DESIGN ANALYSIS OF BULB TURBINE

A DISSERTATION

Submitted in partial fulfillment of the requirements for the award of the degree of

MASTER OF TECHNOLOGY

in

ALTERNATE HYDRO ENERGY SYSTEMS

By

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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in this dissertation, entitled, "**DESIGN ANALYSIS OF BULB TURBINE**", in partial fulfillment of the requirements for the award of the degree of **Master of Technology** in "**Alternate Hydro Energy Systems**", submitted in Alternate Hydro Energy Centre, Indian Institute of Technology, Roorkee is an authentic record of my own work carried out during the period from July 2009 to June 2010 under the supervision of **Dr.R.P.Saini**, Associate Professor, Alternate Hydro Energy Centre and **Dr.B.K.Gandhi**, Professor, Department of Mechanical and Industrial Engineering, Indian Institute of Technology, Roorkee, India.

I have not submitted the matter embodied in this dissertation for award of any other degree.

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ABSTRACT

In view of increasing demand of energy and world's depleting fossil fuel reserves, which provide the major source of energy, the development of renewable energy sources has got great importance. Small hydropower (SHP) is an important renewable energy source. There exists a large potential in the range of low head small hydropower schemes on canal falls and irrigation dams. In low head hydropower schemes, energy per unit discharge is low and discharges to be handled are larger. Therefore the cost of civil structure and electromechanical equipment is higher. Due to higher cost previously sufficient attention was not paid to these low head power schemes but these days, increase in tariff rate and increasing energy demand necessitates to develop low head sites. Now several suitable low head turbines are being offered by manufacturers to make this sector attractive for development.

In the present study, it is found that bulb turbine is more suitable for such conditions. Straight flow passage of bulb turbine leads to less hydraulic losses. In view of above, for low head hydroelectric power plant the major costs is the initial investment in the civil works structure and the electromechanical equipment. If the cost of the structure is reduced, more attractive small hydroelectric installations would be.

In the present study, an attempt has been made to reduce the civil structure cost by reducing the intake structure cost because intakes for bulb turbines are the important structure from cost point of view as they are large in relation to their runner diameters. As the water velocity is low in the intake section of bulb turbine hence the losses are small. Due to this reason intake of bulb turbine can be simplified and shortened to decrease the civil structure cost. In the present study, possibilities for simplifying and shortening the intake have been investigated. For this purpose a typical power house having bulb turbine layout has been selected. The dimensions of various components are worked out in terms of runner diameter which is important parameter for deciding the sizing of components. Four intakes of different geometry has been designed. Head losses and capital cost of material is worked out for each type of intake. Due to simplifications and shortening of the intakes head loss is increases but capital cost is decreases. A study has been carried out by capitalizing the energy loss and comparing change in capital cost for different option of intakes. Based on the study carried out optimum intake selection is recommended.

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CHAPTER-1
INTRODUTION

1.1 PRESENT SCENARIO OF ENERGY

The world is facing twin energy-related threats: that of not having adequate and secure supplies of energy at affordable prices and that of environmental harm caused by consuming too much of it. Soaring energy prices and recent geopolitical events have reminded us of the essential role, energy plays in economic growth and human development, and of the vulnerability of the global energy system to supply disruptions. Safeguarding energy supplies is once again at the top of the international policy agenda. Yet the current pattern of energy supply carries the threat of severe and irreversible environmental damage including changes in global climate. Global primary energy demand in the Reference Scenario is projected to increase by just over one-half between now and 2030 - an average annual rate of 1.6%. Demand grows by more than one-quarter in the period to 2015 alone [1].

Global energy-related carbon dioxide (CO_2) emissions increase by 55% between 2004 and 2030, or 1.7% per year, in the Reference Scenario. They reach 40 gigatonnes (Gt) in 2030, an increase of 14 Gt over the 2004 level. Power generation contributes half of the increase in global emissions over the projection period. Coal overtook oil in 2003 as the leading contributor to global energy-related CO_2 emissions and consolidates this position through to 2030 [1].

Developing countries account for over three-quarters of the increase in global CO_2 emissions between 2004 and 2030 in this scenario. In the developing countries need of energy is increasing at faster rate. Energy is critical in developing countries not only for economic growth but also for social development and human welfare. Driven by rising populations, expanding economies, energy intensive industries, urbanization and a quest for modernization and improved quality of life. The share of developing countries in world emissions rises from 39% in 2004 to over one-half by 2030 [1].

So this scenario emphasizes to take immediate actions to steer the energy system onto a more sustainable path. In the Alternative Policy Scenario, the policies and measures that governments are currently considering aimed at enhancing energy security and mitigating CO_2 emissions are assumed to be implemented. This would result in significantly slower growth in fossil-fuel demand, in oil and gas imports and in emissions. These interventions include efforts to improve efficiency in energy production and use, to increase reliance on non-fossil fuels energy that is clean and green energy like Hydro, Wind, Solar and Bio Energy .

1.2 BENEFITS OF HYDROPOWER

Hydropower conserves our fossil fuel reserves, is in abundant, self-renewing supply, is non polluting and produces no waste streams. The world's hydropower potential from rivers and reservoirs amounts to no less than 15 billion MWh per year-and only about 20 percent of this has been developed so far. Hydropower is not only environmentally friendly, but also cost effective. Hydropower has the highest operating efficiency of all known generation systems. They are largely automated, and their operating costs are relatively low. Hydropower plants also play an important role in water resources management, in preventing flooding making rivers navigable, solving irrigation problems and creating recreation areas [2].

1.3 LIMITATIONS OF LARGE HYDROPOWER

Large hydro is having disadvantages also. Due to large hydro there is huge displacement of population and vast area is submerged in the reservoir these are the main reasons due to which large hydro earned so much social opposition. Tehri and Sardar sarover are the example of not acceptability of large hydro in the common people, although these plants have been commissioned. There are other following drawbacks and reasons so as large hydro not considered as renewable energy source.

- Large hydro does not have the poverty reduction benefits of decentralized renewable. Including large hydro in renewable initiatives would crowd out funds for new Renewable.
- Promoters of large hydro regularly underestimate costs and exaggerate benefits.
- Large hydro will increase vulnerability to climate change.
- Large hydro projects have major social and ecological impacts.
- Efforts to mitigate the impacts of large hydro typically fail.
- Large reservoirs can emit significant amounts of greenhouse gases.
- Large hydro is slow, lumpy, inflexible and getting more expensive.
- Many countries are already over-dependent on hydropower.
- Large hydro reservoirs are often rendered non-renewable by sedimentation.

1.4 SMALL HYDRO POWER

There is a general tendency in all over the world to define small hydro by power output. Different countries follow different norms, the upper limit ranges between 5 to 50 MW. These ranges are given in Table 1.1.

Table1.1 : Worldwide definitions for small hydropower [3]

Country	Capacity
UK	<= 5MW
UNIDO	<=10MW
India	<=25MW
Sweden	<=15MW
Australia	<=20MW
China	<=25MW
New Zealand	<=50MW

In India, small hydro power schemes are classified by the central electricity authority (CEA) as follows in the Table 1.2.

Table 1.2: Various capacity of small hydropower [3]

Type	Station capacity	Unit rating
Micro	Up to 100 kW	Up to 100 kW
Mini	101 kW to 2000 kW	101 kW to 1000 kW
Small	2001 kW to 25000 kW	1001 kW to 5000 kW

1.4.1 Advantages of Small Hydropower

Following are the advantages of small hydro power

- It is reliable, eco-friendly and has mature and proven technology.
- More suitable for sensitive mountain ecology.
- Can be exploited wherever sufficient water flows.
- Does not involve large dams or problem of deforestation, submergence and rehabilitation.
- It is non polluting, entails no waste or production of toxic gases.
- Minimal transmission losses.
- No cost involves in transportation of fuel, which occurs in coal diesel and gas based power plants.
- Operation and maintenance cost is very low.

- Small investment, gestation period & payback period is also low.
- It is very good option for rural electrification.

1.5 PRESENT SCENARIO OF SMALL HYDROPOWER IN INDIA

About 80,000 villages remain yet to be electrified in spite of the highest priority given to rural electrification in India. Most of these villages are located in remote areas, with very low load densities requiring heavy investment in electrifying these villages. In rural areas, energy is needed for cooking, lighting, water pumping, agro and rural industry. In these remote areas transmission of grid power is totally uneconomical. So for these conditions small hydro power is emerging as appropriate answer of energy need. It is also helpful in handling the present energy crisis.

India is blessed with many rivers and mountains offering tremendous hydro potential of major, small, mini micro hydropower. Contribution of small hydropower has grown substantially in the last ten years. Small hydropower (SHP) is among the one of the most promising renewable energy resource today. Presently in India total identified potential is around 14294.24 MW. But we are harnessing 2045.61 MW with the help of 611 sites. Further 225 projects are under construction having capacity of 668.86 MW. Target capacity addition by 2012 is 1400MW [3]. Present status of SHP in India is tabulated in Table 1.3.

Table 1.3: Present status of SHP in India (upto 25 MW Capacity) [4]

Overall potential	**15,000 MW**
Identified potential	14294.24 MW (5403 sites)
Installed capacity (as on 31/06/2007)	2045.61 MW (611 projects)
Under construction (as on 31/06/2007)	668.86 MW (225 projects)
Target capacity addition – 11th Plan (2007-12)	1400 MW

1.6 TYPES OF SHP SCHEMES

Small Hydro Power can also be broadly categorized in three types as follows:

1.6.1 Run-off-River Scheme

Run-of-River hydroelectric schemes are those, in which water is diverted towards power house, as it comes in the stream. Practically, water is not stored during flood periods as well as during low electricity demand periods, hence water is wasted. Seasonal changes in river flow and weather conditions affect the plant's output. After power generation water is again discharged back to the stream. Generally, these are high head and low discharge schemes. The typical run off river scheme is shown in Fig. 1.1.

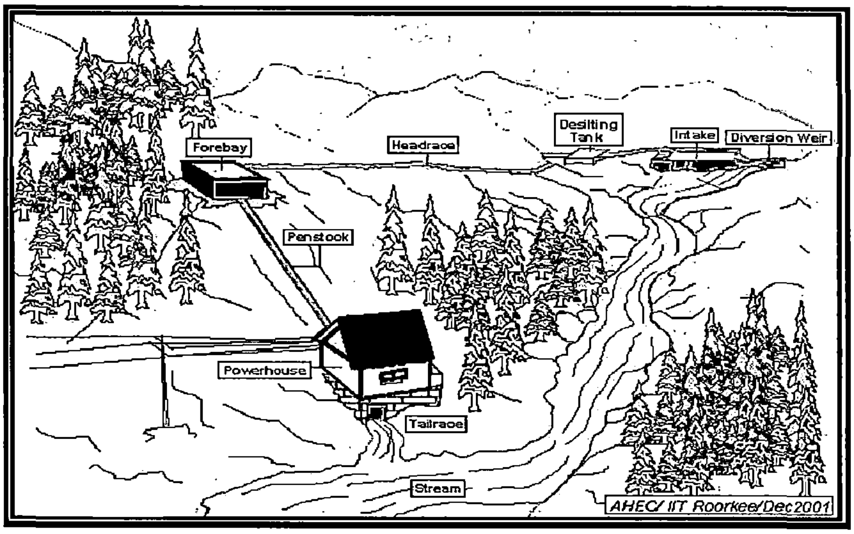


Figure 1.1 : Typical arrangement of runoff river scheme [3]

1.6.2 Canal Based Scheme

Canal based small hydropower scheme is planned to generate power by utilizing the fall in the canal. These schemes may be planned in the canal itself or in the bye pass channel. These are low head and high discharge schemes. These schemes are associated with advantages such as low gestation period, simple layout, no submergence and rehabilitation problems and practically no environmental problems. The typical canal based scheme is shown in Fig. 1.2.

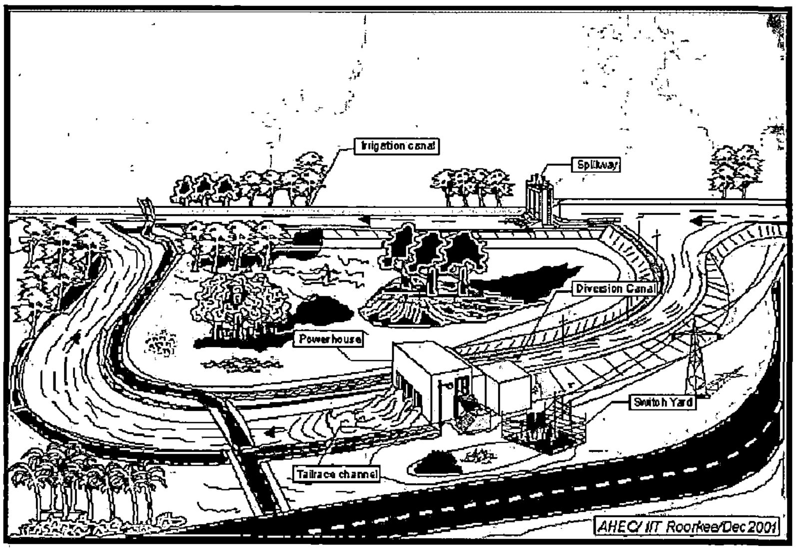


Figure 1.2: Typical arrangement of canal based scheme [3]

1.6.3 Dam Toe Based Scheme

In this case, head is created by raising the water level behind the dam by storing natural flow and the power house is placed at the toe of the dam or along the axis of the dam on either sides. The water is carried to the powerhouse through penstock. Such schemes utilize the head created by the dam and the natural drop in the valley. Typical dam toe based scheme is shown in the Fig. 1.3.

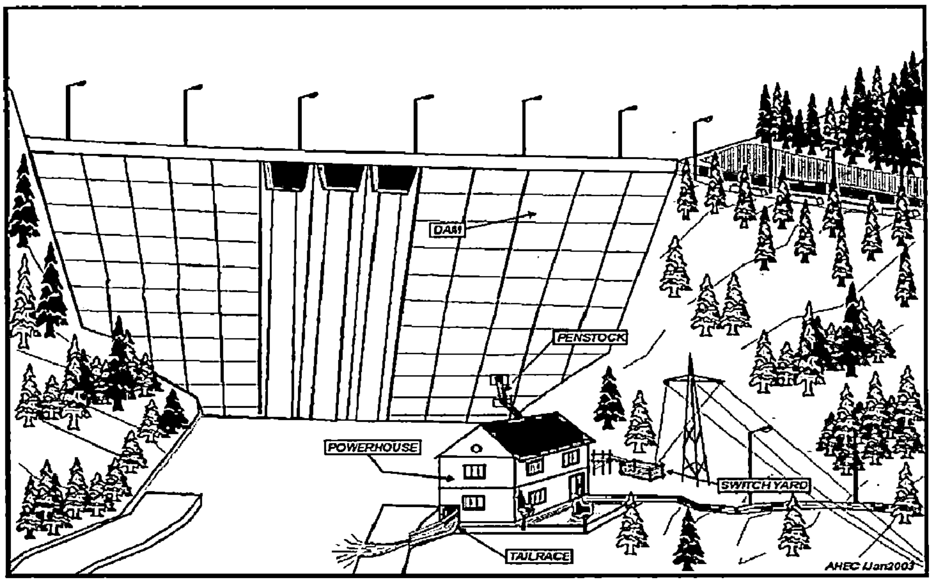


Figure 1.3 : Typical arrangement of dam toe based scheme [3]

1.7 FORMULATION OF HYDRO POWER POTENTIAL

Hydro power is obtained from the potential and kinetic energy of water flowing from a height. The energy contained in the water is converted into electricity by using a turbine coupled to a generator. The hydro power potential of a site is dependent on the discharge and head available at the site. The Power is estimated by the Eq. (1.1)

$$\mathbf{P = 9.81 \times Q \times \mathit{H} \times \eta_o \ kW} \qquad (1)$$

Where, g is the acceleration due to gravity, Q is the discharge in m^3/s, H is head in m, η_o is the overall efficiency of the turbine, generator and gear-box.

The head is relatively constant in run-of-river schemes except for variation in friction losses, with the varying discharge. Whereas, in canal based and dam toe based schemes head also varies depending on water releases and season of release. The design head is so selected that turbine is operated to the maximum time giving optimum energy generation. Energy generation per year is given as

Energy Generation per Year = Power (kW) × Time in hours per year

$$\begin{aligned} &= P \times 24 \times 365 \ \text{kWh/year} \\ &= P \times 8760 \quad \text{kWh/year} \end{aligned} \qquad \textbf{(2)}$$

1.8 DFFERENT COMPONENTS OF SMALL HYDROPOWER

The schematic diagram of Runoff River (ROR) hydro power plant is shown in Fig 1.4.The various components can be categorized in two parts.

A) Civil works components

B) Electro- mechanical equipments

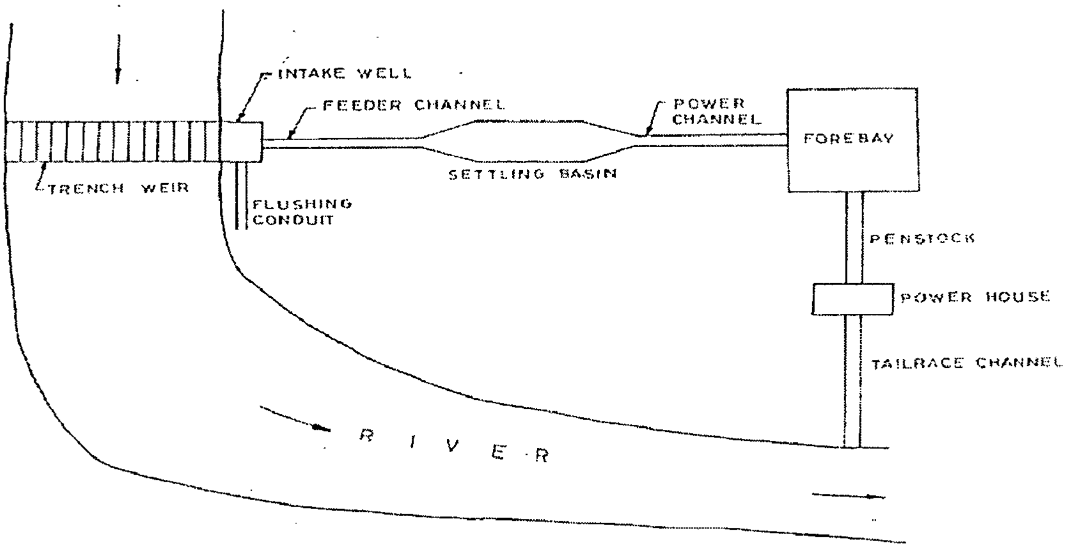


Figure 1.4 : Schematic diagram of ROR plant [6]

1.8.1 Civil Works Components

The purpose of civil work components is to divert the water from stream and convey towards power house. In selecting the layout and types of civil components, due consideration should be given to the requirement for reliability. The various civil components of Runoff river plant are shown in the block diagram in Fig.1.5 and described then after.

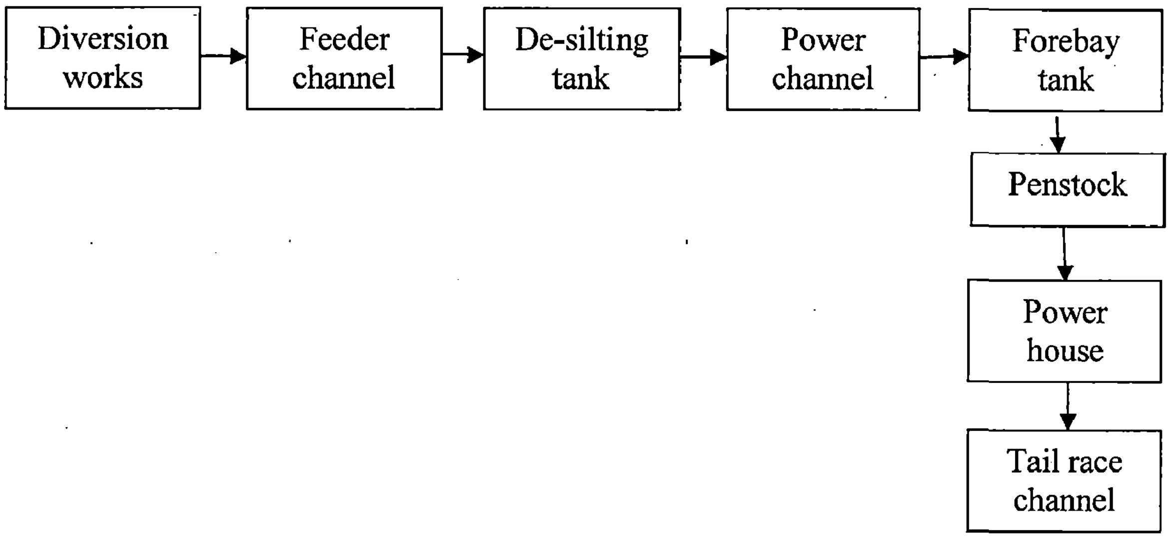


Figure 1.5 : Block diagram of civil components of ROR plant

1.8.1.1 Diversion Works

It's a structure built across a natural stream to divert the water towards the power house for power generation. It may be in the form of barrage or weir, may be gated or non-gated, and may be temporary or permanent. It should be designed in such a manner that [3]

a. It should have a narrow, well defined section.
b. Location should be such that discharge intensity is high.
c. The desired amount of water should be diverted most of the time.

The sediments in water should not be allowed to enter the water intake (as far as possible).

Accumulated objects should be easily flushed downstream. The flow velocity should be controlled to protect the structure from scouring.

1.8.1.2 Feeder Channel and Power Channel (Headrace Channel)

The channel provided between intake and de-silting tank is termed as feeder channel and channel provided between de-silting tank and forebay tank is termed as power channel or head race channel. The primary requirement for a power channel is to have slope as mild as possible with proper conveyance of the discharge. It implies that the resistance of channel to flow should be nil and therefore power channels are invariably lined. The alignment of the channel is usually following the contours and cross drainage works should be avoided as far as possible as they result in head loss. Further, the length of channel should be as short as possible. [3]

1.8.1.3 De-silting Tank (Settling Basin)

The de-silting tank or settling basin is provided between feeder channel and power channel. The water drawn from the river/stream is fed to the turbine, which usually carries suspended particles. These sediments will be composed of hard abrasive materials such as sand, which can cause expensive damage and rapid wear to turbine blades. To remove the suspended particles from the water, flow must be slow down in settling basin so that the silt particles get settled on the floor of the basin. Then, these settled particles are flushed away periodically. [3]

1.8.1.4 Forebay Tank

The forebay tank is provided at the junction of power channel and the penstock, particularly in case of runoff river scheme. It acts as a transition between open flow of water in a power channel and the pressurized flow in a penstock. The forebay [3] can serve the following purposes-

- To provide immediate water demand on starting the generation unit.
- It can serve as a final settling basin where any water borne debris which either passed through the intake or was swept into the channel can be removed before the water passes in to the turbine.
- The forebay provides some storage in case of sudden failure of the system.
- To spill the water in case of sudden shut down or extra water coming to forebay during rains etc.
- It houses the trash rack and penstock.
- Facilitate entry of water in the penstock.
- The location of the forebay is governed by topographical and geological conditions of the site. However, the site of the forebay and power house should be so selected that the penstock has the minimum length.

1.8.1.5 Spillway

The main function of a spillway is to dispose off surplus water from the forebay tank. Design of spillways has a significant effect on the project layout and costs. Its design and capacity depend on capacity of forebay tank, frequency of inflow discharge and geological and other site conditions [3].

1.8.1.6 Penstock

The penstock is the pipe, which conveys water under pressure from the forebay tank to the turbine inlet. The penstock often constitutes a major expense in the total budget and it is therefore worthwhile optimizing the design. The trade-off is between head loss and capital cost. Head loss due to friction in the pipe decreases with increasing pipe diameter. Conversely, pipe costs increase steeply with diameter. Therefore a compromise between cost and performance is required [3].

1.8.1.7 Surge Tank

Surge tank or surge shaft is a reservoir which furnishes space, immediately available for the acceptance or delivery of water to meet the requirements of load changes. It also serves to relieve the water hammer pressure within the penstock, in case of sudden load rejection and sudden load demand. It should always be located as close as possible to the power house in order to reduce the length of penstock to a minimum and preferably on high ground, to reduce the height of surge tank [3].

1.8.1.8 Power House

Power house building for small hydro power stations essentially requires a big hall to accommodate machines (turbines, generators etc.) with sufficient height to accommodate crane operations, and sufficient space for maintenance and control operations. It can be constructed as a steel structure consisting of columns, beams, trusses etc. or it can be reinforced Concrete formed structure with gable frames to accommodate roof (Purlins, sheeting etc.). For remote hilly sites prefab buildings can also be used which are easy to transport and quick to install [3].

1.8.1.9 Draft Tube

The water after doing work on the turbine moves towards the tailrace through a draft tube, which is a concrete tunnel or a riveted steel plate pipe, its cross section gradually increases towards the outlet. The draft tube is a conduit, which connects the runner exit to the tailrace. The tube should be drowned approximately one meter below the lowest tailrace level [7]. The following are the functions of the Draft Tube:

i. The water is discharged freely from the runner; turbine will work under a head equal to the height of the headrace water level above the runner outlet. An airtight draft tube connects the runner to the tailrace; workable head is increased by an amount equal to the height of runner outlet above tailrace.
ii. The draft tube will thus, permit a negative suction head to be established at the runner outlet thus making it possible to install the turbine above the tail race without loss of head.

The water leaving the runner still possesses a high velocity and this kinetic energy would be lost if it is discharged freely as in a Pelton turbine. With the increase in net working head on the turbine, output will also increases, thus raising the efficiency of turbine.

The following are some types of draft tubes used in the Small Hydropower Station

- Straight Divergent Tube
- Moody Spreading Tube
- Simple Elbow Tube
- Elbow Type with a Circular Inlet and a Rectangular outlet section

The Fig. 1.6 shows different types of draft tubes.

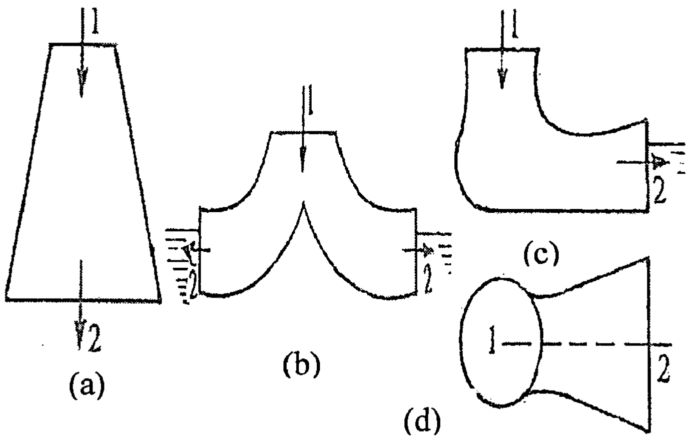


Figure 1.6 : Different types of draft tube (a) Straight divergent tube (b) Moody spreading tube (c) Simple elbow type (d) Elbow type with a circular inlet and a rectangular outlet section [7]

1.8.1.10 Tail Race Channel

From draft tube water enters in the tail race channel. In case of Runoff river plant water is again supplied back to the stream via tailrace channel and similarly, in case of canal based plant water is directed towards the main canal.

1.8.2 Electro Mechanical Equipments

Electro-Mechanical equipments mainly include hydro turbine, generator, speed generator, governor, gates and valves and other auxiliaries.

1.8.2.1 Hydro Turbine

The hydraulic turbine transforms the potential energy of water into mechanical energy in the form of rotation of shaft. It can be broadly classified into two categories according to action of water on moving blades [8].

i. Impulse turbines

ii. Reaction turbines

1.8.2.1.1 Impulse Turbines

In case of impulse turbines the penstock is connected with the nozzle and hence the whole pressure energy of water is transformed into kinetic energy in nozzle only. The water coming out of the nozzle is in the form of a free jet, which strikes with a series of buckets mounted on the periphery of the runner. The water comes in contact with only few of the buckets at a time. Once the water comes out of the nozzle then the pressure is atmospheric throughout hence in case of impulse turbine the casing do not have any hydraulic function to perform but it is necessary only to prevent splashing and to lead the water to the tail race, and also act as a safeguard against accidents. Examples of impulse turbines are Pelton turbine, Turgo- Impulse turbine, Cross flow turbine.

1.8.2.1.2 Reaction Turbines

The water pressure can apply a force on the face of the runner blades, which decreases as it proceeds through the turbine. Turbines that operate in this way are called reaction turbines. It operates with its runner submerged in water. The water before entering the turbine has pressure as well as kinetic energy. All pressure energy is not transformed into kinetic energy as in case of impulse turbine. The moment on the runner is produced by both kinetic and pressure energies. The water leaving the turbine has still some of the pressure as well as the kinetic energy. The pressure at the inlet to the turbine is much higher than the pressure at the outlet. Thus, there is a possibility of water flowing through some passage other than the runner and escape without doing any work. Hence a casing is absolutely essential due to the difference of pressure in reaction turbine. The reaction turbines can be further classified into two main categories based on the direction of flow of water in the runner as follow.

Mixed flow turbine: In this case, water enters from outer periphery of the runner, moves inwards in radial direction and comes out from center in axial direction. Example of mixed flow turbine is Francis turbine.

Axial flow turbines: In this case, water enters from the wicket gates to the runner in the axial direction, moves along the axial direction and comes out in axial direction. Examples of axial flow turbines are: Propeller turbine, Kaplan turbine, Bulb turbine, Straflow turbine.

1.8.2.2 Generator

Generator transforms mechanical energy into electrical energy. Nowadays, only 3 – phase A.C. (alternating current) generators are used in normal practice. There are basically two types of generators

i. Synchronous generator

ii. Induction generator

1.8.2.2.1 Synchronous Generator

The synchronous generator is a rotating machine, generating single or three phase alternating current (A.C.) with a frequency proportional to its rotational speed. It consists of a stationary member called stator comprising windings and a rotating member called rotor, containing magnetic field. The rotor may be a permanent magnet (for very small output) or more conveniently an electro magnet whose coils or windings are fed by an (external) D.C. source called excitation. The rotating magnetic field (created by the rotor) induces voltage and current in the stationary windings. The generator voltage at constant frequency the speed of the rotor must be kept at synchronous speed. The synchronous speed is given as:

Synchronous speed Ns = 120 × f/ P' **(Equation 3)**

Where, Ns is speed in R.P.M, f is the frequency in Hz and P' is the no of poles.

1.8.2.2.2 Induction Generator

The Induction generators are also called as asynchronous generator based on the super synchronous speed at which it operates. Induction generator, basically consist of a stationary winding called stator, enclosed by the machine frame and a rotor with a short circuited winding. Placing a 3-phase A.C. current on the terminal of the 3-phase stator winding, creates a rotating magnetic field in the machine which rotates at a speed, called synchronous speed n_s, depending on the supply frequency and the no. of poles. The rotating field flux cuts the short circuited rotor winding where it induces voltage and current, which in turn produce torque on the rotor. The rotor must always rotate below or above the synchronous speed i.e. at a slip. Otherwise there is no cutting of flux by the rotor conductors and hence no torque is developed. The induction machine operates as a motor when running below synchronous speed and as a generator when the rotor is above synchronous speed. Therefore any induction motor may also be used as a generator, simply by driving at above synchronous speed. The difference between the synchronous speed n_s and the rotor speed n_r is called slip speed n_g, represents the speed of the rotating field viewed from the rotor.

Slip speed $n_g = n_s - n_r$ (Equation 4)

Slip $S = (n_s - n_r) / n_s$ (Equation 5)

If Slip S is negative, i.e. rotor speed is more than synchronous speed, the machine operates as generator. If Slip S is positive, i.e. rotor speed is less than synchronous speed, the machine operates as motor.

1.9 TURBINE-CRITICAL PART OF POWER PLANT

As such, there are various components which come in contact with water, but turbine is the most critical component because it has a considerable influence on the cost of civil works, electrical component as well as overall performance of any hydro power project. Therefore turbine becomes an important part of the power plant from economic point of view as well as for proper running of the power plant.

1.10 CLASSIFICATION OF TURBINES BASED ON HEAD

The capacity of a small hydropower plant is decided by available head (difference between the head water level on upstream side and tail water level on downstream side) and the water discharge available for the scheme. Normally for conventional schemes like run off river or dam based schemes, the head available for the scheme, is moderate, and generally more than about 4 m and in case of hilly regions, the head available is substantially higher often in range of 100 m-300 m. In all such cases, conventional type of turbine such as Kaplan, Francis, Pelton are installed depending upon head as indicated below

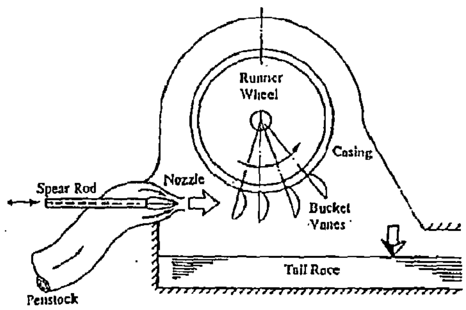
- For high head (Above 200 m) – Pelton
- For medium head (30-200 m)-Francis
- For low head (up to30 m)-Kaplan, Tubular, Bulb, Strawflow

1.10.1 High and Medium Head Turbines

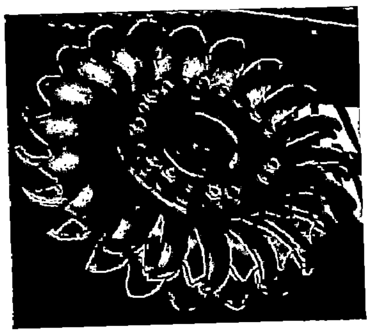
1.10.1.1 Pelton Turbine

It is a high head, free jet, tangential flow impulse turbine. The runner of a Pelton turbine consists of a circular disc on the periphery of which a number of buckets are attached. Which is shown in Fig.7 (a) and (b).The shape of the bucket is of a double hemispherical cup or bowl. Each bucket is divided into two symmetrical parts by a dividing wall which is known as splitter. The jet of water strikes on the splitter of a single bucket at a time, in tangential direction to the periphery of runner. Some of the important characteristics of Pelton turbine are [9]

- It has low specific speed. (Ns = 12 to 30)
- It is generally used for high head and low discharge run off river schemes.
- Chances for the formation of cavitations are negligible.
- It can be installed above tail race level as it operates at atmospheric pressure.
- Horizontal Pelton turbine has negligible axial thrust hence thrust bearings are not required



(a)



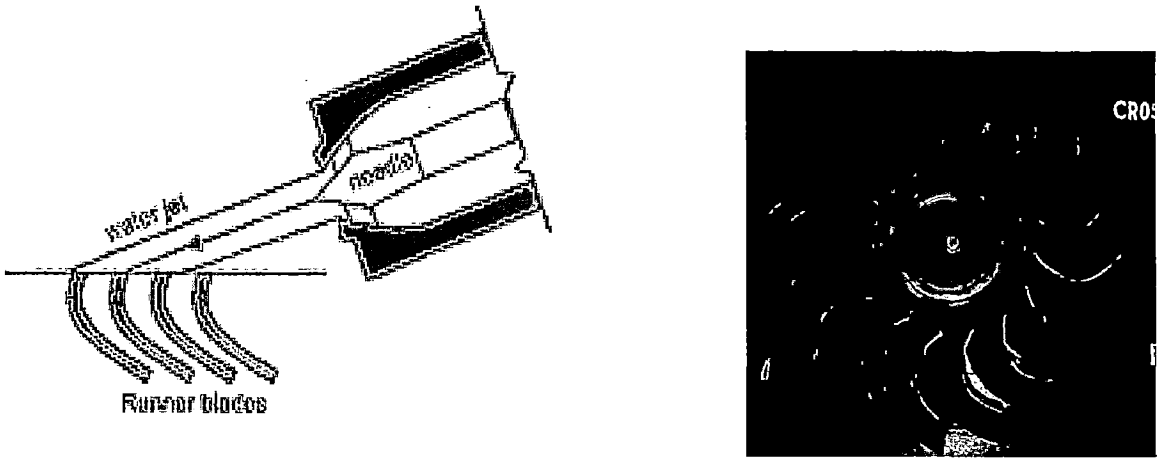
(b)

Figure 1.7 : (a) Main parts of pelton turbine (b) Runner of pelton turbine

1.10.1.2 Turgo- Impulse Turbine

Like Pelton turbine, Turgo-impulse turbine is also an Impulse turbine. However, the shape of the bucket is such that the jet of water strikes the plane of its runner at an angle of around 20° rather than remaining within the same plane. This is shown in Fig.8 (a). In case of Turgo-Impulse turbine the water jet can strike more than one bucket (generally 2 to 3) at a time, which is not possible in Pelton turbine [9]

- It has higher range of specific speed (Ns = 20 to 70) than Pelton turbine.
- Compared to reaction turbines, it is less susceptible to sand erosion.
- Similar to Pelton turbine, it can also be installed above tail race level as it operates at atmospheric pressure.



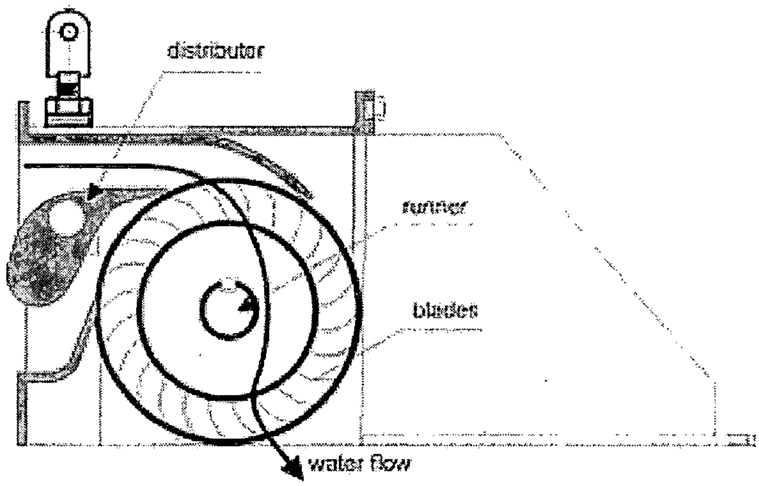
(a) (b)

Figure 1.8 : (a) Position of jet with respect to runner blades (b) Runner of turgo-impulse turbine

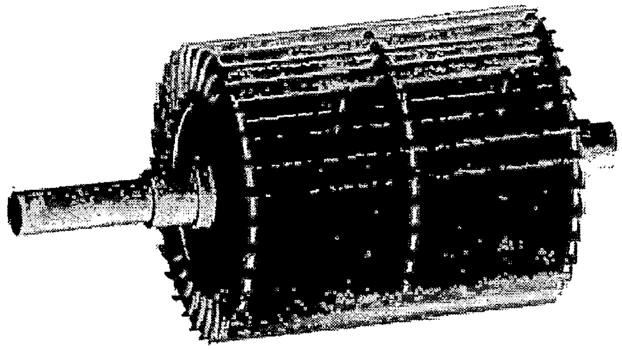
1.10.1.3 Cross flow turbine

Basically cross flow turbine is an impulse turbine, i.e. all the pressure energy is converted into kinetic energy in the nozzle itself. However, as the gap between the nozzle and the runner is very small, the pressure near the outlet of the nozzle is higher than atmospheric pressure. Main parts and runner is shown in Fig.9 (a) and (b).Therefore a small portion of energy is in the form of reaction [9]. Because of the symmetry of the blades the length of the buckets can be increased up to any desired value and hence the flow rate.

- Construction is simple and easy to fabricate.
- It has specific speed range from 20 to 80 (metric).
- Operation is convenient and maintenance is relatively easy.



(a)



(b)

Figure 1.9 : (a) Main parts of cross flow turbine (b) Runner of cross flow turbine

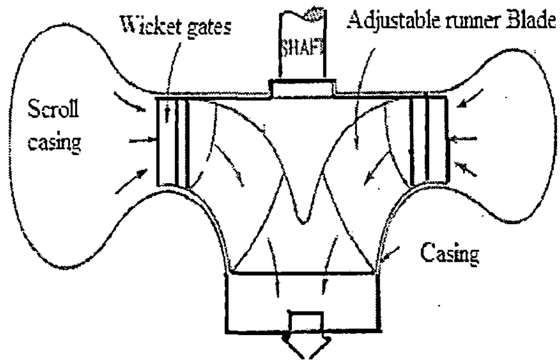
1.10.1.4 Francis Turbine

Francis turbine is an inward mixed flow reaction turbine in which water enters in the runner from the guide vanes towards the centre in radial direction and comes out of the runner in axial direction. Main parts and runner of Francis turbine is shown in Fig.1.10 (a) and (b). It operates under medium heads and also requires medium discharge. Some of the important characteristics of Francis turbine are [9]

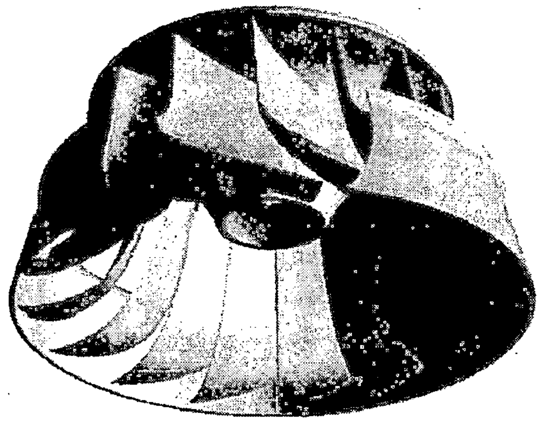
The hydraulic characteristics of its runner are such that operating speeds are lower than comparable Propeller turbines hence the physical size of the Francis turbine is bigger than the Propeller turbine, for the same operating conditions.

The efficiency of the Francis turbine lies in between the Propeller and the Kaplan turbines. The Francis turbine can have both vertical and horizontal settings; generally the vertical units are cheaper, especially when the throat diameter of the turbine is about 1.2 m. For low head applications it can have open flume setting. It has higher specific speed range (80 to 400) than Impulse turbines. It has higher peak efficiency than Impulse turbines but has poor part load efficiency and is not suitable where the fluctuation in discharge is very high.

The turbine can be installed above or below tail race without loss of head.



(a)



(b)

Figure 1.10: Main parts of Francis turbine (b) Runner of Francis turbine

1.10.2 Low Head Turbines

In general turbines used in low head hydro are similar to that of large hydro but the configurations is different from that because low head hydro power plant has to handle more amount of discharge for the same amount of power to generated. For this we need special designed turbines for the low head power plants.

Following are the turbines which are used for low head sites.

1.10.2.1 Vertical Siphon Intake Kaplan Turbine

The construction features of this turbine is quite similar to the basic Kaplan type of turbine with adjustable blades and guide vanes except that the intake is of siphon type and a vacuum breaking / air admission device is used. Layout of turbine is shown in Fig.1.11. In this turbine centre line of spiral casing is kept above the head water level (HWL). During starting of the machine, a vacuum pump creates vacuum in the zone above the HWL of the spiral casing, which raises the water level so that siphon action is initiated and water flow is established, and the turbine starts rotating.

For unit shutdown, atmospheric air is admitted in the zone above HWL of the spiral casing in order to break the siphon action so that water flow is stopped and the unit is stopped. For emergency purpose no turbine inlet valve or an electric motor operated gate is provided as incase of conventional turbine. Breaking of siphon will stop the unit in case of an emergency. However a stoplog gate is provided for maintenance purpose. As the turbine runner and other parts of turbine are much above the tail water level, tailrace gate is also eliminated in this turbine. Elimination of gates results in reduction in cost of the project.

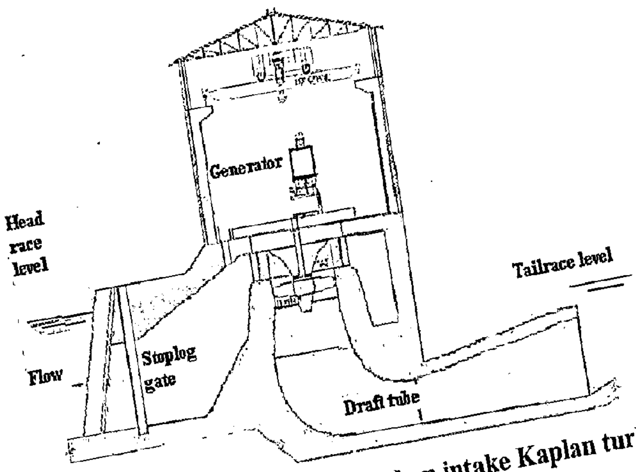


Figure 1.11: Layout of vertical siphon intake Kaplan turbine [3]

1.10.2.2 Inclined Axis, Low Head Kaplan Gear Turbine

The turbine axis is usually inclined at an angle of 15 to 45 degrees to the horizontal. It is primarily intended for use in very low head sites, where the net head is between 2 and 8 meters. Maximum unit capacity is about 2.6MW. The bevel gear bulb turbine has a bulb within the water passage with turbine thrust bearing and right angle gear drive; stay vanes and wicket gates; Kaplan runner and bent cone draft tube. The generator is mounted above the right angle gearbox. A typical layout is given in Fig 1.12.

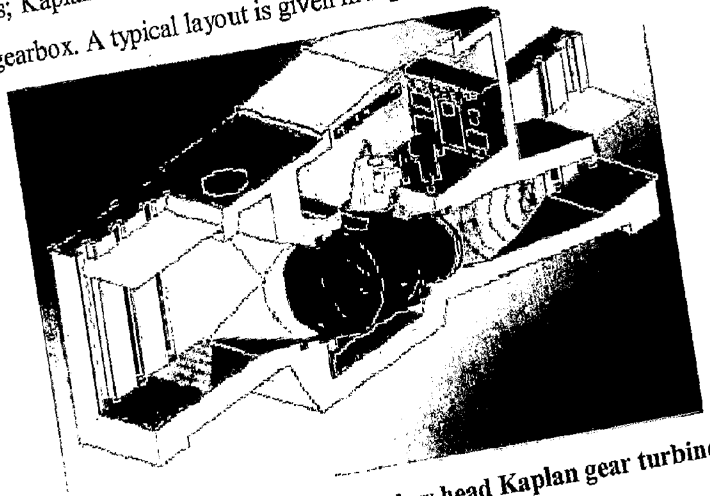


Figure 1.12 Illustration of inclined axis, low head Kaplan gear turbine unit [10]

1.10.2.3 Horizontal Axis Bulb or Pit Kaplan Turbine

The generator is contained within either a bulb or a pit within the upstream water passage. A pit installation has an open-topped bulb, permitting far easier access to the generator. To keep the generator size within reason, there has to be a gear unit to increase generator speed to between 600 and 1000 rpm. A typical installation is shown in Fig.1.13. Downstream of the bulb or pit there are the stay vanes, wicket gates, Kaplan runner and finally a conical draft tube. The runner shaft is usually set about one runner diameter below minimum tail water. As noted in Fig.1.13, there is easy access around the turbine unit for maintenance. Also, as with all horizontal shaft units, the runner can be removed for maintenance without removing the generator. Bulb units are only available in the larger diameters, and should be avoided due to the "confined access" problems associated with the bulb. Pit type units are far preferable.

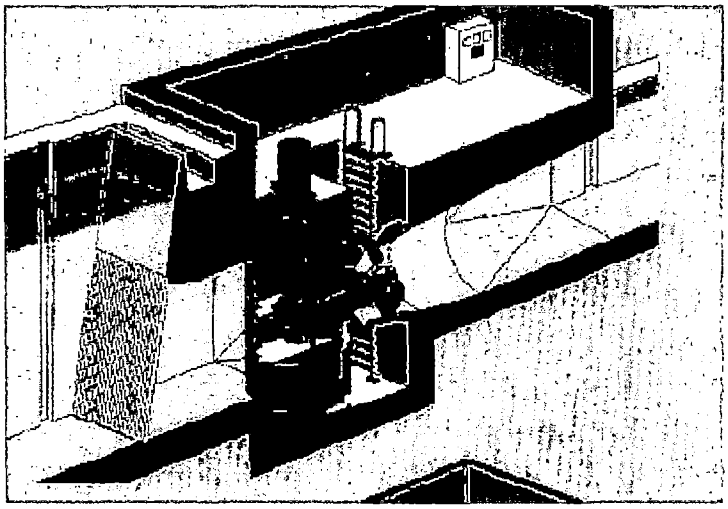


Figure 1.131: Horizontal axis pit Kaplan turbine with right-angle gear driven generator [10]

1.10.2.4 Horizontal Axis "S" Type Kaplan Turbine

This is the most common type of low-head, small hydro plant arrangement. Runner sizes range from 1.0m up to about 4m, heads from 5m to 25m, with power output up to about 12MW. There is a small upstream bulb containing controls for the Kaplan blades, and the thrust bearing. The bulb is held in place by the stay vanes. Immediately downstream is the distributor ring with the wicket gates. The runner is contained within a horizontally split throat ring, which can be removed for access to the runner. Downstream there is a long shaft to the draft tube gland and the turbine/generator guide bearing. After the bearing there could be a speed increaser or a direct connection to the generator. A typical layout is shown in Fig1.14.

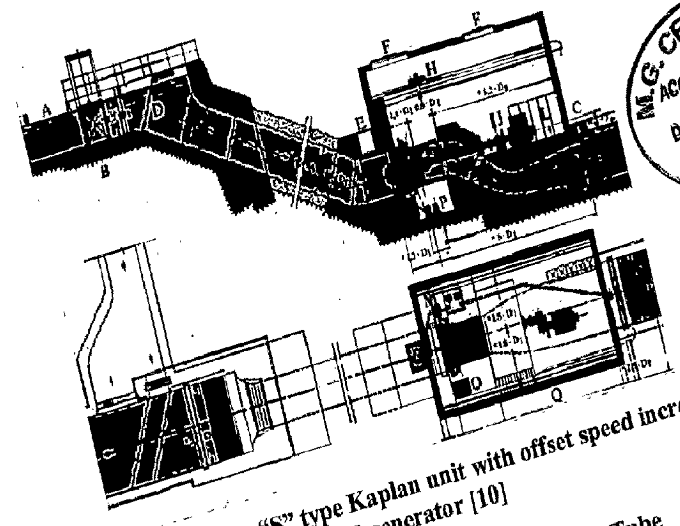
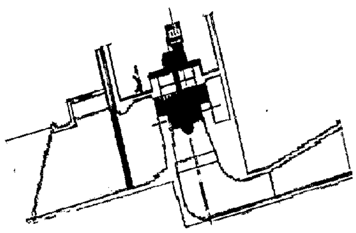


Figure 1.14: Horizontal axis "S" type Kaplan unit with offset speed increaser and high speed generator [10]

1.10.2.5 Vertical Axis Small Kaplan Turbine with Elbow Draft Tube

There are many arrangements for small vertical axis Kaplan units. They are set with the runner well above normal tail water level as shown in Fig.15, thus avoiding the use of draft tube gates. Access to the turbine head cover is restricted, with insufficient headroom. This is the main limitation with this arrangement, and if possible, it is prudent to increase the height of the generator plinth and length of turbine shaft to provide more room for maintenance access. Also, removal of the turbine runner requires removal of the generator and speed increaser, at added cost. For this reason, preference should be given to arrangements with a horizontal shaft, using an "S" unit.



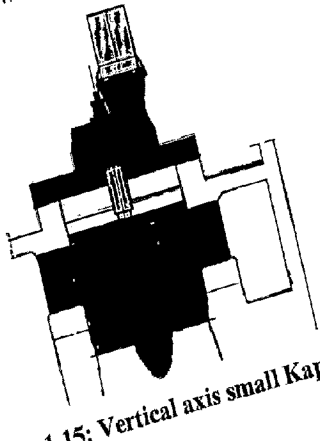


Figure 1.15: Vertical axis small Kaplan unit with offset speed increaser and high-speed generator [10]

1.10.2.6 Horizontal Axis "S" Type Propeller Turbine

This type is identical to the horizontal axis "S" type Kaplan turbine, with the exception of the runner, which is a fixed blade propeller. This turbine has a very peaked efficiency curve, and is only suitable for sites where there is minimal change in head and flow. For this reason, very few are installed, preference being given to the Kaplan alternative. The typical arrangement of Horizontal axis "S" type propeller turbine will be as shown in the Fig.1.16.

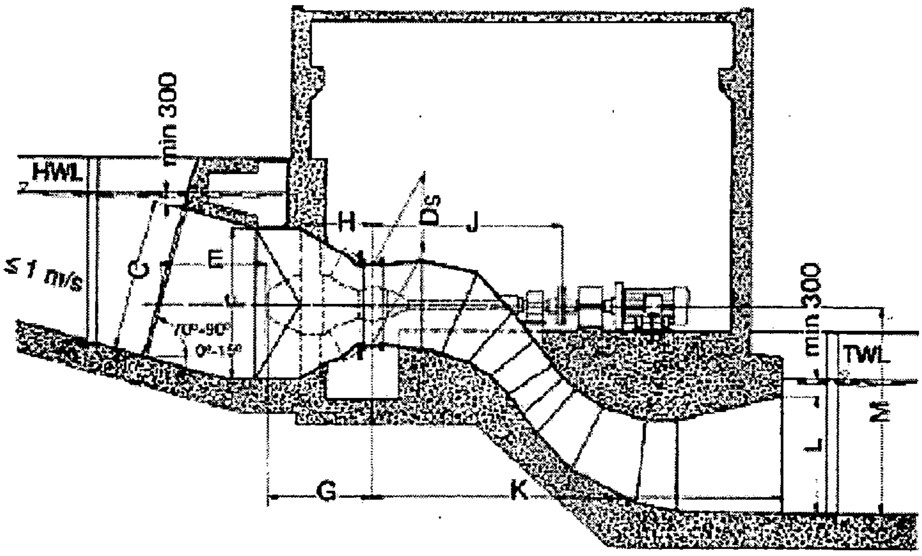


Figure 1.16: Horizontal axis "S" propeller turbine in low head setting geared generator [10]

1.10.2.7 Vertical Axis "Saxo" Axial Flow Kaplan Turbine

This unit has an abrupt bend just upstream of the turbine distributor, as shown in Fig.1.17. Due to the close proximity of the bend to the distributor, and abruptness of the bend, there is a substantial hydraulic loss at the inlet pipe (Bennett). This results in reduction of turbine efficiency as flow increases.

The unit is named after the shape of the water passage, which resembles a saxophone. Due to the presence of the inlet pipe above the turbine, arrangements should be made to remove the turbine runner from below. Also, due to the long shaft above the turbine, substantial headroom above the generator is needed to install the unit. Very few of these units have been built.

1.10.2.8 Bulb Turbine

The Bulb turbine is a type of reaction turbine which is used for the lower heads. It is characterized by having the essential turbine components as well as the generator inside a bulb, from which the name is developed. A main difference from the Kaplan turbine is

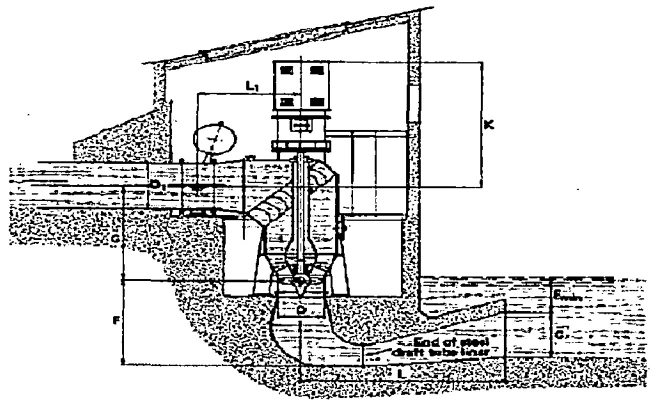


Figure 1.17: Vertical axis "Saxo" axial flow Kaplan Turbine [10]

moreover that the water flows with a mixed axial-radial direction into the guide vane cascade and not through a scroll casing. The guide vane spindles are inclined (normally 60°) in relation to the turbine shaft. Compared to other turbines, this result in a conical guide vane cascade. Layout of bulb turbine is shown in Fig.1.18.

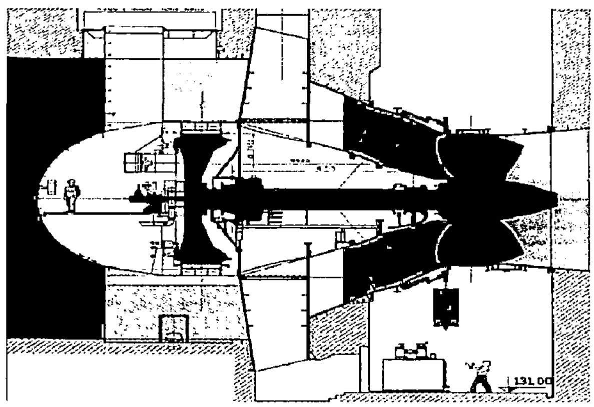


Figure 1.18: Bulb Turbine [11]

1.11 BULB TURBINE IS ADVANTAGEOUS

Bulb turbine has many advantages compared to the other turbines used for similar site conditions. The straight-lined design of the turbine inlet permits a reduction of the distance between the units of above 30 percent and thus saving in the width of the power plant. Also the specific speed increase reaches at least 20%. This means that for a given head and output the use of a bulb turbine unit allows a turbine diameter about 10 % smaller than with a conventional turbine [12].

Besides all above advantages bulb turbines have well adopted existing technologies. In India as well as around the world a lot of bulb turbine installations are there.

CHAPTER-2
LITERATURE REVIEW

2.1 OVERVIEW

A lot of work has been done in the past on technical and operational aspects of bulb turbines and thus a lot of literature is available in journals and other publications. Some of the important literature relating to this work is presented as follows.

2.2 LITERATURE REVIEW

Bulb turbine is well suited for low head hydro schemes. Many investigators have carried out analysis of different aspects of bulb turbine like improving the efficiency and reliability, study of sluicing and reverse operations in bulb turbine and study of cavitation in bulb turbine.

Arshenevskii et al. [13] conducted research on pressure in the runner zone of a hydroelectric station with bulb units. As to ensure reliable operation of a unit it is important to have information on the pressures of the flow before and after the runner, since they permit judging the possibility of discontinuity of the flow, establishing the places of installing vacuum breakers, and estimating the magnitude of the axial hydrodynamic forces acting on the turbine if they cannot be measured directly. Author investigated radial distribution of pressure for bulb turbine and vertical turbine. They found that the pressure in the space before the runner of turbines of bulb units is mainly a function of the reduced discharge and opening of the gate apparatus. They also showed that the non-uniformity of the pressure distribution in the cross section before the runner of a bulb turbine is less than that of a vertical turbine of the same type for the same regimes. They proposed that the high-vacuum zone during the transient process reaches a critical value, which can be avoided by assigning an appropriate regulating regime.

Natartus et al. [14] investigated the reversible regimes. There investigations have shown that when ordinary bulb turbines are used in pump regimes, there occurs a divergence in the optima of efficiencies with respect to the reduced rotational speed, which decreases with decrease of the specific speed. Also the results of experiments make it possible to determine the values of the main parameters during reversible operation of a horizontal bulb unit. They added that in pump regimes the reduced axial force considerably exceeds the corresponding values in the turbine zone of the universal characteristic; with a position of the regulating devices corresponding to the maximum efficiency in the pump regime it is minimal for the given type of runner. The data obtained during investigation can be used when designing reversible bulb units and in calculations of transients end optimization of regulating regimes.

Arshenevskii et al. [15] carried out the work in improving the operating reliability of bulb turbine. They studied some of the most important components of bulb-type turbine units those are the heel and the counter heel. These are responsible for transmitting the axial hydrodynamic forces. The axial forces are found by experimental methods, since until now cases of reliable determination of these forces by theoretical methods are unknown. An axial force directed toward the downstream side is transmitted by the heel and is considered to be positive; a negative axial force directed toward the upstream side is transmitted by the counter heel, which should be designed for the maximum value of this force. They conducted the study for the variation of axial hydrodynamic force according to various specific speed and discharge. With the help their study one can determine the loads, which are transmitted by the heel and the counter heel in a bulb unit. Further their results indicate that the value of the equivalent axial hydrodynamic force depends basically upon the opening of the turbine control and upon the equivalent discharge.

The type of runner and the speed exert an indirect effect, since they affect the variation of discharge, in this connection, the obtained data can be used for preliminary analyses of the operating regimes of bulb units with similar geometrical designs of the internal passages, but with different types of runners. They also concluded that in the absence of the possibility of direct measurement of the axial hydrodynamic force under steady regimes, this force can be calculated with a departure of 10 to15% from the true value, by using the results of the pressure measurements along the radius on the turbine cover or directly inside the flow in the runner zone. The results of the tests for vertical turbines cannot be used directly for bulb units, since the differences in the shapes of the internal passages and the construction of the turbine control lead to an appreciable variation in the pressure distribution and in the value of the axial forces even for runners of the same type.

Samorukov et al. [16] developed the method of improving the output and cavitation properties of hydraulic turbine equipment of low-head hydroelectric stations. They conducted investigations for the purpose of optimizing the design parameters of hydraulic turbines, which determine the dimensions of the block and operating indices of the units for greater output. They used modern statistical experimental design and concluded that the data obtained should be used when modernizing turbines at existing low-head stations They also proposed to create improved and more economical large high-speed turbines and to determine the conditions of applicability of vertical and horizontal bulb units, it is required to conduct comprehensive investigations for optimization and technoeconomic comparison of units with consideration of all factors determining the reliability and economy of the hydro development as a whole and operating stability of the power system.

Ferr et al. [17] presented a paper on flow field due to a row of vortex and source lines spanning a conical annular duct .They derived analytical expressions for the incompressible flow due to a row of vortex and source lines spanning a duct whose walls are coaxial conical surfaces of revolution with a common vertex. The expressions are used in an example to obtain numerical results by a panel method for the velocity distribution of the flow about the inlet guide vane system of a water turbine of bulb type.

Kuppuswamy et al. [18] carried out case study on deformation of outer distributor cone in bulb turbine due to cavitation .In this case study they reported deformation of outer distributor cone in the bulb turbine due to cavitation. They found that top and bottom sides of the outer distributor cone enlarged in size from the initial dimensions at the time of erection of turbine. The left and right hand side dimensions reduced from the erection data and the geometry of outer distributor cone became elliptical after operating machine for 28,000h.In this study they also derived equations, which are useful to predict the condition of the outer distributor cone after a specified period of operation. Results showed that their approach is very efficient and solutions are very accurate. The developed approach proved to be very helpful to bulb turbine design engineers in both design and modification of outer distributor cone.

Rudramoorthy et al. [19] presented a paper on distortion of guide vane assembly in a bulb turbine due to cavitation and reverse water flow. This paper focused on the problems involved in a deep setting bulb turbine in a hydro- electric power plant and possible solutions to avoid cavitation. Analysis is carried out based on the measured data on the distortion in the guide vane assembly. To eliminate cavitation and distortion in these turbines, suggestions were evolved between the deep setting and reverse tailrace water flow into the turbine assembly.

Paine et al. [20] described the rehabilitation of Idaho falls hydroelectric projects. Idaho Falls is located in Southeast Idaho approximately two hours southwest of Yellowstone Park. Previously vertical shaft turbines were there but due to dam failure Idaho falls hydroelectric projects damaged. Author discussed that in rehabilitation of these project bulb turbine technology was compared carefully with conventional vertical shaft and horizontal shaft technology. Key factors in the selection of bulb turbine for this application were

The units operate at efficiencies equal to or better than conventional vertical shaft systems over a relatively large flow range. Because of their straight flow path, the turbine uses a runner having smaller diameter than that required to produce an equivalent amount of power with vertical turbines. This allowed smaller powerhouse, which resulted in saving of $ 1 million per plant. No speed increasing system is required as it is common in other technologies.

Author discussed that these three old hydroelectric sites was well suited to the use of bulb turbines. Their application resulted in an estimated savings of $3' million in civil structure costs because of their small physical size.

Maqauerpeter et al.[21]presented paper on bulb turbine design.He Selected design details and calculation methods are discussed on the example of the Igarapava project, which is presently under construction and represents the first power plant with large bulb turbines in Latin America. The bulb turbine concept of the nineties, which is presented in this paper, is in its conceptual approach based on the experience gained in the industrialized countries throughout the last two decades.

Matsui et al. [22] discussed the effect of pier at draft tube outlet on efficiency performance of bulb turbine. The authors provided 3 piers with different configurations and investigated their effect on turbine efficiency, mounting them separately on identical model turbines. In order to study the effect of piers, both turbine efficiency and wall pressure at the draft tube inlet and outlet were measured. The results indicated that efficiency deterioration due to mounting the pier is slight. However, the effect of the draft tube outlet area ratios has a significant influence

Tsukamoto et al. [23] studied the Sluicing operation. The elimination of spillway has the advantage of reducing the civil cost and making the turbine installation space easily available. For this reason, the surplus water is sometimes discharged from the headwater to the tail water through turbine of Bulb turbine. In sluicing, during flood, head decreases up to great extent. Therefore author considered a method to eliminate spillway, where the rotating speed is force-fully increased up to the rated speed so as to enlarge the turbine discharge (motoring operation). The authors conducted the experiments on the motoring operation of model bulb turbine. He found unique phenomenon between rotating speed and guide vane angle at which the stagnation of discharge appears.

Sunaga et al. [24] introduced recent technology of large capacity bulb turbine-generators. The authors discussed the hydraulic performance, support structure, bulb construction, bearing arrangement, welded guide vanes, fin cooling, outer surface direct cooling, and motor driven fan speed control.

Shiers et al. [25] compared the relative advantages of pit versus bulb hydroelectric turbine-generators using, as an example, an evaluation performed to determine the appropriate equipment for addition to an underdeveloped run-of-river hydro site. The additional capacity was 15 MW, which was operated at the low end of the flow duration curve. The objective covered in the paper was the evaluation of the open-pit and bulb type turbine equipment for the given site conditions. The evaluation concludes that the installation of two units of 7. 5 MW each open-pit turbine-generators was the most appropriate solution for this site development.

Ueda et al. [26] describes the main features of the 34 MW large capacity bulb turbine and generator which is now operating at the Akao Power Station of the Kansai Electric Power Company and the two unit of 16.8 MW each bulb turbines and generators which are operating in the Sakuma No.2 Power Station of the Electric Power Development Company. Other units under manufacture were also described.

Kubota et al. [27] proposed a method which can predict the cavitation phenomena and its altitudinal distribution in the ultra-low head large bulb turbine, whose runner diameter exceeds the net head, using the model test results and the three-dimensional flow analysis through the runner. Also, a new conversion method of cavitation performances was proposed taking the altitudinal distribution of cavitations into consideration. This conversion method, taking the altitudinal distribution of cavitation development into account, can be applied to the conversion of cavitating runaway speed characteristics of ultra-low head large bulb turbine.

Chen et al. [28] analysed the transient flow in units and downstream channel for power station of bulb turbine. Aiming at the characteristics of low head and large discharge of power station of bulb tubular type turbine, the station unit's boundary equations and the boundary equations between draft tube outlet and downstream channel inlet were established. They used the mathematical model for simulating the transient flows in units and downstream channel after load rejection. The model realized an unified analysis of hydraulics and mechanics, pressure flow and unconfined flow in study of transient flows in hydroelectric station. They showed that when unsteady flow in downstream channel is simulated, the maximal rising rate of unit's rotate speed and the maximal water pressure in front of wicket gates increases by 3% and 4.2%, and the minimal water pressure at the draft tube intake reduces 31%, compared with assuming that water level of draft tube outlet is constant; the maximal positive and opposite axial hydraulic thrusts are little affected by the unsteady flow.

Strohmer et al. [29] showed Comparison between the computed stresses and natural frequencies of a large bulb turbine and the values measured on a prototype unit. While the commissioning of a large bulb turbine, they performed a number of measurements for analyzing the vibration behavior of the system and for determining the frequency reducing effect of the surrounding water as well as for determining the loadings on the main components. These specific measurements were performed under stationary and transient operating conditions. In this paper authors discussed measurements and computation model and some selected measuring results are shown. The comparison with computed values is presented. Their results shows that if these measurements agree very well with the assumptions underlying the computation and, together with the experience gained during operation, serve as a basis for a reliable design of a large bulb turbine unit.

Strohmer [30] analyzed the dynamic behavior of a bulb turbine by determining the natural frequencies and the natural vibration modes. They describe design features and the computation model is derived. They discussed Assumptions for the computation and the influence of various parameters on the vibration behavior of the system. The frequency reducing effect of the surrounding water was considered. They measured the frequency of the vibration in the horizontal plane at a prototype bulb turbine and a good correspondence with the computed value was found.

Bolton [31] described that Idaho Falls municipality is replacing existing vertical shaft turbines at its three hydro plants with bulb turbines to take advantage of this smaller, efficient and less costly hydro system. He also concluded that higher costs of other energy sources make low head hydro plants an increasingly attractive alternative.

Kuppuswamy et al. [32] carried out finite element analysis (FEA) using ANSYS software to find out structural stability with stress distribution for bulb turbine and the reasons for high vibration and vertical cracks in the concrete structure of the powerhouse. They analyses the structural stability of stay column junction at top & bottom. Finite element analysis shows that the bulb turbine operation is safe when the stay columns are fitted with pipe jacks. This method reduces stress concentration at the junction of slay column with concrete.

Jian. [33] discussed and stated that a hydroelectric bulb turbine set has the advantages of high technical parameters, excellent operating characteristics, high reliability and cost savings in civil construction, and have recently been widely used in low head hydropower developments. This paper described the local control unit design of the bulb turbine set at the Mowu hydroelectric power plant in the Fujian province.

Velensek et al. [34] presented the research and development work of a standardized bulb turbine for a chain of hydropower plants on the Sava and the Mura Rivers. The procedure of mathematical optimization of the runner from the energy and cavitation viewpoint is presented as well as he determined the losses in the flow section at the inlet and outlet of the power plant.

Vones [35] discussed the comparison between power plants with bulb turbine generating sets and those equipped with conventional vertical shaft power units. The essentials and consequential design requirements for bulb-type turbine generators are outlined. The problems related to the cooling, bearings and design particulars are dealt with at length.

2.3 OBJECTIVE OF PRESENT STUDY

Bulb Turbine, a type of Kaplan hydro turbine, is considered to be one of the most appropriate turbines for getting better efficiencies at lower head. It is characterized by having the essential turbine components as well as the generator inside a bulb, from which the name is developed. It has the advantage of a straight inflow to the distributor and runner and also the outflow through the conical draft tube has no bend. This results in a minimum of losses in the entire application range and therefore higher efficiencies.

The review of literature given above shows that a lot of research work had been carried out on bulb turbines such as to improve the cavitation properties, to ensure reliable operation of a unit it is important to have information on the pressures of the flow before and after the runner, to develop the coating for protecting blade tip, the effect of pier at draft tube outlet on efficiency performance of bulb turbine, investigation of characteristics of bulb turbine during reversible regimes and study on structural stability of stay column and concrete structure. Not much work has been done on design of intake flow passages for bulb turbines to optimize the civil structure cost. Simplifying the design and construction of bulb turbine intakes could lead to cost savings in both material and labor.

The objective of the present work is to study the technical aspects of bulb turbine and investigate possible simplifications and reduction in the length of the intake flow passages for bulb turbines. The head losses associated with these simplifications in the suggested intakes were worked out. Analysis of savings in the cost due to change in intake were made,

CHAPTER-3
BULB TURBINE

3.1 INTRODUCTION

The Bulb turbine is a type of reaction turbine which is used for the lower heads. It is characterized by having the essential turbine components as well as the generator inside a bulb, from which the name is developed. A main difference from the Kaplan turbine is moreover that the water flows with a mixed axial-radial direction into the guide vane cascade and not through a scroll casing. The guide vane spindles are inclined (normally 60°) in relation to the turbine shaft.

The Bulb turbine runner is of the same design as for the Kaplan turbine, and it may also have different numbers of blades depending on the head and water flow. In Vertical Kaplan turbines, the axis is vertical and thus there is hardly any pressure on bearings. Therefore, their failure rate is low. Their cost is also comparable to that of Horizontal turbines. The minus point: they need a spiral development and direction of flow changes many times thus resulting in slightly higher diameter of the runner. The table given below shows some guidelines for the selection of turbines:

Table 3.1: Type of low head turbine

1.	Head	1.5-15m	Tubular
2	Head	5-24 m	Bulb
3	Head	15-60 m	Kaplan

The table 3.1 shows some overlapping of head ranges. This range enables us to choose the better of the two turbines after analyzing other related factors. Thus, Bulb turbines and Vertical Kaplan turbines are always competing stiffly. The choice rests on individual site conditions. In addition to cost, efficiency and output, an important point that should be kept in view is the post-installation performance of these turbines

3.2 GENERAL ARRANGEMENT

A general arrangement of a Bulb turbine plant is shown in Fig.2.1with a vertical section through the unit. The water flows axially towards the unit in the centre of the water conduit and passes the generator, the main stays, guide vanes, runner and draft tube into tailrace channel. These turbines generally used where discharge to be handled in large quantity under relatively low heads.

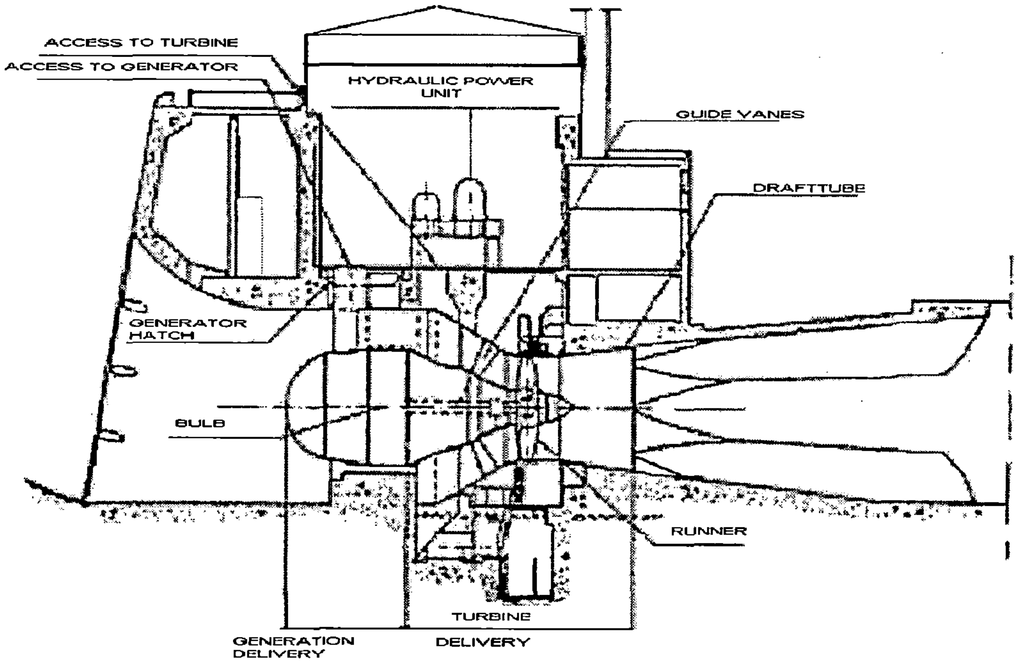


Figure 3.1: Bulb turbine general arrangement [36]

3.3 MAIN COMPONENTS

The Bulb turbine consists of the following main components:

3.3.1 Stay Cone

There are one lower and one upper main stay. An inner stay cone and outer stay cone are welded to the main stays. This outer stay cone forms a part of the outer water conduit contour and is embedded in concrete together with outer parts of the main stays. The generator bulb is bolted to the upstream end of the inner stay cone. These parts are located in the centre of the water flow and forms the inner water conduit contour together with the runner hub. Two side stays are located on each side of the bulb upstream of the main stays for stiffening the bulb and avoid resonant vibrations. The total weight and the hydraulic forces are transferred to the surrounding concrete through the stay cone via the two main stay vane structures. The dynamic as well as the static forces from the turbine and the generator are transferred through the structure to the building foundation.

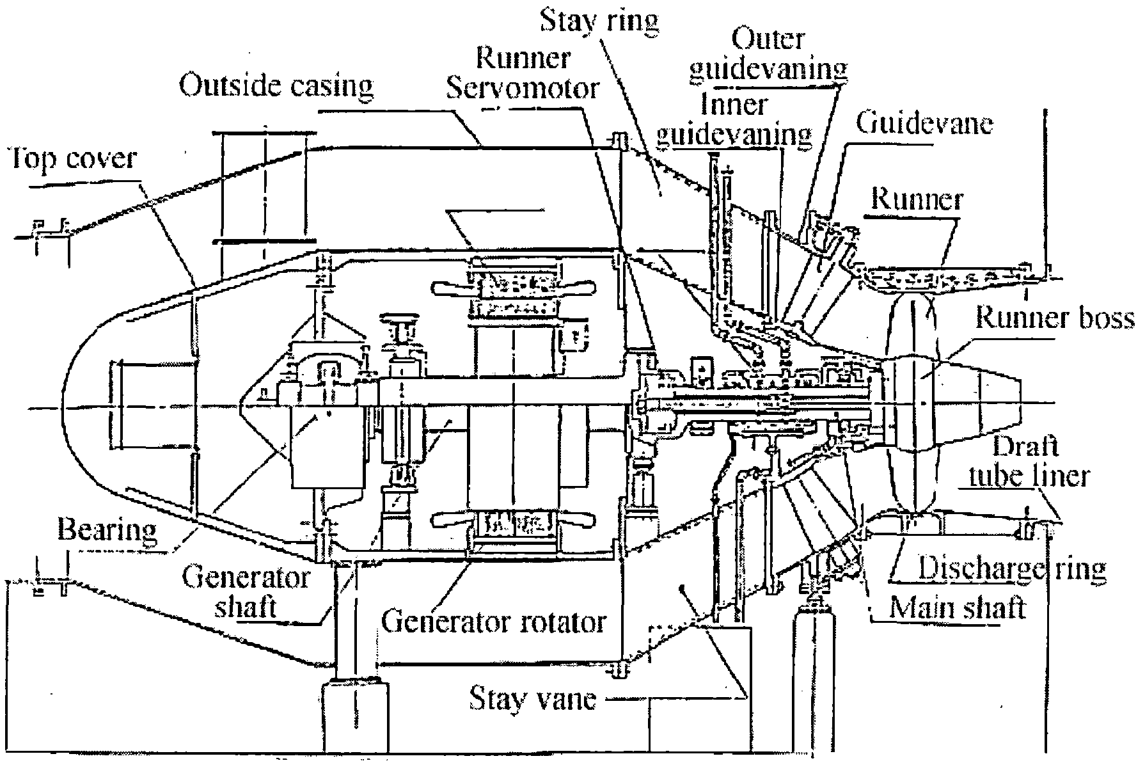


Figure 3.2 Component of bulb turbine

3.3.2 Runner Chamber and Draft Tube Cone

The runner chamber is the connecting part between the outer stay cone and the draft tube cone. The downstream end of the outer cone is provided with a flange to which the runner chamber is bolted. The draft tube cone consists of two or more straight welded steel cones and is embedded in concrete. The upstream end is connected to the runner chamber through a flexible telescope connection. This type of connection is necessary to allow for a certain axial movement of the runner chamber and the outer guide ring because of elongation/contraction due to temperature changes. The length of the steel cone lining is determined by requirements to maximum water velocity at the exit and to avoid damage to the concrete.

3.3.3 Generator Hatch

The generator hatch is normally a part of the turbine delivery. It is located above the generator and provides access to the generator for assembly or dismantling tasks. The hatch consists of a perforated part which forms the outer water conduit contour in the hatch opening. A cylindrical steel mantel with a flange on top is provided for the hatch cover and seal mounting. As the unit's bulb part will rise and lower with filling and draining of the turbine, the seal joint between mantel and hatch cover must allow for a vertical movement of the mantel.

3.3.4 Stay Shield

The stay shields are located between the generator access shaft and the turbine main stay. They form an even wall for the water flow and the stay structure is streamlined at the upstream end to prevent undesired vortex formation. The shields are bolted to the bulb and connected to each other by screw stays for stiffness. They are freely supported against the access way and the main stay to allow for axial movements. The shields are provided with a manhole for inspection and possible maintenance of the space between them.

3.3.5 Rotating Parts

The rotating parts are shown in Fig.3.3 and consists of the following parts

1. Runner
2. Turbine shaft
3. Shaft seal cam,
4. Clamp ring, wear ring and oil thrower ring
5. Turbine bearing oil thrower ring
6. Feedback mechanism and oil piping
7. Oil transfer unit from rotating to stationary parts

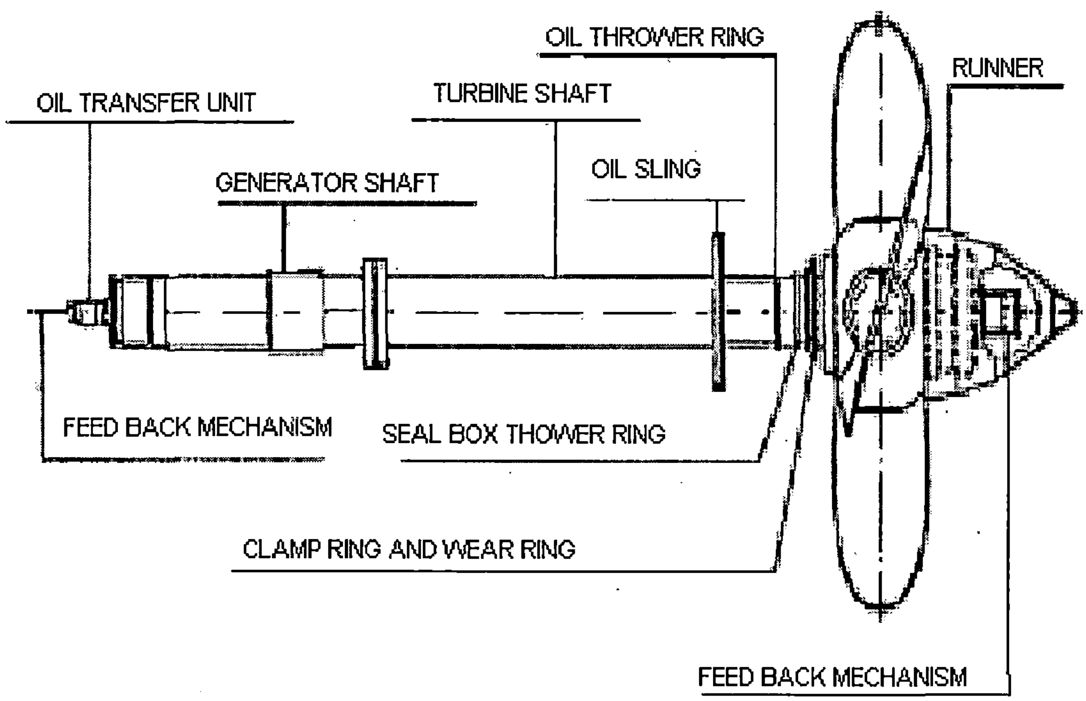


Figure 3.3 : Bulb turbine rotating parts [36]

3.3.5.1 The Runner

The runner is similar to a Kaplan runner and has normally three to five blades made of stainless steel. The blades are designed with flanges and connected to trunnions and levers. The servomotor for moving the blades is normally located inside the hub. Runner and hub is shown in Fig. 3.4 (a) and Fig.3.4 (b). The servomotor consists of a fixed piston and an axially moving cylinder and link supports, links and blade levers are located inside the hub.

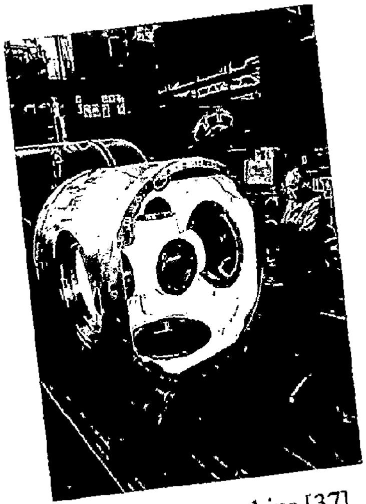


Figure 3.4: (a) Runner [37] (b) Runner hub of bulb turbine [37]

3.3.5.2 The Turbine Shaft

The turbine shaft is made of forged steel and has flanges in both ends. One end is connected to the runner hub and the other to the generator shaft. These joints are pure friction joints.

3.3.5.3 Shaft Seal Box

Several types of shaft seal boxes are in use. This box has radial seal surfaces consisting of a stainless hardened wear disk and two wear rings made of Teflon type fibers.The wear disk is bolted to a cam fixed to the shaft. The wear rings are glued to the seal ring.This is movable and supported in the adjustment ring by means of a membrane. The membrane allows the seal ring to move axially 5 - 6 mm. This is necessary for the shaft movement in the downstream direction when the unit is loaded. In addition allowance must be made for wear of the seal surfaces. The adjustment ring is bolted to the support ring and may be axially adjusted by means of double acting jacking screws. According to the wear range of wear rings the adjustment range of the seal box should be 8 - 10 mm. The auxiliary seal is located in the support ring. This may be pushed against/pulled away from the cam by means of push/pull jacking screws. When this ring is in contact with the cam the wear seal rings may be dismantled without draining the unit. Possible water leakage into the seal box is drained through a pipe to the pump sump. A thrower ring is mounted on the shaft to prevent water leakage along the shaft. A rubber ring is mounted on the upstream end of the shaft and seals against the seal box cover. The seal box is provided with four springs which are pressing the wear seal rings against the seal surface to prevent leakage when the balance system is out of operation, e.g. when filling the turbine.

3.3.5.4 Turbine Bearing

The bearing is sturdy and simple in operation. Maintenance will normally consist of oil change only. The bearing housing is supported in the inner guide ring by means of two yokes and two support stays and rests normally on six wedges. By moving these wedges axially the bearing housing may be vertically adjusted. The bearing housing is split horizontally. The bearing pad shell consists of an upper and a lower part. These are "floating" in the bearing housing where a radial locking pin in the upper part prevents the shells from rotation. The surface of the lower supporting shell and a surface in each end of the upper bearing shell are lined with babbit metal. The oil housing is bolted to the upstream end of the bearing housing. The oil reservoir is fixed to the shaft. The oil scoop and the box are also located inside the oil housing.

3.3.5.5 The Feedback Mechanism and Oil Piping

The feedback mechanism and oil transfer piping are located in the shaft centre. The transfer piping consists of an inner and an outer concentric oil pipe running through the whole shaft length. The inner pipe continues through the oil transfer unit at the upstream end, and it is supported in the outer pipe and connected to the runner servomotor cylinder via yoke. The inner pipe is axially movable and follows the servomotor movement. A pointer mounted to the upstream end is moving along a measurement ruler showing mechanically the servomotor position at any time. .The outer oil pipe is mounted by means of flange connections to the runner hub, turbine shaft and generator shaft respectively.

3.3.5.6 The Oil Transfer Unit

The oil transfer unit is located at the upstream end of the generator shaft and has one fixed and one rotating part consisting of a distribution sleeve and distribution trunnion respectively. The distribution sleeve is fixed to the capsule around the generator shaft end and is provided with pipe connections for oil supply and return as well as leakage oil. The distribution sleeve is provided with a bracket having a measurement scale where the runner servomotor position may be read.

3.3.6 The Guide Vane Mechanism

Two different systems have been used for operating the guide vanes. By means of a link ring a simultaneous movement of the pilot valves for the guide vane servomotors is achieved. The movement is governed by the valves controlling the opening or closing of the guide vanes. The high pressure hoses are connected to the oil pressure system of the unit. The advantages of this system are that even if one guide vane is stuck, the remaining vanes can be moved without any damage. The same will apply if a foreign object is caught and jammed between two vanes during closing, the remaining vanes can be closed without any damage. If required, guide vane system. The disadvantages of this system are the assembly and extensive adjustment work. The total price will approximately be the same as for the regulating ring system which is the other method for moving the guide vanes. The regulating ring system with links and levers is of the same type as for Francis turbines. Due to the conical arrangement of the guide vanes the lever link system must have spherical bearings with large angle movement. The connection between the levers and the vanes is designed as friction joints. This is done to avoid damage to parts if one or several vanes are stuck or if foreign objects are caught between vanes. The friction joint makes it possible for the vane lever to move with the remaining parts of the guide vanes connected to the regulating ring even if the adjacent vane is stuck. A disadvantage may be that large weight of the regulating ring and possible slack in bearings can make the governing inaccurate.

3.4 CONDITION CONTROL

The general principles for condition control are the same as for the Francis turbines. They are:

3.4.1 Runner

The runner should be inspected both from above and below. Particular attention should be given to possible cavitation erosion and scratches on the blades as well as leaks around the blade flange against the hub.

3.4.2 Runner Chamber

The narrow gap between runner and the runner chamber should be checked if foreign objects may have passed the gap and made scratches in the chamber.

3.4.3 Guide Vane Mechanism

For guide vane mechanism with individual vane servomotors on bulb turbines it should be checked that the vanes have an identical movement.

3.4.4 Shaft Seal Box

For Bulb turbines at standstill it should be checked that the water does not flow out of the box along the shaft into the turbine bearing.

3.4.5 Generally for Bulb Turbines

Special attention should be paid to changes in the sound when the unit is in operation.

3.5 MONITORING INSTRUMENTS

The turbine is normally equipped with the following instrumentation:

- A manometer connected to the upstream end of the generator bulb
- A vacuum meter connected to the runner chamber downstream of the runner
- A contact manometer for reading of the pressure in the shaft seal box water pressure pipe, which triggers alarm or alternatively stop signal at too low pressure
- A float switch for alarm for high water level due to too large leak in the shaft seal box.
- A contact thermometer for reading of oil temperature in the bearing housing. The thermometer has two adjustable contacts, one for alarm at high temperature and one for disconnection for the unit on further temperature rise.
- A remote indication thermometer with two temperature sensors for reading of temperature in lower bearing pad.
- A level floats for alarm on low oil level.

3.6 ASSEMBLY AND DISMANTLING

Among the main parts of the bulb turbine it is only the runner which is normally needed to dismantle. The runner chamber is split axially horizontal in two halves. By removing the upper half access to the runner is obtained. The guide vanes cannot be dismantled without extensive work. Repairs of these and the guide surfaces should be performed at the plant.

Bearing and seal box can easily be dismantled. By applying the overhaul seal the seal box may be removed without draining the water canal. Then necessary stairs and floors around the guide vane and the runner chamber may be erected.

The stay shields are adapted against bulb and outer water conduit contour. The shields are mounted as soon as the generator bulb and penstock are completed. Finally the generator hatch dome plate and cover are installed.

3.7 DESIGN OF BULB TURBINE [8]

(1) $Q=\pi DBV_f$

Where V_f =velocity of flow, is determined experimentally (in metre/sec)

D= outer diameter of runner, (in metre)

B= width of runner, (in metre)

Q = the discharge. (in $metre^3/sec$)

(2) K= B/D =0.10 to 0.45

The value k increases as the speed of runner increases

(3) $V_f = \square(2gH)^{0.5}$

Where H is net head available,

Value of □ ranges between 0.15 to 0.30 for slow to fast runner

(4) $kDN/60=k_u(2gH)^{0.5}$

Where N is the rpm of runner

k_u ranges from 0.1 t0 o.9

(5) Inner diameter (D_i) is half of outer diameter (D)

3.8 ADVANTAGES OF BULB TURBINES [11]

The advantages of bulb turbine over other turbines suitable under similar site conditions are as follows

From the hydraulic conditions of bulb turbines considerable advantages result for the civil engineering. Turbine intake and draft tube are easy to form, the spiral can be completely omitted.

The straight-lined design of the turbine inlet permits a reduction of the distance between the units of above 30 percent and thus saving in the width of the power plant.

The height of the power plant can also be reduced.

Besides the essentially reduced volume of civil work also the simple sheeting due to the omission of the elbows reduce the cost of civil works considerably.

The specific speed increase reaches at least 20%. This means that for a given head and output the use of a bulb turbine unit allows a turbine diameter about 10 % smaller than with a conventional turbine. Hydraulic losses due to sharp bends are also reduced. All these together bring to a total gain of about 25 to 35 % of the civil costs.

3.9 DISADVANTAGES OF BULB TURBINE

The disadvantages of bulb turbine over other turbines suitable under similar site conditions are as follows:

- There will be high hydraulic forces transferred to the concrete especially towards the hatch opening for dismantling of the runner chamber and runner. These forces are so high that it has been part of the limitation of maximum head for using Bulb turbines.
- During maximum flood the downstream pressure increases to a level close to upstream level. Then the hydraulic pressure on draft tube and runner chamber will become very high.
- Bulb turbines have a very small inertia GD^2 with a time constant of approximately one second only. This causes a fast increase in speed during load rejection.
- The overhung shafts are exposed to fatigue.
- High humidity is present inside the bulb which can cause the corrosion of several parts.

3.10 EXPERIENCE WITH BULB TURBINES

Technical features of the various power plants in which has Bulb Turbine

3.10.1 Indigenous Experience

A large number of bulb units have been installed in India and have been in satisfactory operations for quite some time. List of some of these installations is given in Table 3.2.

Table 3.2 : Bulb turbine installation in India [38]

S.No	Developer	Location	Turbine size (kW)	Head (m)	Remark
1	Bihar state Hydro-electric power corporation	Eastern Gandak Canal	3x5155	5.3	In operation since 1994
		Western Sone canal	4x1650	3.7	In operation since 1993
		Eastern Sone canal	2x1650	3.7	In operation since 1996
2	Tamilnadu State Electricity Board	Cavery river Mettur dam P.H 1 P.H 2 P.H 3	 17,200 15,600 4,400	 9.0 6.5 3.0	In operation since 1997
3	Haryana State Electricity Board	Western Yamuna P.H 1 P.H 2	9,350 7,310	12.8 11.5	In operation since 1986 - 89

4	Murudeshwar Power Coorperation Ltd, Karnataka	Narayanpur left branch canal	2X5800	7.0	In operation since 1999
5	WBSEB	Teesta canal	4x7850	7.75	In operation since 1996

3.10.2 Worldwide Experience

Table 3.3: The world wide experience of the bulb turbines [39]

S.No	Year of installation	Place	Description about Project
1	1955	Sylvenstein, Germany	First bulb turbine of compact design.
2	1973	Altenwörth, Austria	Most powerful bulb turbines at the time with 44 MW output, runner diameter 6m.
3	1978	St. Mary's, Ontario, Canada	18 MW bulb turbine with 3-blade runners and a diameter of 7.1 m.
4	1978	Ma Ji Tang, Hunan, China	First modern bulb turbine units in China with 18 MW and runner diameter 6.3 m.
5	1979	Sakuma 2, Japan	50 Hz/60 Hz double frequency bulb turbine/generator units with 16.8 MW/ 17 MVA and 4.49 m runner diameter. Pressurized air cooling generators.
6	1982	Shingo 2, Japan	Largest bulb turbine/generator unit in Japan at the time with 40.6 MW/ 40.9 MVA and 5 blades, 5 m runner diameter.
7	1986	New Martinsville, USA	Powerful 3 blades, 7.3 m runner diameter bulb turbine/generator units with 20 MW/ 21.6 MVA.

8	1993	Ybbs-Persenbeug, Austria	Large bulb unit with 48 MW and runner of 7.5 m.
9	1994	Chashma, Pakistan	Eight bulb turbine/generator of 23.7 MW/21.6 MVA ,runner diameter 6.3 m
10	1994	Bailongtan, China	Largest bulb power station in China at the time equipped with six bulb turbine/ generator units each 33 MW/33.7 MVA with runner diameter 6.4m.
11	1995	Karkamis, Turkey	Largest bulb power station in Turkey, with six 35.5 MW bulb turbines with 6.3 m diameter runners.
12	2001	Nina, China	4 units of 40 MW
13	1980	Rock Island,USA	8 units of 53 MW
14	1981	Chautagne,France	2 units of 47 MW
15	1982	Belly ,France	2 units of 47 MW
16	1965	La Rance,France	12 units of 10 MW

3.11 TECHNICAL FEATURE OF SOME TURBINE

3.11.1 World's Bulb Power stations

3.11.1.1 Power Station of Argentina

Main technical parameter of unit for project of extension hydroelectric power station of Argentina (4×7.1MW)

1. Installation capacity	28.4 MW
2. Single set capacity	4×7.1 MW
3. Turbine	
H	16 m
Type	GZ605-WP-275
Diameter of Runner	2.75 m
Rated Output	7.357 MW
Rated Discharge	50.0 m^3/s
Rated Speed	230.8 r/min
Runaway Speed	715 r/min
Rated Efficiency	93.8 %
Max. Efficiency	95.1 %
Suction Head	-3.3 m
4. Generator	
Generator Type	SFWG7100-26/3260
RatedCapacity	7.89/7.1 MVA/MW
Rated Voltage	6.3 kV
Rated Current	417.5 A
Rated Power Factor	0.9 cos□
Rated Speed	230.8 r/min
Insulation Class	F
Flywheel Effect	390 $kN.m^2$
Max. Lift Weight	18 t
5. Governor Type	WST-80-4.0
6. Oil Pressure Unit Type	HYZ-2.5-4.0
7.Excitation Mode	Static Silicon-controlled

3.11.1.2 Manitoba Hydro's Jenpeg Generating Station, Canada

Turbine data	
Rated Net Head	7.3 m
Maximum Net Operating Head	10.7 m

Minimum Net Operating Head	4.9 m
Output at Rated Head	28,900 kW
Output, Maximum	32,000 kW
Rated Discharge	450m^3/sec
Runner Diameter	7.5 m
Specific Speed	1020
No. of Blades	4
No. of Wicket Gates	16

3.11.2 India's Power Stations with bulb turbine

3.11.2.1 Handia (Dimawar) Hydro Electric Project (2x27.5 MW)

Gross Head (One unit running)	14.99 m
Rated Net Head (One unit running)	14.74 m
Installed Capacity	2 X 27.5 MW
Type Of Turbine	Bulb

3.11.2.2 Hoshangabad Hydro Electric Project (2x22.5 MW)

Design Discharge	200 Cumecs (each unit)
Installed Capacity	2 X 22.5 MW
Gross Head	12.87 m
Rated Net Head	12.62 m
Type Of Turbine	Bulb

3.11.2.3 Sone Western Link Canal H.E. Project, Dehri (4x1.65 MW)

Installed capacity	4x1.65 MW
Type of Units	Bulb
Maximum Head	5.5 m.
Minimum Head	3.87 m
Rated Head	3.87 m.
Turbine(s)	1.8 MW, 120 rpm
Bulb Type	Semi Kaplan horizontal shaft

Generator specification	
Type	Synchronous Generator
speed	120 rpm

3.11.2.4 Sone Eastern Link Canal H.E. Project, Barun (2x 1.65 MW)

Installed capacity	2 x1.65 MW
Type of Units	Bulb
Maximum Head	5.21 m.
Minimum Head	3.87 m
Rated Head	3.87 m.
Turbine(s)	1.8 MW, 120 rpm
Bulb Type	Semi Kaplan horizontal shaft
Generator specification	
Type	Synchronous Generator
speed	120 rpm

3.11.2.5 Eastern Gandak Canal H.E. Project, Valmikinagar (3x5 MW)

Installed capacity	3 x 5 MW
Type of Units	Bulb
Maximum Head	7.1 m.
Minimum Head	4.97 m
Rated Head	5.3 m.
Turbine(s)	5.670 MW, 107 rpm
Bulb Type	Kaplan horizontal shaft turbine
Generator specification	
Type	Synchronous Generator
speed	107 rpm

3.11.2.6 Kosi Hydel Power Station (4x4.8 MW)

Generator data	
RPM	62.1

Rated MVA	31.1
Voltage,kV	4.16
Generator Efficiency, % (Including Excitation System Losses)	96.40
Cooling, Stator and Rotor -	Demineralised- Water
Runaway Speed (on cam)	RPM 140
Runaway Speed (off cam)	RPM 192 Dagmara Hydroeletric Project Supaul Bihar on River Kosi
Turbine data	
Type	Kaplan (bulb) with moveable wicket gates & runner blades, horizontal shaft.
Manufacturer	Hitachi Japan
Rated speed	93.8 R.P.M
Rated Head	6.1 m
Rated Output	5600 KW each
Rated Discharge	104.5 cum/sec.
Runner Diameter	4500 mm
Type of governor	Hydro-Mechanical
Generator	
Type	Synchronous, Horizontal Bulb
Rated Capacity	5647 KVA
Generation Voltage	6.6 kV
Rated speed	93.8 R.P.M
No. of Poles	64

3.11.2.7 AGNOOR Small Hydel Project (2x500 KW)

Type of Unit	S-Type Tubular
Maximum Head	4.272 Meters
Minimum Head	3.034 Meters
Design Head	3.37 Meters

Turbine(s): S-Tubular, Horizontal Shaft
Generator: Synchronous generators 2Nos., 500 KW 0.9 p.f., 415 V

3.11.3 Some BHEL supplied projects in India

S. No.	Project	Rating (kW)	Static Head	Speed	Runner Dia.
1.	NIDAMPUR	2 X 500	4.03	136.4	2000
2.	KAKROI	1 X 100	-	125	1500
3.	DAUDHAR	3 X 500	4.89	136.4	2000
4.	KAKATIYA	3 X 230	3.657	166.7	1500
5.	SURATGARH	2 X 2000	-	187.5	2200
6.	MANGROL	3 X 2000	9.601	166.7	2500
7.	TARWANMEDHE	1 X 200	-	166.7	1500
8.	BARNA	2 X 750	12.85	273	1500
9.	EASTERN SONE	2 X 1692	5.21	120	3220
10.	WESTERN SONE	4 X 1692	6.34	120	3220
11.	GANEKAL	1 X 350	3.81	136.4	2000
12.	TEESTA PH-I	3 X 7500	10.224	143	4000
13.	TEESTA PH-II	3 X 7500	12.562	143	4000
14.	TEESTA PH-III	3 X 7500	10.429	143	4000
15.	DHUPDAL	2 X 1400	8.17	158	2500
16.	NARAYANPUR	2 X 4500	-	111.11	4100
17.	MUKERIAN	2 X 9000	8.23	125	4500

CHAPTER 4
ANALYSIS OF BULB TURBINE

4.1 INTRODUCTION

"Bulb type" and "Pit type" turbines which are used in low head turbine units of major size.The pit may be designed with circular or elliptical cross section, depending on the turbine size. For smaller split turbines with cylindrical housing of steel plate, the pit may be arranged also horizontally, perpendicular to the turbine axis. In this case the generator with belt tensioning device is arranged sidewards and may be fixed on the turbine housing. Depending on hydraulic and/or structural reasons, sometimes the arrangement of a streamlined concrete column in front of the pit is necessary. Furthermore, depending on the turbine net head and the type of turbine runner, it is possible to arrange the turbine housing partially or fully above headwater level (partial or total syphon arrangement). Compared with other turbine types, the main advantages of the split turbine under aspects of mechanical engineering are the very compact and simple design, the short turbine shaft with free end for easy arrangement of and good access to the runner control elements. Assembly and disassembly of shaft and bearings are easier as in case of the S-turbine or in case of gearbox drive. Only one shaft seal is required (instead of two for the S-turbine). Manufacturing cost for the draft tube steel liner is less than for the elbow type draft tube of vertical units or for the double bended draft tubes of S-type turbines. Installation time for the split type turbine is shorter than far all other types of low head turbines; in most cases one week is sufficient for placement and concreting if the unit (without generator) is pre-erected in the shop and transported as one piece to site. Under hydraulic aspects the main advantage of the split type, especially against the S-type turbine, is the straight or only slightly bended draft tube. Thus, the losses of efficiency are lower, especially in the range of high turbine discharge. Another advantage of the split type against the S-type is the shorter length of transition of the water duct between trashrack and turbine. The hydraulic geometry of the turbine between distributor (wicket gates) and end of draft tube is identical with that of big units of the "bulb type" or "pit type", resulting in similar high efficiencies and specific outputs.

4.2 RESUMING THE ADVANTAGES OF BELT DRIVES AGAINST GEARBOX DRIVES THE FOLLOWING MAY BE STATED:

- Efficiency of the single-stage belt drive at least of same level as for high quality
- Single-stage gearbox and in each case higher than for multi-stage gearbox

- Ratio of speed increase per stage for belt drive higher than for gearbox
- Shock-torques caused by turbine or generator (grid) are damped and absorbed by the elasticity of the belt
- Belt drive with grease lubricated roller bearings require less maintenance and by far and less automatic monitoring than gearbox drives with oil filling (no oil level, -pressure, circulation, -filtering, no oil coolers, no oil replacement) in total less components of the generating unit (less bearings, no coupling)
- Less cost and time required for spare parts (belt less expensive, quicker and easier replaceable than toothed wheels)
- Less danger in case or runaway speed or the generating unit
- Low cost, short time and easy work required for adaptation to other turbine speed or speed ratio in case of changing hydraulic conditions (e.g. changing head due to changed headwater or tailwater levels)

It has to be emphasized that the belt drive including the bearings and shafts of turbine and generator has to be properly designed and dimensioned. The belt itself requires minimum attention and maintenance but has to be kept clean. The belt drive will show the expected performance only in case that these requirements are fulfilled.

4.3 BULB EFFICIENCY AND RANGE OF APPLICATION

Bulb turbines have a high hydraulic efficiency which bears comparison with the peak figure of 92.5 per cent for the best Kaplans, despite the fact that they operate at a much higher specific discharge. They are also very stable in operation, in spite of their low inertia. Bulb generator losses are a little higher than for conventional design. Efficiency for a power factor of 0.98 is around 97 per cent Average temperature rise is generally low and well within acceptance limit.

The maximum economic size is generally in the range of 7 to 7.5 meter runner diameter and with a maximum head up to 30 meter. The minimum size is determined by the requirements for access into the Bulb; the practical limit being reached when the runner approaches a diameter of 3 to 4 meter.

4.4 DISTINGUISH FEATURES FROM OTHER TURBINE

The higher full-load efficiency and higher flow capacities of bulb turbines can offer many advantages over vertical Kaplan turbines. In the overall assessment of the project, the application of bulb turbines results in higher annual energy and lower relative construction costs.

A main difference from the Kaplan turbine is moreover that the water flows with a mixed axial-radial direction into the guide vane cascade and not through a scroll casing. The guide vane spindles are inclined (normally 60^0) in relation to the turbine shaft. Compared to other turbines, this result in a conical guide vane cascade.

4.4.1 Advantages

i. Bulb turbines have the advantage of a straight inflow to the distributor and runner and also the outflow through the conical draft tube has no bend.This results in a minimum of losses in the entire application range and therefore higher efficiencies

ii. Bulb turbine has many advantages compared to the other turbines used for similar site conditions. The straight-lined design of the turbine inlet permits a reduction of the distance between the units of above 30 percent and thus saving in the width of the power plant. Also the specific speed increase reaches at least 20%. This means that for a given head and output the use of a bulb turbine unit allows a turbine diameter about 10 % smaller than with a conventional turbine

4.4.2 Disadvantages

i. There will be high hydraulic forces transferred to the concrete especially towards the hatch opening for dismantling of the runner chamber and runner. These forces are so high that it has been part of the limitation of maximum head for using Bulb turbines.

ii. During maximum flood the downstream pressure increases to a level close to upstream level. Then the hydraulic pressure on draft tube and runner chamber will become very high.

iii. Bulb turbines have a very small inertia GD^2 with a time constant of approximately one second only. This causes a fast increase in speed during load rejection.

iv. The overhung shafts are exposed to fatigue.

v. High humidity is present inside the bulb which can cause the corrosion of several parts.

vi. Bulb units are only available in the larger diameters, and should be avoided due to the "confined access" problems associated with the bulb

4.5 DIFFICULTIES EXPERIENCED WITH MOST LARGE BULB UNITS

These are mainly associated with the following factors:

- Compact generator design, high heat flux and sharp temperature gradients.
- Heavy, horizontally-overhung turbine components.

4.6 PROBLEM ASSOCIATED

Problems Associated With Bulb Turbine

1. Specific problem in bulb turbine
   - The deformation of outer distributor cone in bulb turbine due to cavitation.The top and bottom sides of the outer distributor cone enlarged in size from the initial dimensions at the time of erection of turbine.
   - Distortion of guide vane assembly in a bulb turbine due to cavitation and reverse water flow.
   - Problems involved in a deep setting bulb turbine in a hydro- electric power plant due to cavitation.
   - The efficiency deterioration due to mounting of draft tube .
   - High vibration and vertical cracks in the concrete structure of the powerhouse.
2. Common problem in bulb turbine
   - Problems related to the cooling, bearings.
   - Erosion of bulb turbine blades.

4.6.1 Major Drawback

4.6.1.1 Cavitation:

Runners Of bulb turbines are axial with adjustable blade pitch angle and the control of both the guide vane opening and the blade pitch angle allows optimized operation of the machine, so called "on cam" operation. For the design operating range a cavity development takes place at the hub of the runner. This type of cavitation is very sensitive to the Thoma number. Any effect of the water cavitation nuclei content is observed for this type of cavitation. However, the air entertainment can have a great influence on the extent of this cavity.Since the blades are adjustable, the runner is not shrouded the tip clearance cavitation takes place in the gap between the blades and the machine casing, leading to an erosion risk even though the head could be low. This type of cavitation is driven by the flow shear layer in this gap and it is not very dependent of the Thoma cavitation number. Even for the case of on cam operation, leading edge cavities can be observed at the inlet of the runner but can be avoided by improving the shape of the blade leading edge.

The relative importance of the many cavitation parameters affecting water turbines in general is very difficult to establish, and bulb turbines are no exception. Experience to date can be summed-up as follows:

1. After more than 20,000 hours in operation, no appreciable cavitation wear has been observed on runner blade and discharge ring of 17% Cr 4% Ni, 13% Cr or 18% Cr 8% Ni steel.
2. Bulb turbine cavitation seems to depend more critically on machine elevation than with Kaplans. This seems reasonable in the light of experience with larger bulb units with runner diameters roughly equivalent to half the head (6 m and 12 m respectively for example) whose discharge rings showed signs of exceptionally severe cavitation wear on their top halves. Bulb turbine elevation, therefore, which is set with reference to the highest point of the machine, is much more critical than for Kaplans, which are set with the midpoint as the datum. There is only one *case* on record of cavitation having affected the bottom half of a discharge ring.
3. Cavitation wear observed on mild or low-alloy steel runner blade has generally been found to originate near the leading edges on the outside of the blades, and to a lesser degree also on the inside. There have also been some cases where cavitation has developed along the rim on the outside of the blade and at the blade edge.

The worst cases of cavitation after 40,000 hours of operation affected areas extending 500 mm inwards from and along a considerable length of the leading edge. Basic counter-measures in the absence of stainless materials and for given operating conditions include the improvement of combined runner blade and wicket gate control efficiency, operating limitation and hydraulic profile geometry.

4.6.1.2 Joints and seals

Joint and seal problems are the same as for conventional turbines but require closer attention as they tend to have more serious consequences.

4.6.1.3 Shaft seals

Like cavitation, shaft seals are a problem for which it seems impossible to establish any clear-cut data, in spite of the wealth of information which tests on a wide variety of turbine designs have provided. Though flat seals or gaskets do not stand up particularly well to varying pressure at transient operating conditions, an increasing number of operators seem to prefer them to radial designs. For cylindrical seals, carbon seals have been found satisfactory in some cases, while polyamide, polyethylene or nylon seals seem preferable in many others. Better results have sometimes been achieved by providing chambers concentrating the pressure on the seal bearing surface and thus reducing the specific pressure, especially for the fourth row of flat seals, which is the one exposed to the highest pressure and at which the water film starts to form.

4.6.1.4 Bolted assemblies

Horizontally-overhung heavy rotating components are assembled to the shafts with short studs and large nuts between 70 and 100 mm in diameter. As the studs undergo cyclic stresses of several hundred bars during runner rotation, the nuts must be tight to a degree that is difficult to determine by conventional methods. For final tightening, the study is expanded by insertion of a heater element or pre-stretched with the aid of screw-jacks; whereupon the nut is run on by hand until it comes up against its bearing surface. The study is then left to contract back to its original length. The difficulties experienced in determining the correct tightness are twofold: (a) determination of the correct stress, (b) how to obtain sufficient elongation for stresses of up 1,500 to 2,500 bars.

4.6.2 Minor drawbacks

In addition the main problem areas described, the following minor drawbacks are associated with bulb units:

4.6.2.1 Noise

They are a little noisier than conventional designs.

4.6.2.2 Wicket gates

Shear link or stud failures when the guide vanes are dosing seem to occur a little more frequently on bulb units than on Kaplans. This is probably because bulb units are set at a shallower depth and generally have more open trashracks (150 to 200 mm), so that more foreign bodies pass through the wicket gates with the flow. More-over, since bulb power plants are equipped with a larger number of units than conventional plants, their operators probably more aware of this problem.

4.6.2.3 Runaway condition/Bypass operation

No major problems seem to have arisen with bulb units operating under the operation under downstream gate control conditions.

4.6.3 Effect of Pier

A bulb turbine is a kind of axial hydraulic turbine and it has a large bulb containing the generating at the centre of the water path. Its runner consists of several adjustable blades. The number of blades normally varies from three to seven and the blade angle is , at maximum, adjustable from minus a few degrees measured from the circumference. The flexibility of the runner under partial load or overload conditions. The draft tube of a bulb turbine is usually designed as a straight diffuser. The energy losses in the draft tube are lower than those in the hydraulic turbine due to the adjustable blades and simple shape. However, only slight energy losses cause a remarkable decrease in turbine efficiency. This is because, bulb turbines are used for low-head hydroelectric power plants. The heads typically vary from 5 to 30 m. Therefore, detailed investigations into the fluid phenomena in a draft tube. In designing the draft tubes with piers, particular attention must be paid to avoid decreases in turbine efficiency while maintaining sufficient structural strength.

Generally, it is known that unsteady boundary layer separations often occur in a draft tube. In addition, the wakes from the runner blades depend on the rotational speed and number of blades. According to a past study on the draft tube a pier , however, the steady computations were confirmed to agree with the experiments qualitatively. What effect the pier had was first investigated. Therefore, two draft tubes with the same shape, one without a pier (case ep1) and the other with a pier (ep2), were simulated. A larger mass flux condition than the best efficiency point was selected among the on-cam conditions of ep1 to examine the differences between flow fields. Second, the optimal shape of the draft tube that would enable the turbine to achieve great efficiency was investigated. Computations were made for different shapes of draft tubes and five cases were selected from these (from sd1 to sd2). The best efficiency point of the original model without a pier was selected. This is because the best efficiency value is one of the important characteristics of hydraulic power plant, and it is often required to be higher than in other existing plants.

There are three causes of total pressure losses in a draft tube, i.e., friction, shock, reverse flow. The pier should be designed to reduced the hydraulic losses due to these

4.6.4 Efficiency Alteration

The efficiency alteration for bulb turbines is mainly due to the development of hub cavitation. As this hub cavity reaches the blade trailing edge, we can notice an efficiency drop. This type of cavitation has already mentioned is very sensitive to the Thoma number and determines the plant NPSE of the machine. Depending on the head of the machine limited development of tip clearance cavitation can be admissible for plant NPSE. Especially for the case of Kaplan or bulb turbines, it Especially for the case of Kaplan or bulb turbines, it can be noticed a strong influence of the Thoma cavitation number on the runaway speed can be noticed a strong influence of the Thoma cavitation number on the runaway speed.

CHAPTER 5

ANALYSIS FOR INTAKE STRUCTURE

5.1 GENERAL

The main obstacle in the development of small hydroelectric power plants has been economics. In most cases, the cost per installed kilowatt for small hydropower is still higher than for fossil fuel plants. For low head hydroelectric installations with head less than 20 m, the major costs are the initial investment in the civil works structure and the electromechanical equipment. If the cost of the structure can be reduced, more small hydroelectric installations would be feasible [40]. In low head water power plant the water velocity is low in the intake section so the losses are small. Saving in capital cost can be achieved by simplifying and shortening the length of the intake.

5.2 METHODOLOGY FOR ANALYSIS OF INTAKE STRUCTURE

A typical layout of small hydropower station as shown in Fig.5.1 is selected for study. The dimensions of various components are worked out in terms of runner diameter which is important parameter for deciding the sizing of components. Indian standard [IS 12800] has been followed for preparing the layout of power house. Tailrace level is assumed at the elevation of 100 m and other elevations are decided as per guidelines of Indian Standard [IS 12800] [41]. All the dimensions are given in terms of runner diameter. For calculating capacity Eq. 1.1(Chapter 1) is used.

For calculation of actual specific speed, initially trial specific speed is assumed 1000 and turbine speed is calculated as per Eq. (5.1).

$$N_S = \frac{N\sqrt{p}}{8.53H^{1.25}} \qquad (5.1)$$

Where p is the power, N is the turbine speed (r.p.m), N_S is specific speed.

Generator speed is chosen nearby this turbine speed so that turbine and generator is directly coupled. Actual specific speed is calculated with respect to selected speed. Now actual specific speed is used in Eq. (5.2) to calculate velocity ratio (ϕ), further velocity ratio (ϕ) will used to calculate runner diameter of turbine as given by Eq. (5.3)

$$\phi = 0.023\ (N_S)^{2/3} \qquad (5.2)$$

$$\text{Diameter of runner (D)} = \frac{84.47\phi\sqrt{H}}{N} \qquad (5.3)$$

The diameter worked out as above is used for preparing the layout of the scheme. Plan and Longitudinal section of power house with bulb turbine layout is shown in Fig. 5.1.

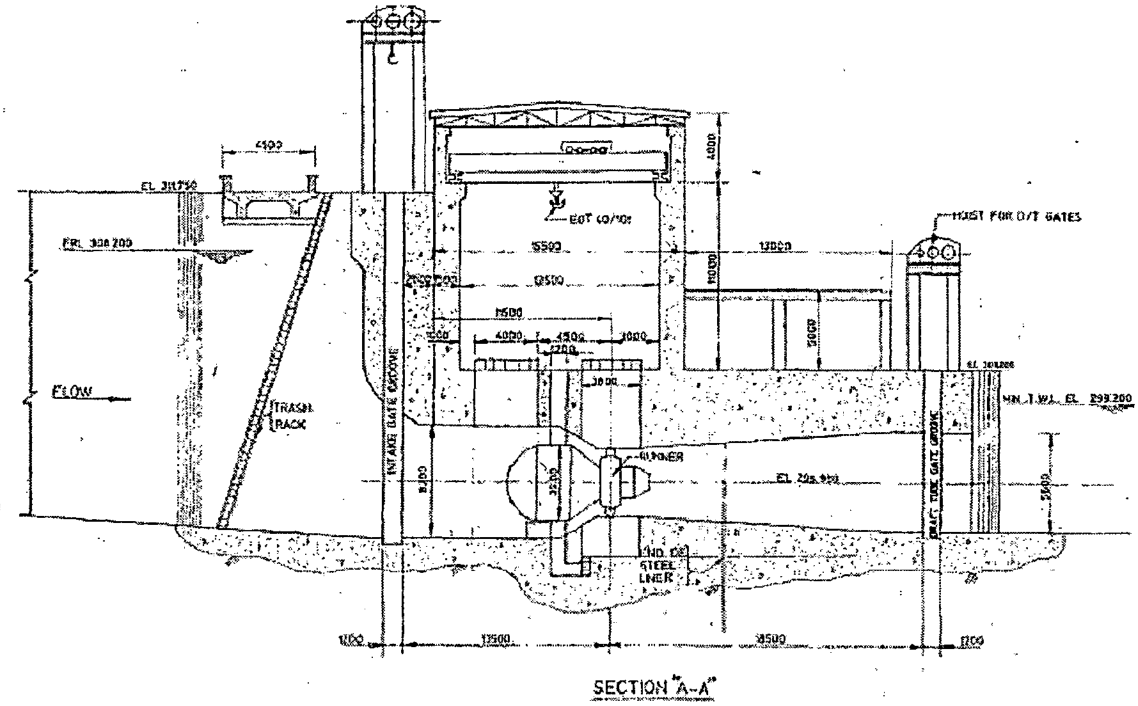


Fig. 5.1 Bulb turbine layout

5.2.1 Head Losses

While calculating head losses in intakes, following losses are considered

i. Bend losses
ii. Friction losses
iii. Contraction losses
iv. Trash rack losses
v. Stoplog and gate slots losses

5.2.1.1 Bend Loss

At the intake water velocity is diverted at an angle due to which bend losses occ Bend losses are calculated by using following equations

$$H_B = K\frac{V^2}{2g} \quad [43] \qquad (5.4)$$

Where H_B is the bend loss in m, V is the mean velocity m/s in the intake.

$V=V_1+V_2/2$, V_1 and V_2 is the velocities m/s at the inlet and outlet of the intake respectivel

V_1 = Discharge (m^3/s) through intake / A_1, A_1 is the area normal to flow direction at inlet of the intake (m^2), V_2 = Discharge (m^3/s) through intake / A_2, A_2 is the area at outlet of the intake (m^2), g is the acceleration due to gravity. In calculations value of g is 9.81 m/s^2. K is the bend loss coefficient and is a function of the ratio of bend radius, R to the intake diameter D and the bend angle.

$$K = k_1 \times k_2 \qquad (5.5)$$

k_1 is the bend loss coefficient as a function of the ratio of bend radius, R to the intake diameter D. Value of k_1 and k_2 are taken from the given Table. 5.1 and 5.2 respectively.

Table-5.1: Bend loss coefficient, k_1 as a function of the ratio of bend radius, R to the intake diameter D [43]

R/D	0.5	1	2	3	4	5	10
k_1	.35	.22	.14	.10	.09	.08	.07

Here for calculation k_1, ratio of mean radius (R_m) in meter and mean intake height (D_m) in meter is taken. k_2 is the bend loss coefficient as a function of bend angle

Table-5.2: Bend loss coefficient, k_2 as a function of bend angle [43]

angle in degree	10	20	30	40	50	60	70	80	90	100	110	120
k_2	.2	.36	.50	.63	.74	.82	.89	.95	1	1.05	1.09	1.12

5.2.1.2 Friction Loss

Friction losses occur due to roughness of the surfaces. It is calculated by using the following equation

$$H_F = \frac{V^2 n^2 L}{\left(D_m/4\right)^{4/3}} \qquad (5.6)$$

Where H_F is the friction losses in m, V is the mean velocity m/s in the intake, L is the length at the center of the intake in meter, D_m is the mean intake height of the intake in meter, n is the rugocity coefficient. Value of n is taken as 0.014.

5.2.1.3 Contraction Loss

Contraction losses occur due to reduction in the opening area at inlet to outlet of intake. These are given by following equation

$$H_C = K_c \frac{V_1^2 - V_2^2}{2g} \text{ [44]} \quad (5.7)$$

Where H_C is the contraction losses in m, V_1 and V_2 are the velocities in m/s at inlet and outlet of the intake, K_c is the contraction coefficient which is taken according to the following table

Table-5.3: Contraction coefficient [44]

Contractions coefficient K_C	
D2/D1	K_C
0	0.5
0.4	0.4
0.6	0.3
0.8	0.1
1.0	0

In the above Table 5.3 D_1 and D_2 is the intake height at the inlet and outlet in meter.

5.2.1.4 Stoplog and Gate Slots Loss

When the gates are fully open and they do not encroach the flow streamlines then losses occurs due to the gate slots only. Similarly losses occur due to stoplog slots. For both stoplog and gate, Slot loss coefficient 0.01 is recommended for each pairs of slots [43]. In full open position loss coefficient is taken for left and right slot that is 0.01.In this way total slot loss coefficient for stoplog and gate = 0.01+0.01= 0.02

$$H_S = 0.02\ V^2/2g \quad (5.8)$$

Where H_S is the stoplog and gate slot losses in m, V is the mean velocity in m/s.

5.2.1.5 Trash Rack Loss

Trash racks are provided at the entrance of intakes to protect turbines from floating debris. Generally the trash rack velocities are on the order of 0.6 to 1.5 m/s in the gross area [43]. Velocities in the net area are somewhat greater, but do not normally result in significant head losses unless the trash rack is blocked. In low head plants the trash rack may provide a significant portion of the total loss in the intake. It is important to consider trash rack losses. Where rack blockage may be frequent, the value of the energy lost may justify larger trash racks, greater spacing between trash rack bars, trash rack cleaning equipment, or racks which can be more readily maintained.

The loss of head due to the trash rack in front of the intake depends on a number of parameters including, but not limited, to the following:

i. Thickness, t, of the bar
ii. Clear space, b, between the bars
iii. Depth, d, of the bar
iv. Shape of the bar cross section
v. Ratio of the area of the trash rack members to the total area
vi. Angle of the rack to the horizontal
vii. Horizontal angle between the approach velocity and the exit velocity
viii. Function and type of rack cleaning

Following formula is used for calculating the trash rack losses

$$H_T = S\left(\frac{t}{b}\right)^{4/3} Sin\beta \frac{V_1^2}{2g} [43] \qquad (5.9)$$

Where H_T is the loss of head through racks in ft, S is the factor depending on bar shape in accordance with Figure 5.1, V_1 is the approach velocity in metre/s, t is the thickness of the bar in metre, b is the clear spacing between bars in metre and β is the angle of bar inclination to horizontal in degrees.

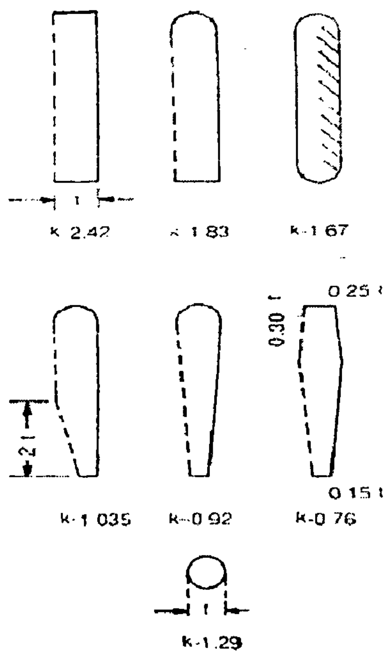


Fig. 5.2 Values of K for different shapes of bars [43]

For intakes, bars rounded from both sides have been selected. Value of K is 1.67for this shape of bar. Different values of according to the shape are given in Fig. 5.2.

5.2.1.6 Total loss

Total losses in the intake is the summation of bend losses, friction losses, contraction losses, trash rack losses, stop log and gate slots losses

Total losses in intake (H_L) $=H_B+H_F+H_C+H_S+H_T$ m

5.2.2 Vortex Prevention

In the typical hydropower intake, water at a low velocity accelerates over a relatively short distance to a velocity generally greater than 2.44 m/s and potentially as high as 6m/s[40]. In order to minimize head losses and provide for the best efficiency of the hydraulic machinery, it is generally desirable to maintain as uniform a flow distribution as possible throughout this process. Low head units such as bulb units which operate with minimal intake submergence are particularly sensitive to upstream flow distributions which may result in vorticity, nonuniform flow in the unit, and less then optimal performance. Vorticity is defined as the flow circulation per unit area and is exhibited as swirling flow patterns. The swirling flow patterns may be either stable or unstable, may occur at the surface or be submerged, and it at the surface, may entrain air, or, if submerged may release air or gas in solution. The formation of vortices is commonly associated with the submergence and orientation of hydroelectric intakes. Idealized geometries are provided that will minimize flow separation and vorticity in both the intake approaches and the power intake.

The prevention of vortices at hydropower intakes is an essential part of the hydraulic design process. The problem most frequently ascribed to vortex formation at an intake is the loss of hydraulic efficiency related to hindrance of flow into the intake. Vortices may also:

i. Produce nonuniform flow conditions
ii. Introduce air into flow, potentially creating rough operating conditions for hydraulic machinery including vibration, cavitation, and unbalanced loadings.
iii. Increase head losses and decrease efficiency
iv. Draw debris into the intake

There are also potential effects on the safety of persons traveling on the surface of the water in the vicinity of a vortex. Vortex formation may also result in unusual loadings on structures in the approach, such as retaining walls, by causing depressed water surfaces and pressures in the vicinity of the vortex. Vortices may result in erosion of bed and embankment material.

The general means for preventing vorticity is to ensure adequate submergence on the intake. Providing a deep enough approach flow minimizes the surface velocity and the potential for development of swirl. The submergence required is dependent on approach conditions and the orientation of the intake as well as the final velocity and the characteristic dimension of the intake.

According to Indian Standard (IS11388) for intakes to prevent vortices, the centre line of intake should be so located as to ensure submergence requirements given in Fig. 4.3. For large size intakes at power plants for which Froude no. (F_r) is less than or equal to 1/3, ratio of submergence depth (h) and intake height varies from 1 to 1.5. For the installation in which Froude no. (F_r) is greater than or equal to the 1/3 submergence to intake height ratio is calculated by the following formula [43]

$$h/d = 0.5 + 2\,F_r \qquad (5.10)$$

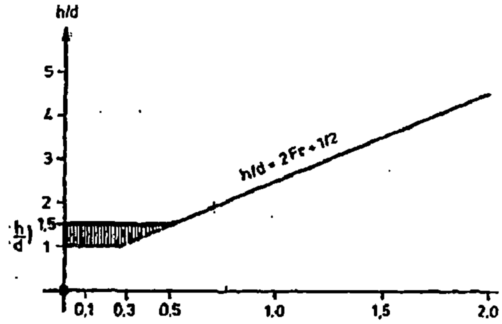


Fig. 5.3 Recommended submergence for intakes [43]

Froude no. is calculated by using the following equation

$$\text{Froude no. (Fr)} = \frac{V_2}{\sqrt{gd}} \qquad (5.11)$$

Where V_2 is the velocity at outlet of intake in m/s, d is the intake height, g = 9.81 m/s^2.

5.2.3 Cost Estimation

In comparison of cost, the most contributing item is the reinforced cement concrete (RCC) including reinforced steel, which has been used for cost comparison. For estimation of cost, volume of sub structure and super structure is separately determined which in turn will give the volume of material used. Following rates are taken of RCC for substructure and super structure.

RCC M-20 in substructure including steel = Rs. 4797 per m^3

RCC M-20 in super structure including steel = Rs.5771per m^3

5.3 MODIFIED DESIGNS OF INTAKE STRUCTURE

For the bulb turbine power plant as shown in Fig. 5.1, four different intakes have been designed as shown in Fig. 5.4. Intake-1 has a geometry which is generally followed in present days. In Intake-2 curved surfaces of roof top is converted in straight edges. As given in the figure 5.4

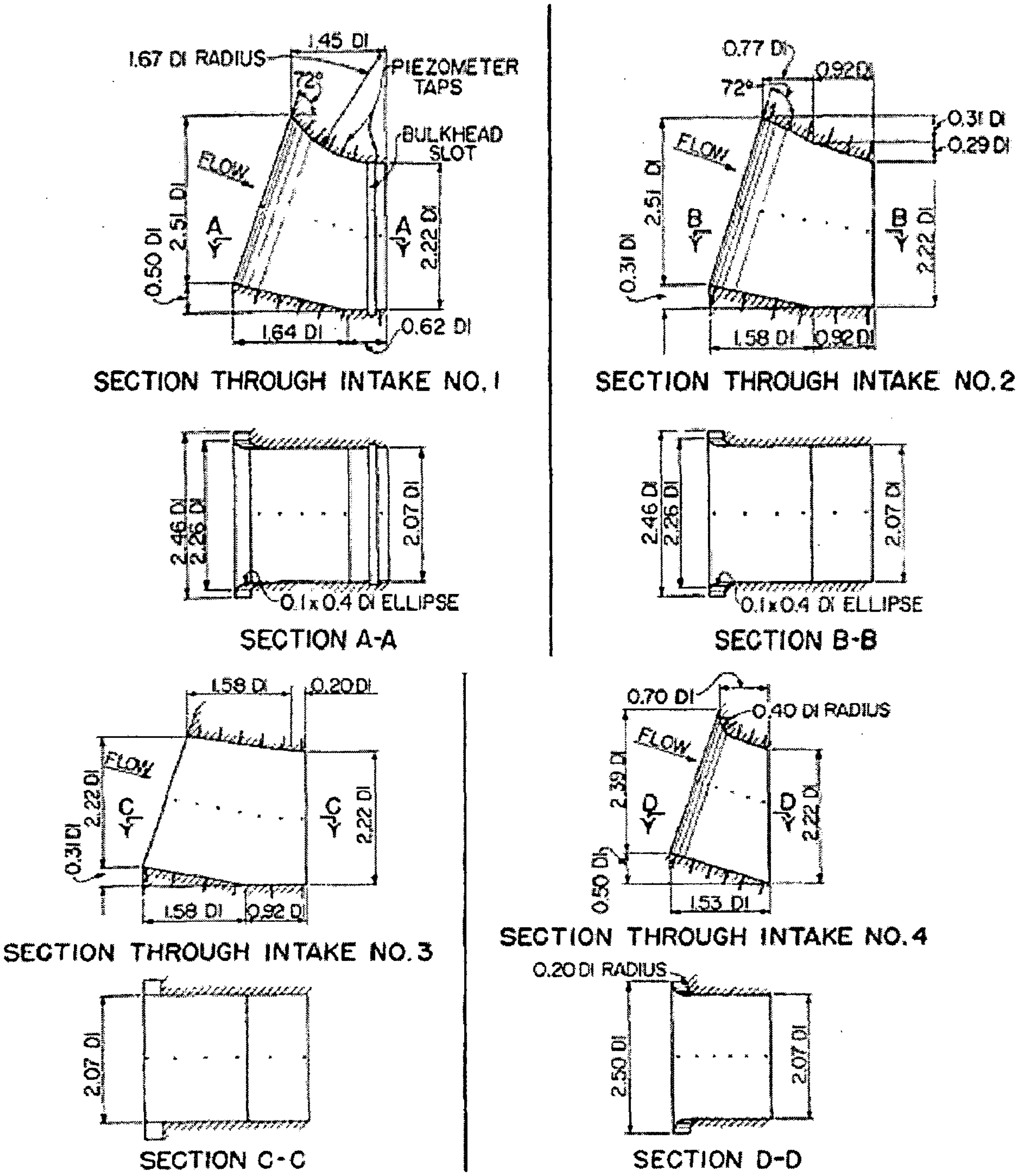


Figure 5.4: Various intakes design

5.4 CALCULATION FOR RUNNER DIAMETER

Installed capacity is 5 MW. Discharge has been taken 90 m^3/s. design head is 7 m. As. Overall efficiency of plant is taken 87 % by taking turbine efficiency 0.91 % and generator efficiency 97 %. With the help of Eq. (5.1) turbine speed is calculated. Trial specific speed is taken1000. At 161 rpm turbine can be directly connected to the generator having speed of 165 rpm. Now corresponding to this speed, actual specific speed is calculated which is around 1050.Using this actual specific speed in Eq. (5.2) velocity ratio (ϕ) is calculated. Further with the help of Eq. (5.3), runner diameter is calculated which is given in Table 5.4.

Table-5.4: Calculation of runner diameter

Capacity in MW	Trial specific speed (N_S)	N in rpm	Actual specific speed (N_S)	Velocity ratio(ϕ)	Runner diameter (D) in m
5	1000	161	1050	2.365	3.900

5.5 CALCULATION FOR HEAD LOSSES

Head losses are calculated for different type of intakes as per methodology described in para 5.2.1.

5.5.1 Calculation for Bend Losses

Bend losses for different intakes are calculated by using Eq. (5.4) and Eq. (5.5) which is given in article 5.4. Bend loss coefficient k_1 and k_2 is taken from table 5.2 and K is calculated by using Eq. (5.5).

Table-5.5: Bend losses

Types of Intake	Mean radius (R_m) in M	Intake height (H_m) in m	Ratio R/ H_m	θ in deg.	Bend loss coefficient k_1	Bend loss coefficient k_2	Bend loss coefficiet (K)	Mean velocity (V) in m/s	bend loss (H_B) in m
Intake-1	22.035	8.814	2.5	18	0.12	0.328	0.03936	1.3575	0.00369
Intake-2	22.035	8.814	2.47	15	0.1212	0.280	0.03396	1.3575	0.00318
Intake-3	20.424	8.073	2.53	10	0.119	0.200	0.02380	1.52	0.00282
Intake-4	17.34	8.814	1.96	5	0.14	0.110	0.06020	1.419	0.00158

5.5.2 Calculation for Friction Losses

Friction losses are worked out by using Eq. (5.6).Values of parameters used in Eq. (5.6) for different intakes are given in Table 5.6 along with the friction losses.

Table-5.6: Friction losses

Types of Intake	Mean velocity (V) in m/s	Rugocity coefficient (n)	Length(L) in m	mean intake height (D_m) in m	Friction loss(H_F) in m
Intake-1	1.3575	0.014	8.814	9.2235	0.001050
Intake-2	1.3575	0.014	9.750	9.2235	0.001159
Intake-3	1.52	0.014	9.750	8.6580	0.001581
Intake-4	1.419	0.014	5.967	8.9895	0.000802

5.5.3 Calculation for Contraction Losses

Contraction losses are calculated according to the Eq. (5.7). Ratio D_2/D_1 is calculated. From the table 5.3 contraction coefficient is selected which is matched with the value of ratio D_2/D_1.Values of contraction loses for different intakes are given in Table 5.7.

Table-5.7: Contraction losses

Types of Intake	Intake height at the inlet (D_1) in m	Intake height at the outlet (D_2) in m	$\frac{D_2}{D_1}$	Contraction coefficient K_C	Inlet velocity (V_i) in m/s	Outlet velocity(V_2) in m/s	contraction loss (H_C) in m
Intake-1	9.789	8.658	0.88	0.06	1.195	1.52	0.0053
Intake-2	9.789	8.658	0.88	0.06	1.195	1.52	0.0100
Intake-3	8.658	8.658	1	0	1.528	1.52	0.00
Intake-4	9.321	8.658	0.93	0.04	1.318	1.52	0.0053

5.5.4 Calculation for Stop log and Gate Slot Losses

Eq. (5.8) is used for calculating the stop log and gate slot losses. Stop log and Gate slot Losses for different intakes are given in Table 5.8.

Table- 5.8: Stop log and gate slot losses

Types of Intake	slot loss coefficient	Velocity at the gate slot (V) in m/s	Stop log and Gate slots losses (H_G) in m
Intake-1	0.02	1.3575	0.001878
Intake-2	0.02	1.3575	0.001878
Intake-3	0.02	1.52	0.002355
Intake-4	0.02	1.419	0.002052

5.5.5 Calculation for Trash Rack Losses

Eq. (5.9) is used for calculation of trash rack losses. Factor depending upon bar shape, thickness and spacing between bars are same for all intakes. Trash rack losses for different intakes are given in Table 5.9.

Table-5.9: Trash rack losses

Types of Intake	Factor depending upon shape of bar (K)	Thickness of bars (t) in inches	clear spacing between bars (b) in inches	angle of bar inclination to horizontal β in degrees	velocity (V_l) feet/s	Trash rack Loss (H_T) in Feet	Trash rack Loss(H_T) In m
Intake-1	1.67	0.4	5.2	72^0	5.23	0.072438	0.022079
Intake-2	1.67	0.4	5.2	75^0	5.43	0.07807	0.024168
Intake-3	1.67	0.4	5.2	80^0	7.35	0.14304	0.032590
Intake-4	1.67	0.4	5.2	85^0	5.23	0.072438	0.022104

5.5.6 Calculation for Total Losses

Total loss is calculated by adding all losses. Total losses for different intakes are given in Table 5.10.

Table-5.10: Total head losses

Types of Intake	Bend losses (H_B) in m	Friction losses (H_F) in m	Contraction losses (H_C) in m	Stoplog and Gate slot losses (H_S) in m	Trash rack losses(H_T) in m	Total losses (H_L) in m
Intake-1	0.00369	0.001050	0.0053	0.001878	0.022079	0.03394
Intake-2	0.00318	0.001159	0.0100	0.001878	0.024168	0.04037
Intake-3	0.00282	0.001581	0.00	0.002355	0.032590	0.03934
Intake-4	0.00158	0.000802	0.0053	0.002052	0.022104	0.03183

5.6 COST ESTIMATION

Cost estimation is carried out for different intakes by the methodology elaborated in article 5.6. Total cost for different intakes is given in table 5.12.

Table-5.12: Cost for intakes

Types of Intake	Volume of RCC required in sub structure in m^3	Volume of RCC required in super structure in m^3	Total RCC required in intake structure in m^3	Cost of RCC used in substructure in Rs.	Cost of RCC used in super structure in Rs.	Total cost of material in Rs.
Intake-1	115.33	190.87	306.2	553238	1101510	1654748
Intake-2	108.58	190.03	298.61	520858	1096663	1617521
Intake-3	96	170.54	266.54	460512	984186	1444698
Intake-4	72.18	112.64	184.82	346247	650045	996292

CHAPTER-6

RESULTS AND DISCUSSIONS

6.1 HYDRAULIC ANALYSIS OF INTAKE STRUCTURES

Under hydraulic analysis of intake structure head loss and vortex prevention analysis is considered.

6.1.1 Head Loss Analysis

Fig.6.1 present the different head losses for worked out for different intakes. Value of each type of head loss can be compared for various intakes. Fig. 6.1 shows that the trash rack losses are the major part of head losses. Friction losses form the smallest part of the total losses. Trash rack loss is seen as maximum in case of Intake-3.

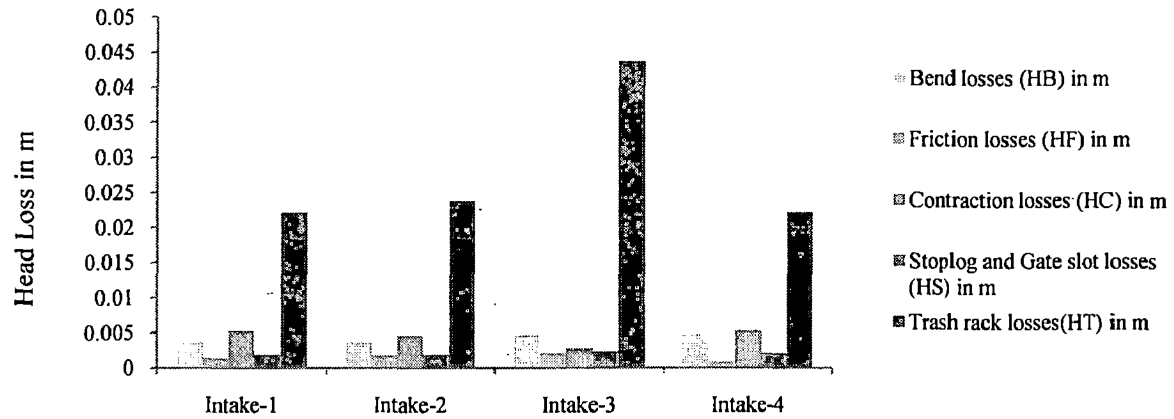


Fig. 6.1 Comparison of head losses for different intakes

Total head losses for different intakes are represented in bar chart and in graphical form in Fig. 6.1 and Fig. 6.2 respectively. Total head losses is seen as maximum for intake-3 and minimum for intake-1

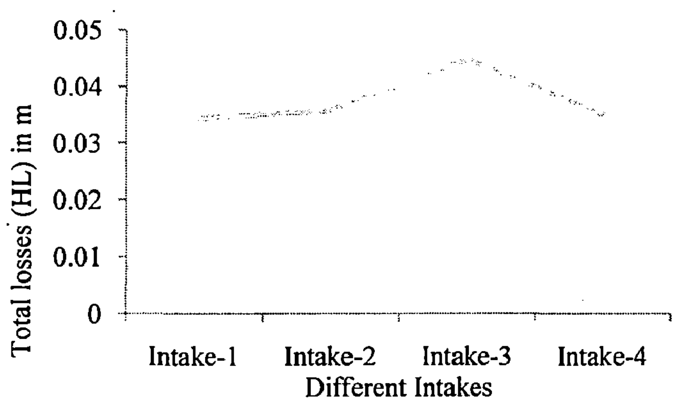


Fig. 6.2 Comparison of total head losses for different intakes

6.1.2 Vortex Prevention

Froude no. is worked out for all four intakes. The Froude no. should be less than 1/3 for power plants those have approach velocity less than 3m/s. For all intakes the value of h/d calculated is 1.1514 which is in the range of 1 to 1.5. So vortex is prevented.

6.1.3 Financial Analysis

Estimated cost for all intakes as worked out is presented in Fig. 6.2. Intake structure cost decreases from for intake-1 to intake-4. Its cost is more from intake-4 but still less than the all other intakes.

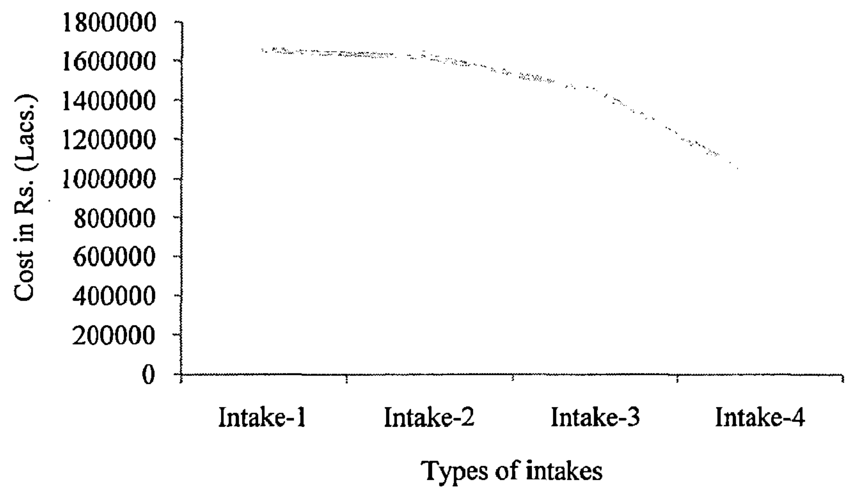


Fig. 6.3: Comparison of cost for

Due to shortening of the intakes, head loss increases which results in revenue loss per year. Shortening of the intake also results in reduction of the capital cost. Therefore to select the optimum option, financial analysis is carried out. For this purpose increase in total head loss is calculated in comparison to the Intake-1 for all four Intakes. Corresponding to change in head loss, energy loss and revenue loss is calculated.

Energy loss (kWh/year) = $9.81 \times Q \times \Delta H \times 0.87 \times 0.65 \times 8760$

Where ΔH is increase in head loss, Q =90 m^3/s and 0.87 is the overall efficiency. Plant load factor is assumed 0.65. 8760 are the total hours in a year.

Intake-1 is considered as base. Increase in head loss, energy loss (kWh/year) and Revenue loss (Rs./year) as compared to the intake-1 for different intakes is worked out and shown in Table 5.5. The selling price of energy is different in different state and now governed by the respective regulatory commissions. In different state the selling price of electricity varies from Rs.2.40 to Rs.4.50. For comparison purpose an average value of Rs. 3.50 per unit which is prevailing in most of the states is taken.

Table-6.1: Energy loss (kWh/year) and revenue loss (Rs. /year)

Types of intake	Total head loss difference in comparison to Intake-1 (m)	Energy loss difference in comparison to Intake-1 (kWh/year)	Revenue losses difference in comparison to Intake-1 (Rs./year)
Intake-2	0.00643	28122.84	20420.75
Intake-3	0.00540	23617.94	33753.97
Intake-4	0.00630	2755.46	9644

For comparison, revenue loss due to head loss is capitalized. For capitalizing the revenue loss, annual cost is taken as 18% of the capital cost as described below.

Interest rate as prevailing-12%

Operation and maintenance cost including insurance - 2%

Depreciation - 4%

Total annual expenditure -18%

Capitalized revenue losses (Rs.) due to head loss and Capital cost saving in comparison of Intake-1(Rs.) is shown in Table 6.2.

Table-6.2: Capitalized revenue losses (Rs.) and capital cost saving (Rs.)

Types of intake	Capitalized revenue losses(Rs.)	Capital cost saving in comparison of Intake-1(Rs.)	Difference
Intake-2	917216	37227	-879989
Intake-3	1358344	210050	-1148294
Intake-4	230305	658456	428151

Table presents that the intake-4 is optimum as it gives maximum saving in cost. Intake-3 is the second best option. Intake-2 show the cost difference as negative due to more head losses.

CHAPTER 7
CONCLUSIONS

7.1 CONCLUSION

A study on bulb turbine has been carried out and possibilities of simplifications and shortening of the intakes are investigated. A typical layout of bulb turbine power house is selected having discharge and net head as 90 m^3/s and 7 m respectively. Installed capacity and diameter has been worked out to be 2×5 MW and 3.90 m respectively. Four intakes of different geometry have been designed and dimensions of intakes are considered in terms of runner diameter. Based on the study following conclusions are drawn.

i. A study of bulb turbine covering its general features, advantages and disadvantages over other type of turbines under the similar site conditions has been made. It is observed that straight flow passages of bulb turbine results in low hydraulic losses and better efficiency.

ii. A study for the possible simplifications size reductions for intakes of bulb turbine showed that significant simplifications and size reduction is possible without much increasing the head losses and adversely affecting the flow conditions.

iii. Four intakes of different designs have been analyzed and designated as given in Table 7.1.

Table- 7.1: Different intakes

Types of intakes	Length of intake	Height of intake	Trash rack orientation
Intake-1	2.26 D	2.365 D	72^0
Intake-2	2.50 D	2.365 D	75^0
Intake-3	2.50 D	2.220D	80^0
Intake-4	1.53 D	2.305 D	85^0

iv. Total losses which comprises of bend losses, friction losses, contraction losses, slot losses and trash rack losses, has been worked out. Capital cost is estimated for each intake.

v. It is found that head loss is minimum in case of Intake-4 which is 0.035 m and maximum in Intake-3 having the value of 0.045 m.

vi. Capital cost of intake-1 comes out to be is Rs. 16.54 lacs which is maximum and minimum capital cost is Rs. 9.96 lacs in case of Intake-4.

vii. The revenue loss is capitalized and compared with the capital cost of the intakes. It is found that Intake-4 is optimum intake .

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