 120 and generates 14 MW. If the water enters the wheel at angle of 11°20′ and leaves the blade radially with no velocity of whirl, what will be the inlet and outlet blade angles? Assume the hydraulic efficiency to be 92 per cent.

Solution

$$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$120 = \frac{N\sqrt{14000}}{(150)^{5/4}}$$

$$\therefore \qquad N = 532 \text{ rpm}$$

$$V_{b1} = \frac{\pi DN}{60} = \frac{\pi \times 1.5 \times 532}{60} = 41.76 \text{ m/s}$$

$$\eta_h = 0.92 = \frac{V_{w1}V_{b1} - V_{w2}V_{b2}}{gH}$$

$$= \frac{V_{w1}V_{b1}}{gH}, \text{ since } V_{w2} = 0$$

$$V_{w1} = \frac{0.92 \times 9.81 \times 150}{41.76} = 32.42 \text{ m/s}$$

$$\tan \alpha_1 = \tan 11°20' = 0.2 = \frac{V_{f_1}}{V_{w_1}}$$

$$\therefore \quad V_{f_1} = 0.2 \times 32.42$$

$$= 6.49 \text{ m/s} = V_{f_2}$$

$$\tan \beta_1 = \frac{V_{f_1}}{V_{b1} - V_{w1}} = \frac{6.49}{41.76 - 32.42} = 0.6955$$

$$\therefore \quad \beta_1 = 34°49'$$ *Ans.*

Since the inner diameter is half the outer diameter,

$$V_{b2} = \frac{V_{b1}}{2} = \frac{41.76}{2} = 20.88 \text{ m/s}$$

$$\tan \beta_2 = \frac{V_{f_2}}{V_{b2} - V_{w2}} = \frac{6.49}{20.88 - 0} = 0.3108$$

$$\therefore \quad \beta_2 = 17.27 = 17°16'$$

Example 10.8 The following data relate to a Francis turbine: Net head = 70 m; Speed = 700 rpm; Overall efficiency = 85 per cent; Shaft power = 350 kW; Hydraulic efficiency = 92 per cent; Flow ratio, $V_{f1}\sqrt{2gH} = 0.22$; Breadth ratio, $B/D = 0.1$; Outer diameter of the runner = 2 × inner diameter of runner; Velocity of flow, V_f = constant; Outlet discharge = radial. The thickness of vanes occupies 6 per cent of circumferential area of the runner. Determine (a) the guide vane angle (b) the runner vane angles at inlet and outlet (c) the diameters of the runner at inlet and outlet; (d) the width of the wheel at inlet.

Solution

The velocity triangles are shown in Fig. E10.8.

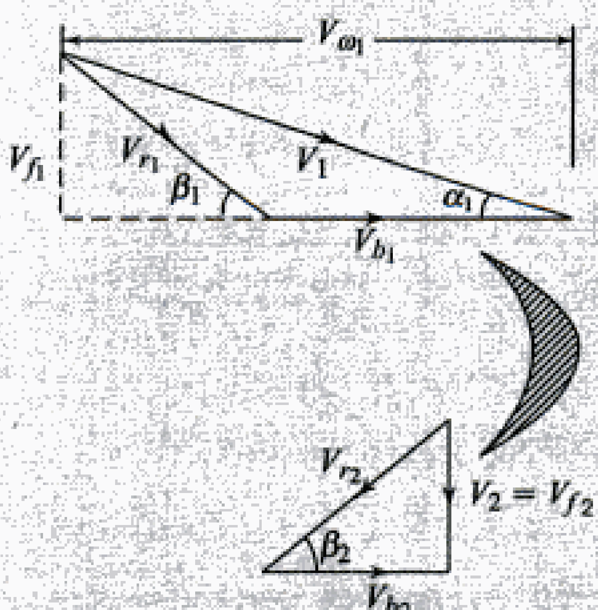


Fig. E10.8

Flow ratio, $\frac{V_{f1}}{\sqrt{2gH}} = 0.22$

$$\therefore \quad V_{f1} = 0.22\sqrt{2 \times 9.81 \times 76} = 8.153 \text{ m/s}$$
$$= V_{f2}$$
$$A_b = 0.94\,\pi D_1 B_1$$

Since discharge at outlet is radial,

$$V_{w2} = 0,\ V_{f2} = V_2 = 8.153 \text{ m/s}$$

Now
$$\eta_0 = \frac{\text{shaft power}}{\text{water power}} = \frac{350 \text{ kW}}{1000 \times Q \times 9.81 \times 70 \times 10^{-3}}$$

$$\therefore \quad Q = \frac{350}{9.81 \times 70 \times 0.85} = 0.5996 \text{ m}^3/\text{s} \cong 0.6 \text{ m}^3/\text{s}$$

$$Q = 0.94\,\pi D_1 B_1 \times V_{f1}$$
$$0.6 = 0.94\,\pi D_1 \times 0.1\,D_1 \times 8.153$$

$$\therefore \quad D_1 = 0.4992 \text{ m} \cong 0.5 \text{ m}$$ *Ans.* (c)

$$\therefore \quad B_1 = 0.1 \times 0.5 = 0.05 \text{ m} = 5 \text{ m}$$ *Ans.* (d)

$$D_2 = \frac{0.5}{2} = 0.25 \text{ m}$$ *Ans.* (c)

$$V_{b1} = \frac{\pi D_1 N}{60} = \frac{\pi \times 0.5 \times 700}{60} = 18.33 \text{ m/s}$$

$$\eta_h = \frac{V_{w1} V_{b1}}{gH} = \frac{V_{w1} \times 18.33}{9.81 \times 70} = 0.92$$

$$\therefore \quad V_{w1} = 34.47 \text{ m/s}$$

$$\tan\alpha = \frac{V_{f1}}{V_{w1}} = \frac{8.153}{34.47} = 0.2365$$

$$\therefore \quad \alpha = 13.3° = 13°\ 18'$$ *Ans.* (a)

$$\tan\beta_1 = \frac{V_{f1}}{V_{w1} - V_{b1}} = \frac{8.153}{34.47 - 18.33} = 0.5051$$

$$\therefore \quad \beta_1 = 26.8° = 26°\ 148'$$ *Ans.* (b)

$$\tan\beta_2 = \frac{V_f}{V_{b2}}$$

$$\frac{V_{b1}}{r_1} = \frac{V_{b2}}{r_2} = \omega$$

$$\therefore \quad V_{b2} = \frac{D_2}{D_1} V_{b1} = \frac{18.33}{2}$$
$$= 9.17 \text{ m/s}$$

$$V_b = \frac{\pi DN}{60} = 45.41$$

∴ Pitch circle diameter of the wheel, D is,

$$D = \frac{45.41 \times 60}{\pi \times 300} = 2.89 \text{ m} \qquad \textit{Ans.} \text{ (d)}$$

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} = \frac{300\sqrt{260.000}}{(475)^{5/4}} = 68.98 \qquad \textit{Ans.} \text{ (e)}$$

Jet ratio,

$$\frac{D}{d} = \frac{2.89}{0.227} = 12.73$$

∴ Number of buckets = $\frac{D}{2d} + 15 = \frac{12.73}{2} + 15 = 21.37$ or 22 Ans. (f)

Work done per kg, is

$$E = \frac{(V_1 - V_{b1})(1 - k\cos\theta)V_{b1}}{g}$$

$$= \frac{(94.6 - 45.41)(1 - 0.98\cos 165°) \times 45.41}{9.81}$$

$$= 443.24 \text{ kg–m/kg} \qquad \textit{Ans.} \text{ (g)}$$

$$\eta_h = \eta_{head} \times \eta_{dis} = \frac{443.24}{475} \times 0.9975$$

$$= 0.93 \text{ or } 93\% \qquad \textit{Ans.} \text{ (h)}$$

Example 10.10 Water is supplied to an axial flow turbine under a gross head of 35 m. The mean diameter of the runner is 2 m and it rotates at 145 rpm. Water leaves the guide vanes at 30° to the direction of the runner rotation and at mean radius the angle of the runner blade at outlet is 28°. If 7 per cent of the gross head is lost in the casing and guide vanes, and the relative velocity is reduced by 8 per cent due to friction in the runner, determine the blade angle at inlet and the hydraulic efficiency of the turbine.

Solution

Net heat $H = 0.93 \times 35$

$= 32.6$ m

∴ $V_1 = \sqrt{2gH} = \sqrt{2 \times 9.81 \times 32.6}$

$= 25.3$ m/s

$$V_b = \frac{\pi DN}{60} = \frac{\pi \times 2 \times 145}{60} = 15.2 \text{ m/s}$$

With Reference to Fig. E10.10,

$$V_{r1}\sin\beta_1 = V_1 \sin\alpha$$

$$V_{r1}\cos\beta_1 = V_1\cos\alpha - V_b$$

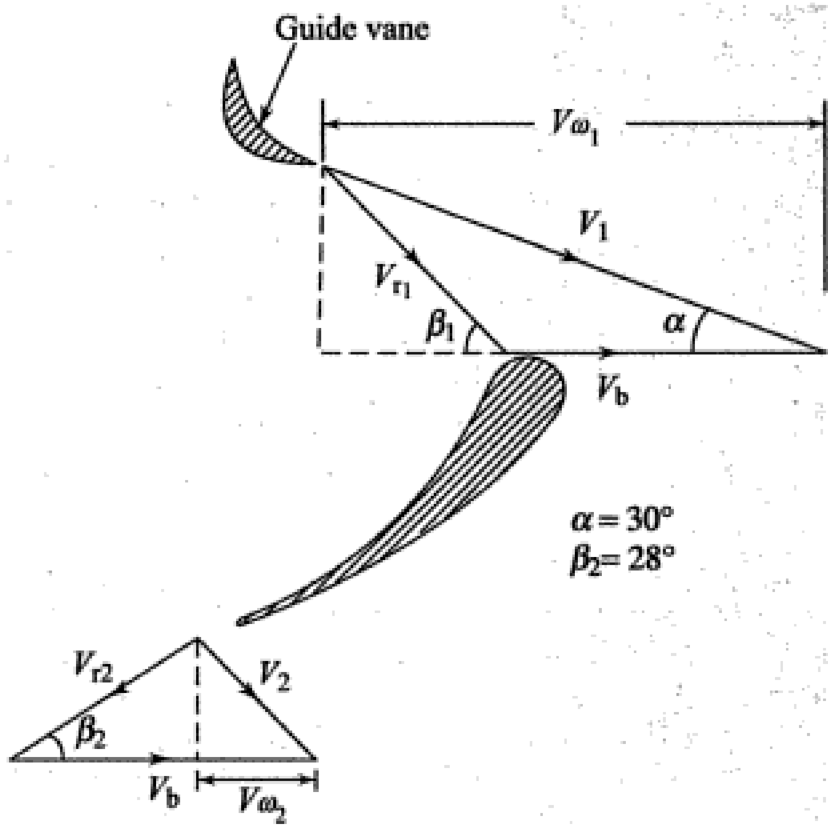


Fig. E10.10

$$\therefore \quad \tan\beta_1 = \frac{V_1 \sin\alpha}{V_1 \cos\alpha - V_b} = \frac{25.3 \sin 30°}{25.3 \cos 30° - 15.2} = 1.8852$$

$$\therefore \quad \beta_1 = 62.1°$$

$$V_{r1} = \frac{25.3 \sin 30°}{\sin 62.1°} = 14.31 \text{ m/s}$$

$$V_{r2} = 0.92 \times 14.31 = 13.17 \text{ m/s}$$

$$V_{w1} = V_1 \cos 30° = 21.91 \text{ m/s}$$

$$V_{w2} = V_b - V_{r2} \cos 28°$$

$$= 15.2 - 13.17 \times 0.883 = 3.57 \text{ m/s}$$

$$E = \frac{V_b(V_{w1} - V_{w2})}{g} = \frac{15.2(21.91 - 3.57)}{9.81} = 28.42 \text{ m}$$

$$\therefore \quad \eta_h = \frac{28.42}{35} = 0.812 \quad \text{or} \quad 81.2\% \quad \textit{Ans.}$$

Example 10.11 A Kaplan turbine develops 10000 kW under a head of 12 m when the following conditions prevail. Speed ratio = 2, flow ratio = 0.65, diameter of hub = 0.3 times the external diameter of the vane and the overall efficiency = 94 per cent. Estimate (a) the speed (b) the diameter of the runner and (c) the specific speed.

Solution

$$P = \rho Q g H \eta \times 10^{-3}$$

$$10000 = Q \times 9.81 \times 12 \times 0.94$$
$$Q = 90.37 \text{ m}^3/\text{s}$$

Flow ratio,

$$\phi' = \frac{V_{f1}}{\sqrt{2gH}} = 0.65$$
$$V_{f1} = 0.65 \sqrt{2 \times 9.81 \times 12} = 9.97 \text{ m/s}$$

Area of flow,

$$A_b = \frac{90.37}{9.97} = 9.064 \text{ m}^2$$
$$A_b = \frac{\pi}{4}\left(D^2 - d_b^2\right) = 9.064$$
$$D^2 - (0.3D)^2 = \frac{9.064 \times 4}{\pi} = 11.54$$
$$D^2 = \frac{11.54}{0.91} = 12.682$$

Runner diameter, $D = 3.56$ m *Ans.* (b)

Speed ratio $$\phi = 2 = \frac{V_b}{\sqrt{2gH}} = \frac{V_b}{\sqrt{19.62 \times 12}}$$

$$V_b = 30.69 \text{ m/s} = \frac{\pi DN}{60}$$
$$N = \frac{30.69 \times 60}{\pi \times 3.56} = 164.6 \quad \text{or} \quad 165 \text{ rpm}$$

Synchronous speed,

$$N = \frac{120 f}{p}$$

If we take $\frac{120 f}{p} = 165$, then $p = \frac{120 \times 50}{165} = 36.36$

Let 36 poles or 18 pairs of poles are taken.

Then $$N = \frac{120 \times 50}{36} = 166.7 \text{ rpm}$$ *Ans.* (a)

Specific speed $$N_s = \frac{N\sqrt{P}}{H^{5/4}}$$

$$= \frac{166.7\sqrt{10000}}{25^{5/4}} = 746$$ *Ans.* (i)

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$$\frac{180\sqrt{30000}}{H_p^{5/4}} = 210$$

$$\therefore \quad H_p^{5/4} = \frac{6\sqrt{30000}}{7} \quad \therefore H_p = 54.61 \text{ m}$$

Substituting, $\frac{D_p}{D_m} = \frac{285}{180}\sqrt{\frac{54.61}{4.5}} = 5.516$ *Ans.*

Flow through the turbine

$$Q_p = \frac{P_p}{\rho g H_p \eta} = \frac{30000 \times 1000}{1000 \times 9.81 \times 54.61 \times 0.88}$$

$$= 63.6 \text{ m}^3/\text{s}$$ *Ans.*

Example 10.15 Tests conducted on a one-fifth scale model of a Francis turbine under a head of 1.5 m indicated that it could develop 5 kW power at 450 rpm. Determine the speed and power of a full sized turbine while working under a head of 30 m.

Solution

$$\frac{D_m}{D_p} = \frac{N_p}{N_m}\sqrt{\frac{H_m}{H_p}}$$

$$\frac{1}{5} = \frac{N_p}{450}\sqrt{\frac{1.5}{30}}$$

$$\therefore \quad N_p = 90\sqrt{20} = 402 \text{ rpm}$$ *Ans.*

$$N_s = \frac{N_m\sqrt{P_m}}{H_m^{5/4}} = \frac{450\sqrt{5}}{(1.5)^{5/4}} = 606.16$$

Again, $N_s = \frac{N_p\sqrt{P_p}}{H_p^{5/4}} = 606.16 = \frac{402\sqrt{P_p}}{30^{5/4}} = 606.16$

$\therefore$ P_p = Power of the full sized turbine = 11208 kW *Ans.*

Example 10.16 A turbine works under a head of 19 m and has a maximum flow rate of 3 m^3/s and a speed of 600 rpm. If it has to work in another plant under a head of 5 m, at what speed must the turbine run in order to attain approximately the same efficiency and what will be the maximum flow rate?

Solution

$$\frac{D_m}{D_p} = \sqrt{\frac{H_m}{H_p}} \cdot \frac{N_p}{N_m}$$

Since the same turbine is used in both places, the diameter is the same.

Kaplan turbines $820 = \dfrac{250\sqrt{P}}{30^{5/4}}$

$\therefore$ $P = 53033.57$ kW

$\therefore$ Number of turbines $= \dfrac{89614.35}{53033.57} = 1.69$

i.e. 2 turbines *Ans.* (b)

Example 10.18 The following data refers to a proposed hydroelectric power plant:

Available head = 27 m, Catchment area 430 sq. km, Rainfall = 150 cm/year, Percentage of total rainfall utilized = 65%, Penstock efficiency = 95%, Turbine efficiency = 80%, Generator efficiency = 86% and Load factor = 0.45.

(a) Calculate the power developed.

(b) Suggest suitable turbines for the plant.

Solution

Quantity of water available per year

$$= (430 \times 10^6)\ \text{m}^2 \times 1.50\ \text{m} \times 0.65$$

$$= 419.25 \times 10^6\ \text{m}^3$$

Quantity of water available per second

$$Q = \frac{419.25 \times 10^6}{365 \times 24 \times 3600} = 13.29\ \text{m}^3$$

Power developed

$$P = h_p \times h_1 \times \eta_G\, \rho Q g H$$

$$= 0.95 \times 0.8 \times 0.86 \times 1000 \times 13.29 \times 9.81 \times 27 \times 10^{-3}$$

$$= 2300\ \text{kW}$$ *Ans.* (a)

$$\text{Load factor} = \frac{\text{Average load}}{\text{Peak load}} = 0.45$$

$$\therefore \quad \text{Peak load capacity} = \frac{2300}{0.45} = 5111\ \text{kW}$$

If two machines of equal capacity are provided,

$$\text{Capacity of each unit} = \frac{5111}{2 \times 0.86} = 2971.5\ \text{kW}$$

As the available head is low, Kaplan turbines are suggested. Two such turbines, each of 3000 kW capacity, may be installed.

Example 10.19 The run off data of a river at a particular site is tabulated in Table E10.19(a).

Table E10.19 (a)

Month	*Mean discharge (millions of cu.m.)*	*Month*	*Mean discharge (millions of cu.m.)*
January	30	July	80
February	25	August	100
March	20	September	110
April	0	October	65
May	10	November	45
June	50	December	30

(a) Draw the hydrograph and find the mean flow.
(b) Draw the flow duration curve.
(c) Find the power developed if the head available is 90 m and the overall efficiency of generation is 86 per cent. Assume each month of 30 days.

Solution

The hydrograph of the given data is shown in Fig. E10.19 (a).

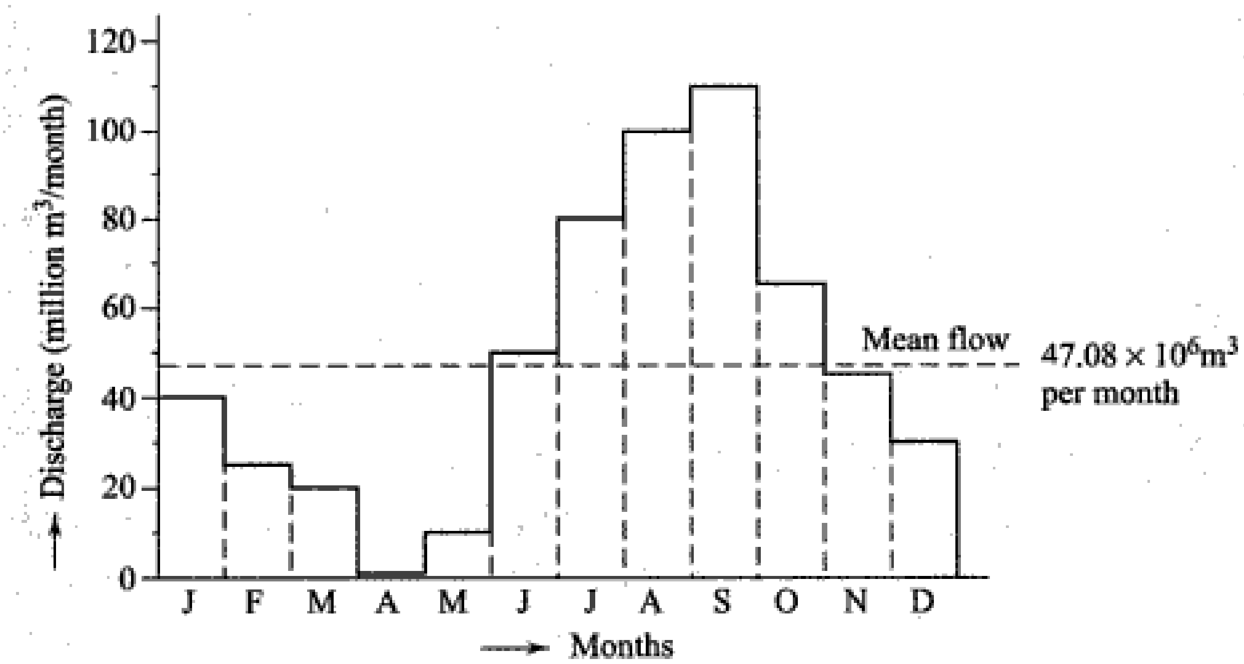

Fig. E10.19 (a) *Hydrograph*

The mean discharge

$$= \frac{30 + 25 + 20 + 0 + 10 + 50 + 80 + 100 + 110 + 65 + 45 + 30}{12}$$

$$= \frac{565}{12} = 47.08 \text{ million m}^3\text{/s}$$

To obtain the flow duration curve, it is necessary to find the lengths of time during which certain flows are available as given in Table E10.19 (b).

Table E10.19 (b)

Discharge per month (million m³)	*Total number of months during which flow is available*	*Percentage time*
0	12	100
10	11	91.7
20	10	83.3
25	9	75.0
30	8	66.7
40	6	50.0
50	5	41.7
60	4	33.3
70	3	25.0
80	3	25.0
90	3	25.0
100	2	16.7
110	1	8.3

The flow duration curve from the above data shown in Fig. E10.19 (b).

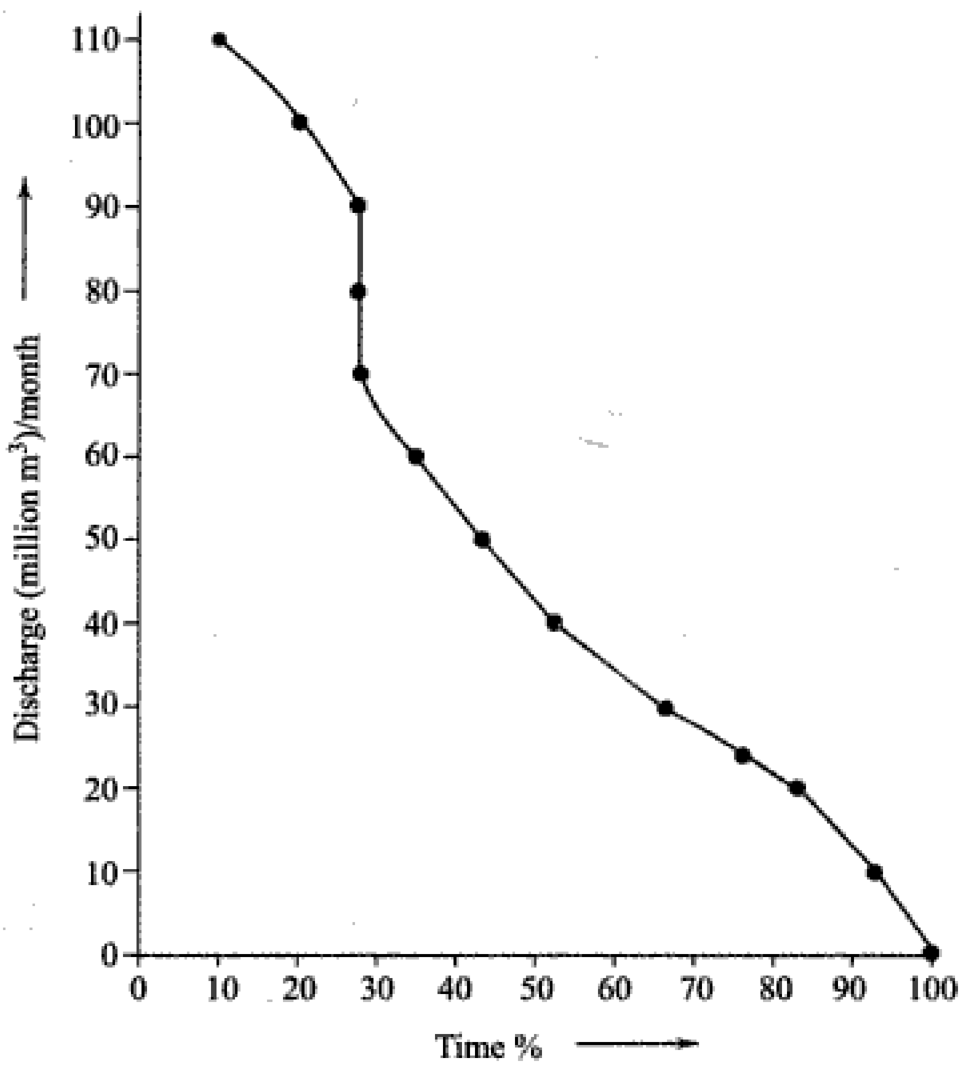

Fig. E10.19 (b) *Flow Duration Curve*

Power developed,

$$P = rQgH\eta_0 \times 10^{-3} \text{ kW}$$

$$= \frac{47.08 \times 10^6 \times 9.81 \times 90 \times 0.86}{30 \times 24 \times 3600}$$

$$= 13.79 \text{ MW}$$ *Ans.*

SHORT-ANSWER QUESTIONS

10.1 What are the two key parameters of water on which the magnitude of hydro-power depends?

10.2 Enlist the advantages and disadvantages of water power.

10.3 Explain the optimization of hydro-thermal mix in meeting the power demand of a certain region.

10.4 Discuss the factors which should be considered while selecting a site for a hydroelectric plant.

10.5 What do you understand by hydrology? Explain the hydrological cycle.

10.6 What do you mean by (i) hydrograph (ii) Flow duration curve and (iii) power duration curve? What is primary power and secondary power?

10.7 What is a mass curve? What does the slope of the curve at a point indicate?

10.8 Explain what you mean by storage and pondage. Why are they required?

10.9 State the essential elements of a hydroelectric power plant.

10.10 What is a catchment area? Why is a reservoir required?

10.11 State the functions of a dam. How are dams classified? Briefly describe a few important types of dams. How would you select the site and the type of the dam?

10.12 What is a spillway? Why are spillways required? What are the different types of spillways?

10.13 Explain the following terms
- (i) Headrace
- (ii) Tailrace
- (iii) Canal
- (iv) Flume
- (v) Tunnel
- (vi) Pipeline
- (vii) Penstock

10.14 What is a surge tank? Why is it important in a hydro-plant?

10.15 What is the function of a draft tube? Briefly explain the different types of draft tubes?

10.16 Enlist the various equipment provided in a powerhouse.

10.17 Explain the different methods of classifying a hydroelectric power plant. What is a run off river plant?

10.18 Explain with a neat sketch a pumped storage plant. What are its advantages?

10.19 What are mini and micro-hydel plants? Why are they important these days?

10.20 How does a hydraulic turbine convert energy? What is a waterwheel?
10.21 Describe the classification of hydraulic turbines in different categories.
10.22 How is the size of turbine ascertained?
10.23 Classify the hydro-turbines according to head, power, size and specific speed.
10.24 What type of turbine would you recommend for the following heads and why?
 (a) 1000 m
 (b) 150 m
 (c) 20 m
10.25 Explain with a neat sketch the principle of operation of a Pelton turbine.
10.26 What are the functions of (a) spear and (b) deflector plate in a Pelton wheel?
10.27 Deduce the ratio between the peripheral velocity of the runner and the velocity of the jet for attaining (a) maximum efficiency and (b) maximum power.
10.28 Explain the following terms.
 (i) Jet ratio
 (ii) Speed ratio
10.29 How are the number of jets in a Pelton wheel ascertained?
10.30 "The number of buckets in a Pelton wheel is a function of the jet ratio". Explain.
10.31 How is the degree of reaction, R, of a hydraulic turbine defined? Explain the cases for $R = 0$, $R = 0.50$ and $R = 1$.
10.32 Explain with a neat schematic diagram the operation of a Francis turbine. What are its advantages?
10.33 Draw the velocity diagrams of an inward-flow Francis turbine and derive the expression of blading efficiency in terms of vane angles.
10.34 What are Kaplan turbines? How is a Kaplan turbine different from a propeller turbine? Explain the characteristic features of a Kaplan turbine.
10.35 What is a Deriaz turbine? What is its importance?
10.36 What is a bulb turbine? Where is it used?
10.37 Define specific speed of a turbine. Derive its expression in terms of speed, power and head.
10.38 What is scale ratio? What is its importance?
10.39 Define unit speed, unit power and unit discharge and derive their relevant relations.
10.40 What do you understand by cavitation? What are its effects? How can it be minimized?
10.41 Write short notes on the following.
 (i) pitting of turbine blades, and its prevention (ii) servo-motors.
10.42 What is the synchronous speed of the turbine runner? How is it estimated?
10.43 Explain with a neat sketch the governing principle of an impulse turbine. What are the functions of needle valve and the deflector?
10.44 How is the governing of a reaction turbine carried out? Explain with a neat sketch. What is the function of relief valve?

10.45 When and why are surge tanks and forebays provided? Explain a few types of surge tanks.

10.46 Discuss with neat sketches the characteristic curves related to the performance of hydraulic turbine.

10.47 What is "runaway speed"? How does it affect the turbine design?

10.48 How is the type of turbine selected in a certain hydro-plant? Discuss the effects of head, specific speed, height of installation, the operating characteristics and the capacity on the selection process.

PROBLEMS

10.1 A Pelton wheel is required to develop 4500 kW at 400 rpm operating under an available head of 360 m. There are two equal jets and the bucket angle is 170°. The bucket pitch circle diameter is 1.82 m. Taking k for the buckets as 0.85, determine (a) the efficiency of the runner and (b) the diameter of each jet.

Ans. (a) 0.9106 (b) 0.103 m

10.2 In a Pelton wheel the diameter of the bucket circle is 2 m and the deflecting angle of the bucket is 162°. The jet has 165 mm diameter, the pressure behind the nozzle is 700 kPa and the wheel rotates at 320 rpm. Neglecting friction, find the power developed by the wheel and the hydraulic efficiency.

Ans. 351.9 kW, 0.616

10.3 A Pelton wheel develops 8 MW under a net head of 130 m at a speed of 200 rpm. Assuming $c_v = 0.98$, hydraulic efficiency = 87 per cent, speed ratio = 0.46 and the ratio of jet-to-wheel diameter = 1/9, determine (a) the flow required (b) the diameter of the wheel (c) the diameter and number of jets needed.

Ans. (a) 7.51 m^3/s,(b) 2.17 m, (c) 0.242 m, 3.

10.4 A Pelton wheel has to develop 12 MW under a head of 300 m at a speed of 500 rpm. If the diameter of the jet is not to exceed 1/9 of the wheel diameter, estimate the number and diameter of the jets, diameter of the bucket wheel and the quantity of flow. Assume overall efficiency = 88 per cent, $C_v = 0.97$ and $\phi = 0.45$.

Ans. $D = 1.32$ m, $d = 0.147$ m, $n = 4$ and $Q = 4.63$ m^3/s

10.5 A Pelton wheel to be designed is to run at 300 rpm under an effective head of 150 m. The ratio of the nozzle diameter to the pitch circle diameter is 1/12. Assuming efficiency = 84%, $C_v = 0.98$ and speed ratio = 0.45, determine (a) the diameter of the wheel, (b) diameter of the jet (c) the quantity of water flow (d) the minimum number of buckets required; and (e) the power developed.

Ans. (a) 1.55 m (b) 0.129 m (c) 0.694 m^3/s (d) 21 (e) 858 kW.

10.6 A jet of 75 mm diameter strikes the bucket of an impulse wheel and gets deflected by an angle of 165°. The speed of the bucket is 45.5 m/s. Find the velocity of the jet for maximum efficiency and the power developed.

Ans. 91 m/s, 1545 kW.

10.7 An inward flow reaction turbine having an overall efficiency of 75 per cent delivers 132 kW. The head H is 9 m, velocity of the periphery of the wheel is 54 m/s and the radial velocity is 18 m/s. The wheel makes 120 rpm. The hydraulic losses in the turbine are 20 per cent of the available energy. Determine the discharge at the inlet, the guide blade angle, the wheel blade angle and the diameter and width of the wheel. Assume radial discharge.

Ans. 2 m^3/s, $a_1 = 56° 49'$ $\beta_1 = 23°$, $D = 2.86$ m $B = 0.037$ m.

10.8 In a Francis turbine of low specific speed, the velocity of flow from inlet to exit of the runner remains constant. If the turbine discharges radially, show that the degree of reaction R can be expressed as

$$R = \frac{1}{2} - \frac{1}{2}\left[\frac{\cot\beta_1}{\cot\alpha - \cos\beta_1}\right]$$

where α and β_1 are the guide and runner vane angle respectively and the degree of reaction is equal to the ratio of pressure drop to the hydraulic work done in the runner, assuming that the losses in the runner and negligible.

10.9 A Francis turbine with an overall efficiency of 76 per cent is required to produce 180 kW. It is working under a head of 8 m. The peripheral velocity is $0.25\ (2\ gH)^{1/2}$ and the radial velocity of flow is $0.95\ (2\ gH)^{1/2}$. The wheel runs at 150 rpm and the hydraulic losses in the turbine are 20 per cent of the available energy. Assuming radial discharge, determine (a) the guide vane angle (b) the wheel vane angle at inlet (c) the diameter of the wheel at inlet; and (d) the width of the wheel at inlet.

Ans. (a) 30° 45′, (b) 35° 12′, (c) 0.398 m, (d) 0.203 m

10.10 An inward flow reaction turbine works under a head of 22.5 m. The external and internal diameters of the runner are 1.35 m and 1 m respectively. The angle of guide vanes is 15° and the moving vanes are radial at inlet. Radial velocity of flow through the runner is constant and there is no velocity of whirl at outlet. Determine the speed of the runner in rpm and the angle of vane at outlet. If the turbine develops 375 kW, find the specific speed. Neglect friction losses.

Ans. 206.5 rpm, 19°53′, 81.6

10.11 Two inward flow reaction turbines have the same runner diameter of 0.60 m and the same efficiency. They work under the same head and they have the same velocity of flow of 6 m/s. One of the runners A revolves at 520 rpm and has an inlet vane angle of 65°. If the other runner B has an inlet vane angle of 110°, at what speed should it run?

Ans. 600 rpm.

10.12 Water enters an inward flow turbine at an angle of 22° to the tangent to the outer rim and leaves the turbine radially. If the speed of the wheel is 300 rpm and the velocity of flow is constant at 3 m/s, find the necessary angles of blades when the inner and outer diameters of the turbine are 0.3 m and 0.6 m respectively. If the width of the wheel at inlet is 0.15 m, calculate the power developed. Neglect the thickness of blades.

Ans. $\beta_1 = 59°54'$, $\beta_2 = 32°32'$, $P = 61.15$ kW.

size if (a) Francis turbines having specific speed not greater than 200 or (b) Kaplan turbines of specific speed not greater than 600, are used.

Ans. (a) 16, (b) 3.

10.21 A run off of 30 m^3/s is available at 7.5 m head for generating the desired power. The turbine efficiency is 85 per cent. (a) Is it feasible to develop the desired power by two turbines with 50 rpm and the specific speed of turbine not greater than 450? (b) What type of runner is required to be used? (c) What is the diameter of the runner if the speed ratio is 0.85?

Ans. (a) $As/N_s = 203$, two turbine units can be used. (b) Francis turbine, (c) 3.93 m.

10.22 From the following table of mean monthly discharge, draw the following curves.

Month	*Discharge (m^3/s)*	*Month*	*Discharge (m^3/s)*
January	100	July	1100
February	325	August	1300
March	400	September	1000
April	700	October	800
May	850	November	600
June	900	December	300

(a) the hydrograph (b) the flow duration curve.

10.23 From the following table of the mean monthly discharge for 12 months of a river at a site, draw (a) the hydrograph and find the average monthly flow, and the power available at mean flow of water for head 90 m and overall efficiency of generation 90 per cent. Assume 30 days in each month.

Month	April	May	June	July	August	September
$Q - m^3 \times 10^6$	500	200	1500	2500	3000	2400
Month	October	Nov.	Dec.	January	Feb.	March
$Q - m^3 \times 10^6$	2000	1500	1500	1000	800	600

(b) Draw the flow duration curve from the data in the hydrograph.

10.24 The following data pertain to a hydroelectric plant. Available head = 140 m, catchment area = 2000 sq. km; annual average rainfall = 145 cm, turbine efficiency = 85%, generator efficiency = 90%, percolation and evaporation losses = 16%. Determine the power developed and suggest the type of turbine to be used if the runner speed is to be kept below 240 rpm.

Ans. 8.11 MW, Pelton turbine with 4 jets may be used.

10.25 (a) Discuss the differences between Kaplan, Francis and Pelton turbines and state the types of power plants they are suitable for.

(b) At a particular hydroelectric power plant site the discharge of water is 400 m^3/s and the head is 25 m. The turbine efficiency is 88 per cent. The generator is directly coupled to the turbine having frequency of generation as 50 cycles/s and number of poles as 24. Calculate the least number of turbines required if (a) a Francis turbine is used with a specific speed of 300 (b) a Kaplan turbine with a specific speed of 750 is used.

1. A Pelton wheel is required to developed (4.5 MW) at (400 rpm) operating under an available heat of (360 m). There are two equal jets and the bucket angle is (170^o). The diameter of bucket pitch circle is (1.82 m). Taking (k=0.85), Determine (a) The overall efficiency of the runner and (b) the diameter of each jet. **[Ans. (a) 0.9106 (b) 0.103 m]**

2. In a Pelton wheel, the diameter of the bucket circle is (2 m) and the deflecting angle of the bucket is (162^o). The jet has (165 mm) diameter, the water pressure behind the nozzle is (700 kPa) and the wheel rotates at (320 rpm). Neglecting friction, find the power developed by the wheel and the hydraulic efficiency. **[Ans. 351.9 kW, 0.616]**

3. A Pelton wheel develops (8 MW) under a net head of (130 m) at a speed of (200 rpm). Assuming $(Velocity\ coefficient, C_v = 0.98), (Hydraulic\ efficiency, \eta_h = 87\%), (Speed\ ratio, \psi = 0.46)\ and\ \left(Ratio\ of\ jet\ to\ wheel\ diameter \frac{d}{D} = 1/9\right)$ Determine (a) The volumetric flow required (b) The diameter of the wheel (c) The diameter and number of jets needed. **[Ans. (a) 7.51 m^3/s (b) 2.17 m (c) 0.242 m, 3]**

4. A Pelton wheel has to develop (12 MW) under a head of (300 m) at a speed of (500 rpm). If the diameter of the jet is not to exceed (1/9) of the wheel diameter. Estimate the number and diameter of jets, diameter of the bucket wheel and the volumetric flow rate. Assume$(Overall\ efficiency, \eta_o = 88\%), (Velocity\ coefficient, C_v = 0.97), and (Speed\ ratio, \psi = 0.45)$. **[Ans. D=1.32 m, d=0.147 m, n=4, and Q=4.63 m^3/s]**

5. A Pelton wheel to be designed is to run at (300 rpm) under an effective head of (150 m). The ratio of the nozzle diameter to pitch circle diameter is (1/12). Assume$(Overall\ efficiency, \eta_o = 84\%), (Velocity\ coefficient, C_v = 0.98), and (Speed\ ratio, \psi = 0.45)$. Determine (a) the diameter of the bucket wheel (b) diameter of jet (c) the volumetric flow rate (d) The minimum number of buckets required and (e) The power developed. **[Ans. (a) 1.55 m (b) 0.129 m (c) 0.694 m^3/s (d) 21 (e) 858 kW]**

6. A jet of (75 mm) diameter strikes the bucket of an impulse wheel and gets deflected by an angle of (165^o). The speed of the bucket is (45.5 m/s). Find the velocity of the jet for maximum efficiency and the power developed. **[Ans. 91 m/s, 1545 kW]**

7. An inward flow reaction turbine having an overall efficiency of (75%) delivers (132 kW). The head is (9 m), the peripheral velocity of the wheel is (54 m/s) and the radial velocity is (18 m/s). The wheel runs at (120 rpm). The hydraulic losses in the turbine are (20%) of the available energy. Determine the discharge at the inlet, the guide blade angle, the wheel blade angle and the diameter and width of the wheel. Assume radial discharge. $[\boldsymbol{Ans.\, Q = 2\, m^3/s, \alpha_1 = 56^o 49', \beta_1 = 23^o, D_1 = 2.86\, m, B_1 = 0.037\, m}]$

8. A Francis turbine with an overall efficiency of (76%) is required to produce (180 kW). It is working under a head of (8 m). The peripheral velocity is $\left(0.25\sqrt{2gH}\right)$ and the radial velocity of flow is $\left(0.95\sqrt{2gH}\right)$. The wheel runs at (150 rpm) and the hydraulic losses in the turbine are (20%) of the available energy. Assuming radial discharge, determine (a) the guide vane angle (b) the wheel vane angle at inlet (c) the diameter of the wheel at inlet and (d) the width of the wheel at inlet. $[\boldsymbol{Ans.\, (a)\, 30^o 45', (b)\, 35^o 12', (c)\, 0.398\, m, (d)\, 0.203\, m}]$

9. An inward flow reaction turbine works under a head of (22.5 m). The external and internal diameters of the runner are (1.35 and 1 m) respectively. The angle of guide vanes is (15^o) and the moving vanes are radial at inlet. Radial velocity of flow through the runner is constant and there is no velocity of whirl at outlet. Determine the speed of the runner in rpm and the angle of vane at outlet. If the turbine develops (375 kW), find the specific speed. Neglect friction losses. **[Ans. 206.5 rpm, $19^o 53'$, 81.6]**

10. Two inward flow reaction turbines have the same runner diameter of (0.6 m) and the same overall efficiency. They work under the same head and they have the same velocity of flow of (6 m/s). One of runners A runs at (520 rpm) and has an inlet vane angle of (65^o). If the other runner B has an inlet vane angle of (110^o), at what speed should it run?
[Ans. 600 rpm]

11. Water enters an inward flow turbine at an angle of (22^o) to the tangent to outer rim and leaves the turbine radially. If the speed of the wheel is (300 rpm) and the velocity of flow is constant at (3 m/s), find the necessary angles of blade when the inner and outer diameters of the turbine are (0.3 and 0.6 m) respectively. If the width of the wheel at inlet is (0.15 m), calculate the power developed. Neglect the thickness of blades. $[\boldsymbol{Ans.\, \beta_1 = 59^o 54', \beta_2 = 32^o 32', P = 61.15\, kW}]$

12. A Kaplan turbine develops (9000 kW) under a net head of (7.5 m). The overall efficiency of the wheel is (86%). The speed ratio is (2.2), and the flow ratio is (0.66). The diameter of the hub is 0.35 times the external diameter of the wheel. Determine the diameter of the runner and the specific speed of the runner. **[Ans. 5.005 m, 785.76]**

13. A Kaplan turbine working under a head of (25 m) develops (16000 kW) shaft power. The outer diameter of the runner is (4 m) and the hub diameter is (2 m). The guide blade angle is (35^o). The hydraulic and overall efficiency are (90% and 85%) respectively. If the velocity of whirl is zero at outlet, determine the runner vane angles at inlet and outlet and speed of turbine. $[\boldsymbol{Ans. 43.88^o, 22.06^o, 94.16\ rpm}]$

14. A Kaplan turbine works under a head of (22 m) and runs at (150 rpm). The diameters of the runner and the hub are (4.5 m and 2 m) respectively. The flow ratio is (0.43). The inlet blade angle is (163˚19′). If the turbine discharges radially at outlet, determine the guide vane angle and the outlet blade angle. $[\boldsymbol{Ans. \alpha_1 = 59.85^o, \beta_2 = 14.32^o}]$

15. A Kaplan turbine develop (7350 kW). The net available head is (5.5 m). Assume that the speed ratio is (0.68) and the overall efficiency is (60%). The diameter of the hub is 1/3 of the diameter of the runner. Find the diameter of the runner, its speed and its specific speed. **[Ans. 6.69 m, 65.86 rpm, 670.37]**