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HYDRO POWER PLANT TECHNOLOGY

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CHAPTER 1

CHAPTER 1

INTRODUCTION TO HYDROPOWER

1.1 INTRODUCTION

This chapter describes hydropower technology. It starts with a brief historical overview of how the technology has evolved (Section 1.1), a discussion of resource potential and how it may be affected by climate change (Section 1.2), and a description of the technology (Section 1.3) and its social and environmental impacts (Section 1.6). Also included is a summary of the present global and regional status of the hydropower industry (Section 1.4) and the role of hydropower in the broader energy system (Section 1.5), as well as a summary of the prospects for technology improvement (Section 1.7), cost trends (Section 1.8), and potential deployment in both the near term (2020) and long term (2050) (Section 1.9). The chapter also covers the integration of hydropower into broader water management solutions (Section 1.10). In this chapter, the focus is largely on the generation and storage of electrical energy from water; the use of hydropower in meeting mechanical energy demands is covered only peripherally.

1.1.1 Source of energy

Hydropower is generated from water moving in the hydrological cycle, which is driven by solar radiation. Incoming solar radiation is absorbed at the land or sea surface, heating the surface and creating evaporation where water is available. A large percentage—close to 50% of all the solar radiation reaching the Earth's surface—is used to evaporate water and drive the hydrological cycle. The potential energy embedded in this cycle is therefore huge, but only a very limited amount may be technically developed. Evaporated water moves into the atmosphere and increases the water vapor content in the air. Global, regional and local wind systems, generated and maintained by spatial and temporal variations in the solar energy input, move the air and its vapor content over the surface of the Earth, up to thousands of kilometers from the origin of evaporation. Finally, the vapor condenses and falls as precipitation, about 78% on oceans and 22% on land. This creates a net transport of water from the oceans to the land surface of the Earth, and an equally large flow of water back to the oceans as river and groundwater runoff. It is the flow of water in rivers that can be used to generate hydropower, or more precisely, the energy of water moving from higher to lower elevations on its way back to the ocean, driven by the force of gravity.

1.1.2 History of hydropower development

Prior to the widespread availability of commercial electric power, hydropower was used for irrigation and operation of various machines, such as watermills, textile machines and sawmills. By using water for power generation, people have worked with nature to achieve a better lifestyle. The mechanical power of falling water is an old resource used for services and productive uses. It was used by the Greeks to turn water wheels for grinding wheat into flour more than 2,000 years ago. In the 1700s, mechanical hydropower was used extensively for milling and pumping. During the 1700s and 1800s, water turbine development continued. The first hydroelectric power plant was installed in Cragside, in Grand Rapids, Michigan, when a dynamo driven by a water turbine was used to provide theatre and storefront lighting. In 1881, a brush dynamo connected to a turbine in a flour mill provided street lighting at Niagara Falls, New York. The breakthrough came when the electric generator was coupled to the turbine and thus the world's first hydroelectric station (of 12.5 kW capacity) was commissioned on 30 September 1882 on Fox River at the Vulcan Street Plant, Appleton, Wisconsin, USA, lighting two paper mills and a residence.

Early hydropower plants were much more reliable and efficient than the fossil fuel-fi red plants of the day (Baird, 2006). This resulted in a proliferation of small- to medium-sized hydropower stations distributed wherever there was an adequate supply of moving water and a need for electricity. As electricity demand grew, the number and size of fossil fuel, nuclear and hydropower plants increased. In parallel, concerns arose around environmental and social impacts (Thaulow et al., 2010).

Hydropower plants (HPP) today span a very large range of scales, from a few watts to several GW. The largest projects, Itaipu in Brazil with 14,000 MW 2 and Three Gorges in China with 22,400 MW,3 both produce between 80 to 100 TWh /yr. (288 to 360 PJ/yr.). Hydropower projects are always site-specifi c and thus designed according to the river system they inhabit. Historical regional hydropower generation from 1965 to 2009 is shown in Figure 1.1.

The great variety in the size of hydropower plants gives the technology the ability to meet both large centralized urban energy needs as well as decentralized rural needs. Though the primary role of hydropower in the global energy supply today is in providing electricity generation as part of centralized energy networks, hydropower plants also operate in isolation and supply independent systems, often in rural and remote areas of the world. Hydro energy can also be used to meet mechanical energy needs, or to provide space heating and cooling. More recently hydroelectricity has also been investigated for use in the electrolysis process for hydrogen fuel production, provided there is abundance of hydropower in a region and a local goal to use hydrogen as fuel for transport (Andreassen et al., 2002; Yumurtacia and Bilgen, 2004; Silva et al., 2005).

Hydropower plants do not consume the water that drives the turbines. The water, after power generation, is available for various other essential uses. In fact, a significant proportion of hydropower projects are designed for multiple purposes (see Section 1.10.2). In these instances, the dams help to prevent or mitigate floods and droughts, provide the possibility to irrigate agriculture, supply water for domestic, municipal and industrial use, and can improve conditions for navigation, fishing, tourism or leisure activities. One aspect often overlooked when addressing hydropower and the multiple uses of water is that the power plant, as a generator of revenue, in some cases can help pay for the facilities required to develop other water uses that might not generate sufficient direct revenues to finance their construction.

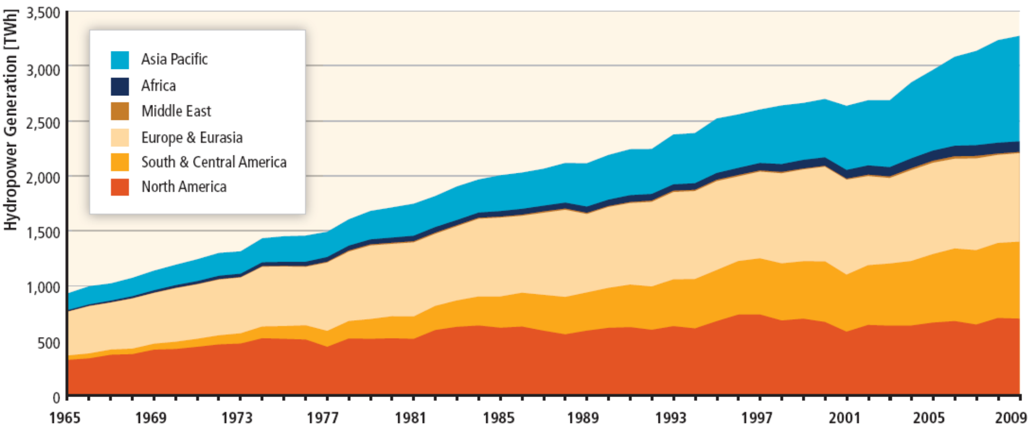


Figure 1. 1 Hydropower generation (TWh) by region (BP, 2010).

1.2 Resource potential

Hydropower resource potential can be derived from total available flow multiplied by head and a conversion factor. Since most precipitation usually falls in mountainous areas, where elevation differences (head) are the largest, the largest potential for hydropower development is in mountainous regions, or in rivers coming from such regions. The total annual runoff has been estimated as 47,000 km3, out of which 28,000 km3 is surface runoff, yielding a theoretical potential for hydropower generation of 41,784 TWh/yr.. (147 EJ/yr..) (Rogner et al., 2004). This value of theoretical potential is similar to a more recent estimate of 39,894 TWh/yr.. (144 EJ/yr..) (IJHD, 2010). Section 1.2.1 discusses the global technical potential, considering that gross theoretical potential is of no practical value and what is economically feasible is variable depending on energy supply and pricing, which can vary with time and by location.

1.2.1 Global Technical Potential

The International Journal on Hydropower & Dams 2010 World Atlas & Industry Guide (IJHD, 2010) provides the most comprehensive inventory of current hydropower installed capacity and annual generation, and hydropower resource potential. The Atlas provides three measures of hydropower resource potential, all in terms of annual generation (TW/yr.): gross theoretical, technically feasible, 4 and economically feasible. The total worldwide technical potential for hydropower is estimated at 14,576 TWh/yr.(52.47 EJ/yr.), over four times the current worldwide annual generation (IJHD, 2010).

This technical potential corresponds to a derived estimate of installed capacity of 3,721 GW.6 Technical potentials in terms of annual generation and estimated capacity for the six world regions7 are shown in Figure 1.2. Pie charts included in the figure provide a comparison of current annual generation to technical potential for each region and the percentage of undeveloped potential compared to total technical potential. These charts illustrate that the percentages of undeveloped potential range from 47% in Europe and North America to 92% in Africa, indicating large opportunities for hydropower development worldwide.

There are several notable features of the data in Figure 1.2. North America and Europe, which have been developing their hydropower resources for more than a century, still have sufficient technical potential to double their hydropower generation, belying the perception that the hydropower resources in these highly developed parts of the world are exhausted. However, how much of this untapped technical potential is economically feasible is subject to time-dependent economic conditions.

Actual development will also be impacted by sustainability concerns and related policies. Notably, Asia and Latin America have comparatively large technical potentials and, along with Australasia/Oceania, the fraction of total technical potential that is undeveloped is quite high in these regions. Africa has a large technical potential and could develop11 times its current level of hydroelectric generation in the region. An overview of regional technical potentials for hydropower is given in Table 1.1.

Understanding and appreciation of hydropower technical potential can also be obtained by considering the current (2009) total regional hydropower installed capacity and annual generation shown in Figure 1.3. The reported worldwide total installed hydropower capacity is 926 GW producing a total annual generation of 3,551 TWh/yr. (12.8 EJ/yr.) in 2009.

Figure 1.3 also includes regional average capacity factors calculated using current regional total installed capacity and annual generation (capacity factor = generation/(installed capacity x 8,760 hrs)).

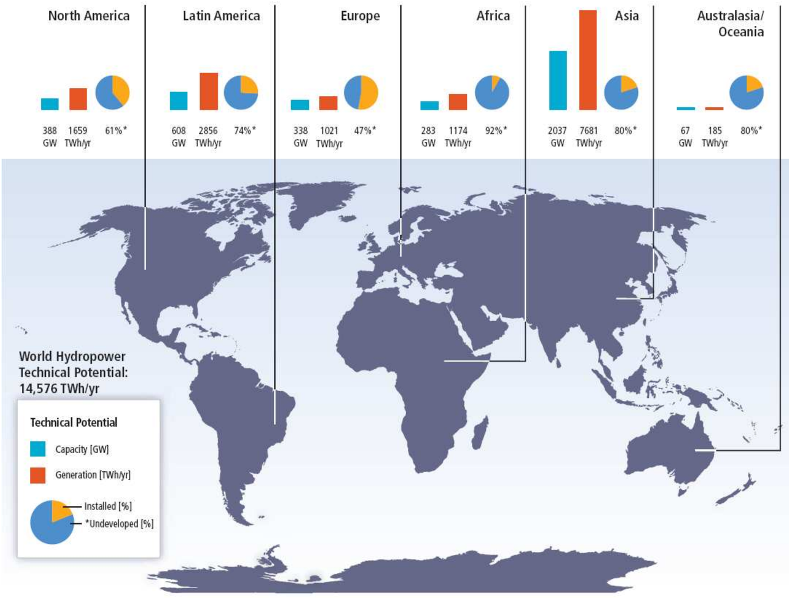


Figure 1. 2 Regional hydropower technical potential in terms of annual generation and installed capacity, and percentage of undeveloped technical potential in 2009. Source: IJHD (2010).

It is interesting to note that North America, Latin America, Europe and Asia have the same order of magnitude of total installed capacity while Africa and Australasia/Oceania have an order of magnitude less—Africa due in part to the lack of available investment capital and Australasia/ Oceania in part because of size, climate and topography. The average capacity factors are in the range of 32 to 55%. Capacity factor can be indicative of how hydropower is employed in the energy mix (e.g., peaking versus base-load generation), water availability, or an opportunity for increased generation through equipment upgrades and operation optimization. Generation increases that have been achieved by equipment upgrades and operation optimization have generally not been assessed in detail, but are briefly discussed in Sections 1.3.4 and 1.8.

The regional technical potentials presented above are for conventional hydropower corresponding to sites on natural waterways where there is significant topographic elevation change to create useable hydraulic head. Hydrokinetic technologies that do not require hydraulic head but rather extract energy in-stream from the current of a waterway are being developed. These technologies increase the potential for energy production at sites where conventional hydropower technology cannot operate. Nontraditional sources of hydropower are also not counted in the regional technical potentials presented above. Examples are constructed waterways such as water supply and treatment systems, aqueducts, canals, effluent streams and spillways. Applicable conventional and hydrokinetic technologies can produce energy using these resources. While the total technical potentials of in-stream and constructed waterway resources have not been assessed, they may prove to be significant given their large extent.

Taple 1. 1. Regional hydropower technical potential in terms of annual generation and installed capacity (GW); and current generation, installed capacity, average capacity factors in percent and resulting undeveloped potential as of 2009. Source: IJHD (2010).

World region	Technical potential, annual generation TWh/yr (EJ/yr)	Technical potential, installed capacity (GW)	2009 Total generation TWh/yr (EJ/yr)	2009 Installed capacity (GW)	Un-developed potential (%)	Average regional capacity factor (%)
North America	1,659 (5.971)	388	628 (2.261)	153	61	47
Latin America	2,856 (10.283)	608	732 (2.635)	156	74	54
Europe	1,021 (3.675)	338	542 (1.951)	179	47	35
Africa	1,174 (4.226)	283	98 (0.351)	23	92	47
Asia	7,681 (27.651)	2,037	1,514 (5.451)	402	80	43
Australasia/Oceania	185 (0.666)	67	37(0.134)	13	80	32
World	**14,576 (52.470)**	**3,721**	**3,551 (12.783)**	**926**	**75**	**44**

Figure 1. 3 reference pipe section for sizing.

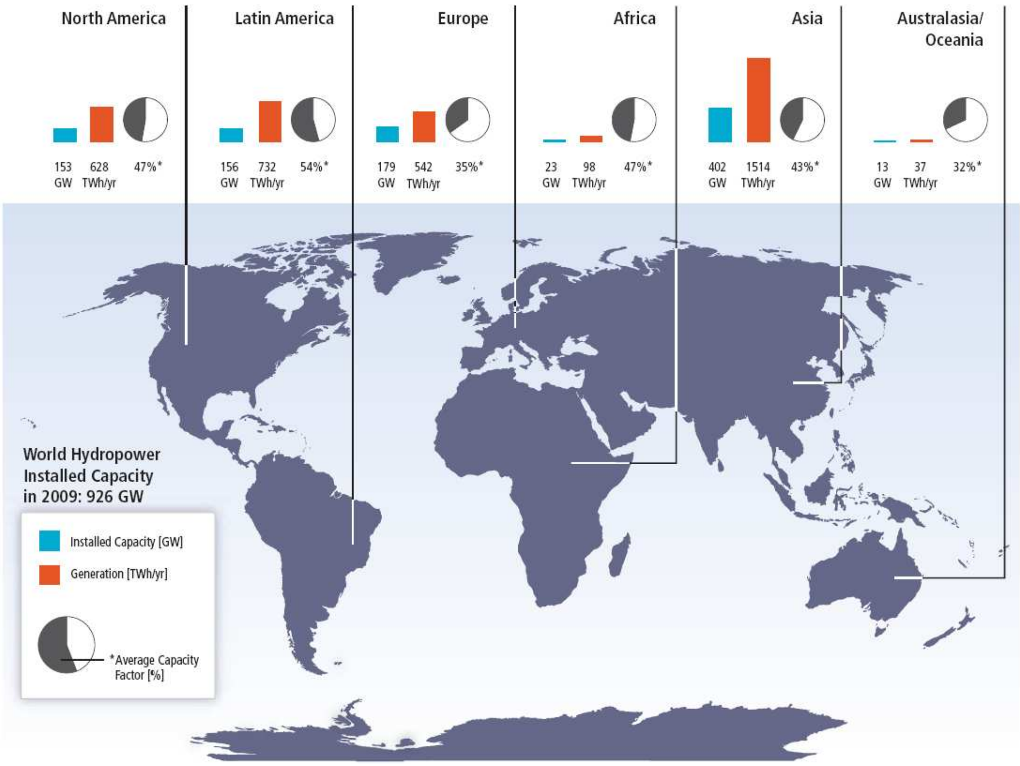


Figure 1. 4 Total regional installed hydropower capacity and annual generation in 2009, and average regional capacity factors (derived as stated above). Source: IJHD (2010).

1.2.2 Possible impact of climate change on resource potential

The resource potential for hydropower is currently based on historical data for the present climatic conditions. With a changing climate, this resource potential could change due to:

• Changes in river flow (runoff) related to changes in local climate, particularly in precipitation and temperature in the catchment area. This may lead to changes in runoff volume, variability of fl ow and seasonality of the flow (e.g., by changing from spring/summer high fl ow to more winter flow), directly affecting the resource potential for hydropower generation.

• Changes in extreme events (floods and droughts) may increase the cost and risk for the hydropower projects.

• Changes in sediment loads due to changing hydrology and extreme events. More sediment could increase turbine abrasions and decrease efficiency. Increased sediment load could also fi ll up reservoirs faster and decrease the live storage, reducing the degree of regulation and decreasing storage services.

The work of IPCC Working Group II (reported in IPCC, 2007b) includes a discussion of the impact of climate change on water resources. Later, a technical paper on water was prepared based on the material included in the previous IPCC reports as well as other sources (Bates et al., 2008). The information presented in this section is mostly based on these two sources, with a few additions from more recent papers and reports, as presented, for example, in a recent review by Hamududu et al. (2010).

1.2.2.1 Projected changes in precipitation and runoff

A wide range of possible future climatic projections have been presented, with corresponding variability in projection of precipitation and runoff (IPCC, 2007c; Bates et al., 2008). Climate projections using multi model ensembles show increases in globally averaged mean water vapor, evaporation and precipitation over the 21st century. At high latitudes and in part of the tropics, nearly all models project an increase in precipitation, while in some subtropical and lower mid-latitude regions, precipitation is projected to decrease. Between these areas of robust increase or decrease, even the sign of projected precipitation change is inconsistent across the current generation of models (Bates et al., 2008).

Changes in river flow due to climate change will primarily depend on changes in volume and timing of precipitation, evaporation and snowmelt. A large number of studies of the effect on river flow have been published and were summarized in IPCC (2007b). Most of these studies use a catchment hydrological model driven by climate scenarios based on climate model simulations. Before data can be used in the catchment hydrological models, it is necessary to downscale data, a process where output from the global climate model is converted to corresponding climatic data in the catchments. Such downscaling can be both temporal and spatial, and it is currently a high priority research area to find the best methods for downscaling.

A few global-scale studies have used runoff simulated directly by climate models (Egré and Milewski, 2002; IPCC, 2007b). The results of these studies show increasing runoff in high latitudes and the wet tropics and decreasing runoff in mid-latitudes and some parts of the dry tropics. Figure 1.4 illustrates projected changes in runoff by the end of the century, based on the IPCC A1B scenario8 (Bates et al., 2008).

Uncertainties in projected changes in the hydrological systems arise from internal variability in the climatic system, uncertainty about future greenhouse gas and aerosol emissions, the translations of these emissions into climate change by global climate models, and hydrological model uncertainty. Projections become less consistent between models as the spatial scale decreases. The uncertainty of climate model projections for freshwater assessments is often taken into account by using multi-model ensembles (Bates et al., 2008). The multi-model ensemble approach is, however, not a guarantee of reducing uncertainty in mathematical models.

Global estimates as shown in Figure 1.4 represent results at a large scale, and cannot be applied to shorter temporal and smaller spatial scales. In areas where rainfall and runoff are very low (e.g., desert areas), small changes in runoff can lead to large percentage changes. In some regions, the sign of projected changes in runoff differs from recently observed trends. Moreover, in some areas with projected increases in runoff, different seasonal effects are expected, such as increased wet season runoff and decreased dry season runoff. Studies using results from fewer climate models can be considerably different from the results presented here (Bates et al., 2008).

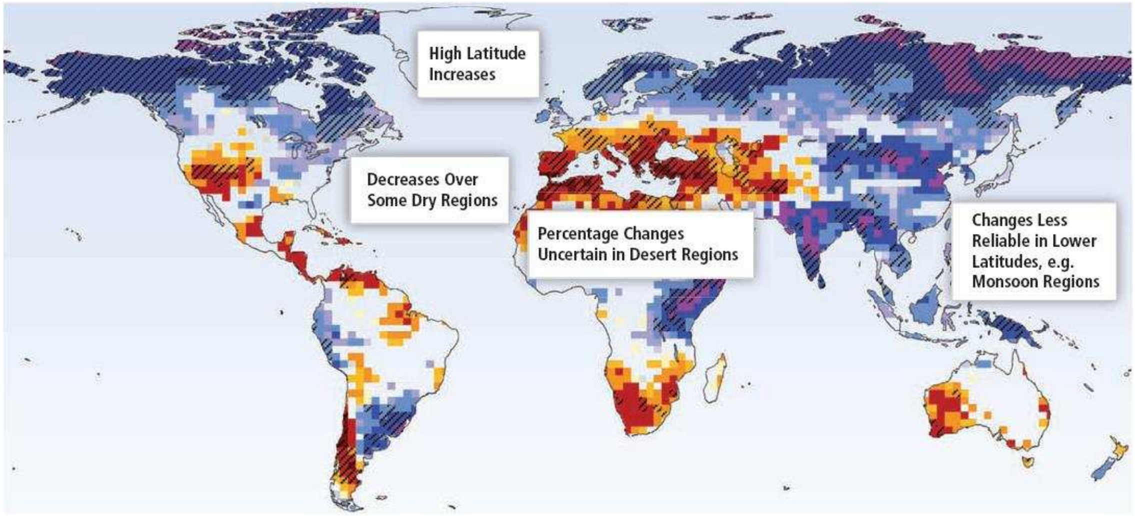


Figure 1. 5 Large-scale changes in annual runoff (water availability, in percent) for the period 2090 to 2099, relative to 1980 to 1999. Values represent the median of 12 climate model projections using the SRES A1B scenario. White areas are where less than 66% of the 12 models agree on the sign of change and hatched areas are where more than 90% of models agree on the sign of change. Source: IPCC (2007a).

1.2.2.2 Projected impacts on hydropower generation

Though the average global or continent-wide impacts of climate change on hydropower resource potential might be expected to be relatively small, more significant regional and local effects are possible. Hydropower resource potential depends on topography and the volume, variability and seasonal distribution of runoff. Not only are these regionally and locally determined, but an increase in climate variability, even with no change in average runoff, can lead to reduced hydropower production unless more reservoir capacity is built and operations are modified to account for the new hydrology that may result from climate change.

In order to make accurate quantitative predictions of regional effects it is therefore necessary to analyze both changes in average flow and changes in the temporal distribution of fl ow, using hydrological models to convert time series of climate scenarios into time series of runoff scenarios. In catchments with ice, snow and glaciers it is of particular importance to study the effects of changes in seasonality, because a warming climate will often lead to increasing winter runoff and decreasing runoff in spring and summer. A shift in winter precipitation from snow to rain due to increased air temperature may lead to a temporal shift in peak flow and winter conditions (Stickler and Alfredsen, 2009) in many continental and mountain regions. The spring snowmelt peak would then be brought forward or eliminated entirely, with winter flow increasing. As glaciers retreat due to warming, river flows would be expected to increase in the short term but decline once the glaciers disappear (Bates et al., 2008; Milly et al., 2008).

Summarizing available studies up to 2007, IPCC (2007b) and Bates et al. (2008) found examples of both positive and negative regional effects on hydropower production, mainly following the expected changes in river runoff. Unfortunately, few quantitative estimates of the effects on technical potential for hydropower were found. The regional distribution of studies was also skewed, with most studies done in Europe and North America, and a weak literature base for most developing country regions, in particular for Africa. The summary below is based on findings summarized in Bates et al. (2008) and IPCC (2007b) unless additional sources are given. In Africa, the electricity supply in a number of states is largely based on hydroelectric power. However, few available studies examine the impacts of climate change on hydropower resource potential in Africa. Observations deducted from general predictions for climate change and runoff point to a reduction in hydropower resource potential with the exception of East Africa (Hamududu et al., 2010).

In major hydropower-generating Asian countries such as China, India, Iran, Tajikistan etc., changes in runoff are found to potentially have a significant effect on the power output. Increased risks of landslides and glacial lake outbursts, and impacts of increased variability, are of particular concern to Himalayan countries (Agrawala et al., 2003). The possibility of accommodating increased intensity of seasonal precipitation by increasing storage capacities may become of particular importance (Iimi, 2007).

In Europe, by the 2070s, hydropower potential for the whole of Europe has been estimated to potentially decline by 6%, translated into a 20 to 50% decrease around the Mediterranean, a 15 to 30% increase in northern and Eastern Europe, and a stable hydropower pattern for western and central Europe (Lehner et al., 2005).

In New Zealand, increased westerly wind speed is very likely to enhance wind generation and spill over precipitation into major South Island watersheds, and to increase winter rain in the Waikato catchment. Warming is virtually certain to increase melting of snow, the ratio of rainfall to snowfall, and to increase river flows in winter and early spring. This is very likely to increase hydroelectric generation during the winter peak demand period, and to reduce demand for storage.

In Latin America, hydropower is the main electrical energy source for most countries, and the region is vulnerable to large-scale and persistent rainfall anomalies due to El Niño and La Niña, as observed in Argentina, Colombia, Brazil, Chile, Peru, Uruguay and Venezuela. A combination of increased energy demand and droughts caused a virtual breakdown of hydroelectricity in most of Brazil in 2001 and contributed to a reduction in gross domestic product (GDP). Glacier retreat is also affecting hydropower generation, as observed in the cities of La Paz and Lima.

In North America, hydropower production is known to be sensitive to total runoff, to its timing, and to reservoir levels. During the 1990s, for example, Great Lakes levels fell as a result of a lengthy drought, and in 1999, hydropower production was down significantly both at Niagara and Sault St. Marie. For a 2°C to 3°C warming in the Columbia River Basin and BC Hydro service areas, the hydroelectric supply under worst case water conditions for winter peak demand is likely to increase (high confidence). Similarly, Colorado River hydropower yields are likely to decrease significantly, as will Great Lakes hydropower. Northern Québec hydropower production would be likely to benefit from greater precipitation and more open-water conditions, but hydropower plants in southern Québec would be likely to be affected by lower water levels. Consequences of changes in the seasonal distribution of flows and in the timing of ice formation are uncertain.

In a recent study (Hamududu and Killingtveit, 2010), the regional and global changes in hydropower generation for the existing hydropower system were computed, based on a global assessment of changes in river flow by 2050 (Milly et al., 2005, 2008) for the SRES A1B scenario using 12 different climate models. The computation was done at the country or political region (USA, Canada, Brazil, India, China, and Australia) level, and summed up to regional and global values (see Table 1.2).

Taple 1. 2 Power generation capacity in GW and TWh/yr (2005) and estimated changes (TWh/yr) due to climate change by 2050. Results are based on an analysis using the SRES A1B scenario in 12 different climate models (Milly et al., 2008), UNEP world regions and data for the hydropower system in 2005 (US DOE, 2009) as presented in Hamududu and Killingtveit (2010).

REGION	Power Generation Capacity (2005)		Change by 2050 TWh/yr (PJ/yr)
	GW	TWh/yr (PJ/yr)	
Africa	22	90 (324)	0.0 (0)
Asia	246	996 (3,586)	2.7 (9.7)
Europe	177	517 (1,861)	-0.8 (-2.9)
North America	161	655 (2,358)	0.3 (≈1)
South America	119	661 (2,380)	0.3 (≈1)
Oceania	13	40 (144)	0.0 (0)
TOTAL	737	2931 (10,552)	2.5 (9)

In general the results given in Table 1.2 are consistent with the (mostly qualitative) results given in previous studies (IPCC, 2007b; Bates et al., 2008). For Europe, the computed reduction (-0.2%) has the same sign, but is less than the -6% found by Lehner et al. (2005). One reason could be that Table 1.2 shows changes by 2050 while Lehner et al. (2005) give changes by 2070, so a direct comparison is difficult. It can be concluded that the overall impacts of climate change on the existing global hydropower generation may be expected to be small, or even slightly positive. However, results also indicated substantial variations in changes in energy production across regions and even within countries (Hamududu and Killingtveit, 2010).

Insofar as a future expansion of the hydropower system will occur incrementally in the same general areas/watersheds as the existing system, these results indicate that climate change impacts globally and averaged across regions may also be small and slightly positive. Still, uncertainty about future impacts as well as increasing difficulty of future systems operations may pose a challenge that must be addressed in the planning and development of future HPP (Hamududu et al., 2010). Indirect effects on water availability for energy purposes may occur if water demand for other uses such as irrigation and water supply for households and industry rises due to the climate change. This effect is difficult to quantify, and it is further discussed in Section 1.10.

1.3 Classification of hydropower plants

Head and also installed capacity (size) are often presented as criteria for the classification of hydropower plants. The main types of hydropower, however, are run-of-river, reservoir (storage hydro), pumped storage, and in-stream technology. Classification by head and classification by size are discussed in Section 1.3.1. The main types of hydropower are presented in

Section 1.3.2. Maturity of the technology, status and current trends in technology development, and trends in renovation and modernization follow in Sections 1.3.3 and 1.3.4 respectively.

The classification of hydro-electric plants is based upon (Fig. 2.1):

(a) Quantity of water available

(b) Available head

(c) Nature of load

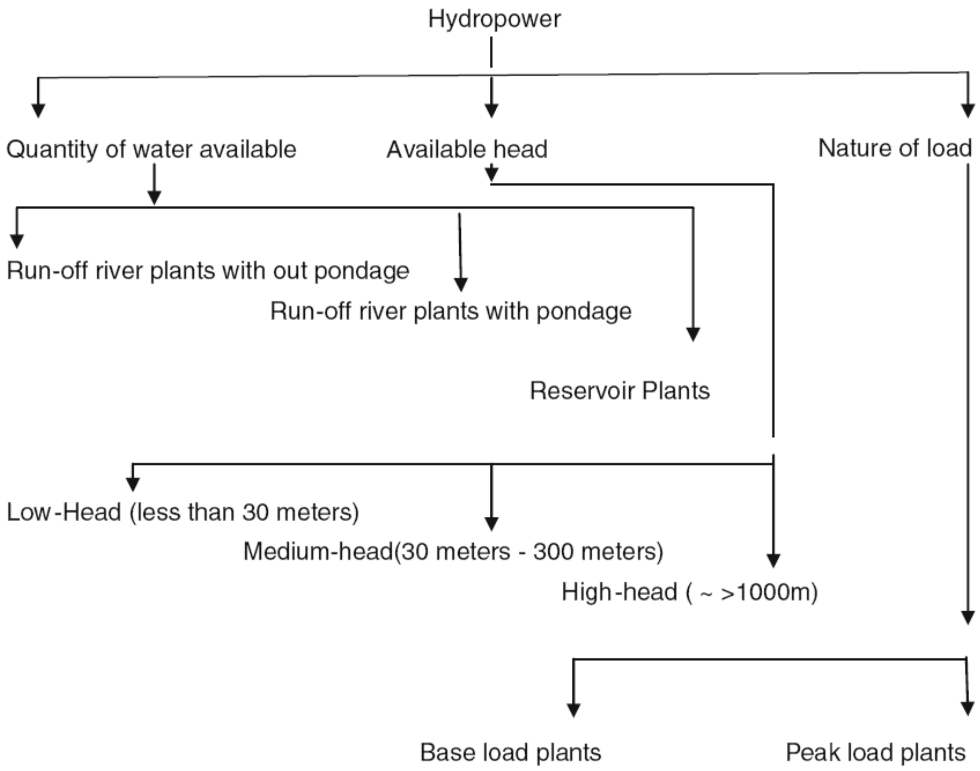


Figure 1. 6 Showing the classification overview of hydro-power plants

1.3.1 Classification with respect to quantity of water available

1- run-off-river
   1-1- run-off-river without pondage
   1-2- run-off-river with pondage
2- reservoir plants
   2-1- storage hydropower.
   2-2- pumped storage

1-1- Run-off river plants without poundage: These plants don't have storage or pond ages to store water; Run-off river plants without pondages uses water as it comes. The plant can use water as and when available. Since, generation capacity of these types of plants these plants depend on the rate of flow of water, during rainy season high flow rate may mean some quantity of water to go as waste while during low run-off periods, due to low flow rates, the generating capacity will be low.

1-2- Run-off river plants with pondage: In these plants, pondage allows storage of water during lean periods and use of this water during peak periods. Based on the size of the storage structure provided, it may be possible to cope with hour- to- hour fluctuations. This type of plant can be used on parts of the load curve as required, and is more useful than a plant without pondage. If pondage is provided, tail race conditions should be such that floods do not raise tail-race water level, thus reducing the head on the plant and impairing its effectiveness. This type of plant is comparatively more conscientious and its generating capacity is unabased on available rate of flow of water.

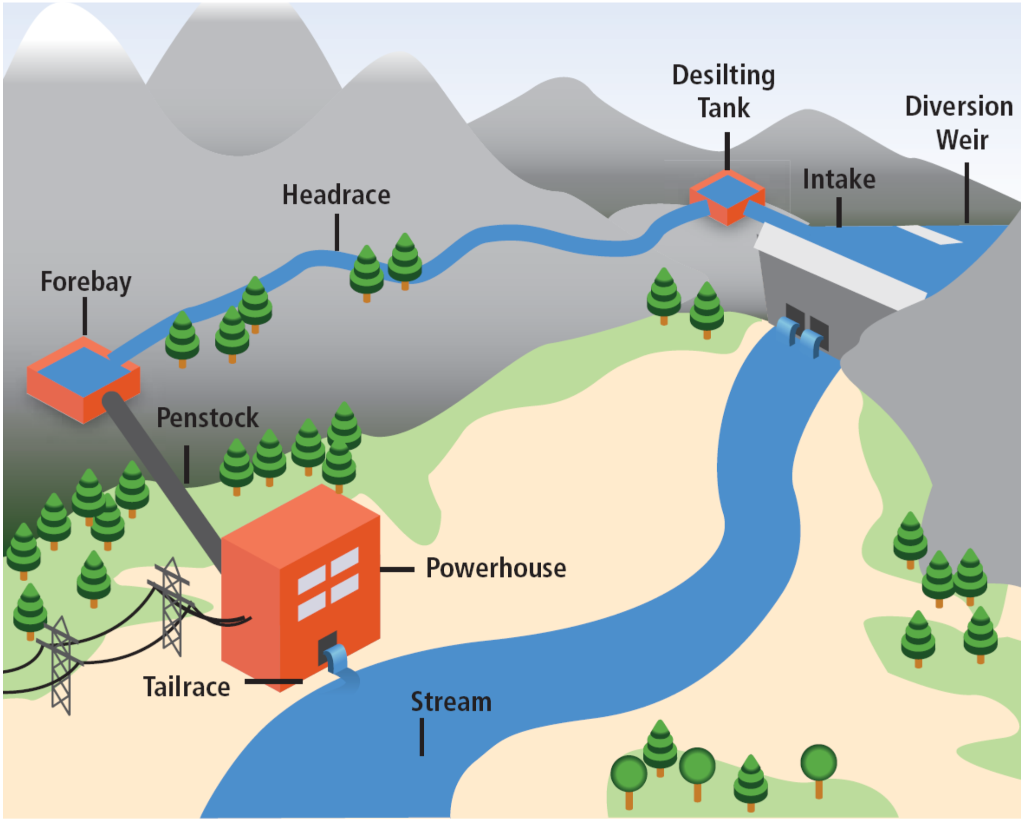


Figure 1. 7 Run-of-river hydropower plant.

2-1- storage hydropower: Hydropower projects with a reservoir are also called storage hydropower since they store water for later consumption. The reservoir reduces the dependence on the variability of infl ow. The generating stations are located at the dam toe or further downstream, connected to the reservoir through tunnels or pipelines. (Figure 1.6). The type and design of reservoirs are decided by the landscape and in many parts of the world are inundated river valleys where the reservoir is an artifi cial lake. In geographies with mountain plateaus, high-altitude lakes make up another kind of reservoir that often will retain many of the properties of the original lake. In these types of settings, the generating station is often connected to the lake serving as reservoir via tunnels coming up beneath the lake (lake tapping). For example, in Scandinavia, natural high-altitude lakes are the basis for high pressure systems where the heads may reach over 1,000 m. One power plant may have tunnels coming from several reservoirs and may also, where opportunities exist, be connected to neighboring watersheds or rivers. The design of the HPP and type of reservoir that can be built is very much dependent on opportunities offered by the topography.

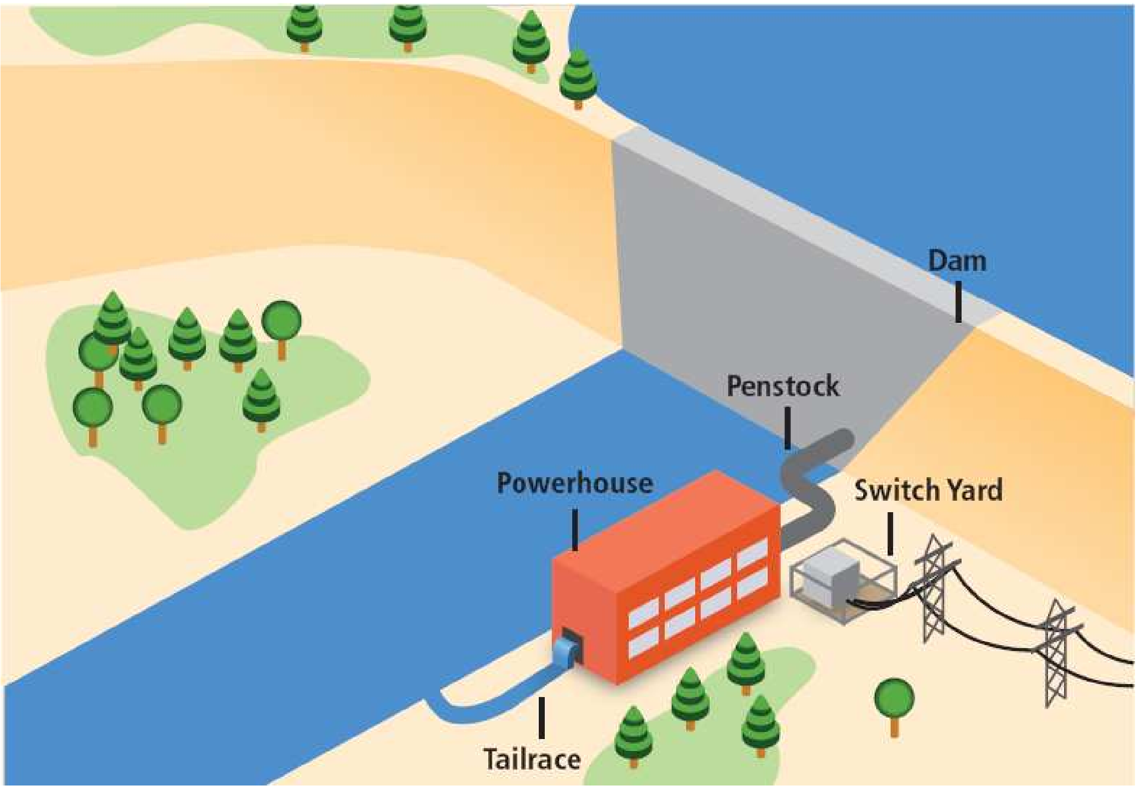


Figure 1. 8 Typical hydropower plant with reservoir

2-2- pumped storage: Pumped storage plants are not energy sources, but are instead storage devices. In such a system, water is pumped from a lower reservoir into an upper reservoir (Figure 1.7), usually during off-peak hours, while flow is reversed to generate electricity during the daily peak load period or at other times of need. Although the losses of the pumping process make such a plant a net energy consumer overall, the plant is able to provide large-scale energy storage system benefits. In fact, pumped storage is the largest-capacity form of grid energy storage now readily available worldwide (see Section 1.5.5).

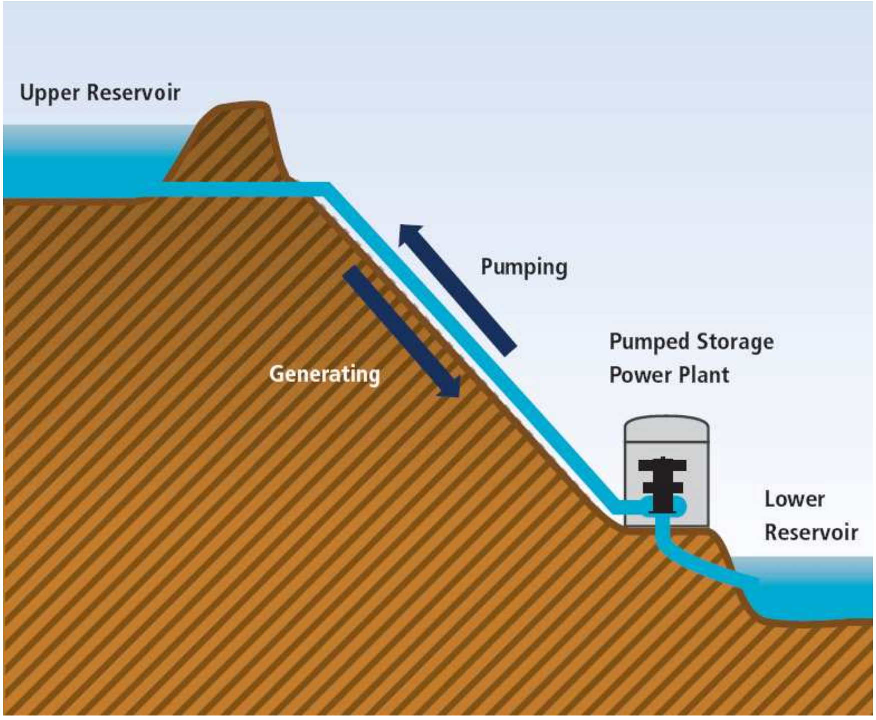


Figure 1. 9 Typical pumped storage project.

1.3.2 Classification According to Availability of Water Head

(i) Low-head (less than 30 m) hydro-electric plants: ‘‘Low head’’ hydro-electric plants are power plants which generally utilize heads of only a few meters or less. Power plants of this type may utilize a low dam or weir to channel water, or no dam and simply use the ‘‘run of the river’’. Run of the river generating stations cannot store water, thus their electric output varies with seasonal flows of water in a river. A large volume of water must pass through a low head hydro plant’s turbines in order to produce a useful amount of power. Hydro-electric facilities with a capacity of less than about 25 MW (1 MW = 1,000,000 W) are generally referred to as ‘‘small hydro’’, although hydro-electric technology is basically the same regardless of generating capacity.

(ii) Medium-head (30–300 m) hydro-electric plants: These plants consist of a large dam in a mountainous area which creates a huge reservoir. The Grand Coulee Dam on the Columbia River in Washington (108 m high, 1,270 m wide, and 9,450 MW) and the Hoover Dam on the Colorado River in Arizona/ Nevada (220 m high, 380 m wide, and 2000 MW) are good examples. These dams are true engineering marvels. In fact, the American Society of Civil Engineers as designated Hoover Dam as one of the seven civil engineering wonders of the modern world, but the massive lakes created by these dams are a graphic example of our ability to manipulate the environment—for better or worse. Dams are also used for flood control, irrigation, recreation, and often are the main source of potable water for many communities. Hydro-electric development is also possible in areas such as Niagra Falls where natural elevation changes can be used.

(iii) High-head hydro-electric plants: ‘‘High head’’ power plants are the most common and generally utilize a dam to store water at an increased elevation. The use of a dam to impound water also provides the capability of storing water during rainy periods and releasing it during dry periods. This results in the consistent and reliable production of electricity, able to meet demand. Heads for this type of power plant may be greater than 1,000 m. Most large hydro-electric facilities are of the high-head variety. High-head plants with storage are very valuable to electric utilities, because they can be quickly adjusted to meet the electrical demand on a distribution system.

1.3.3 Classification According to Nature of Load

(i) Peak Load Plants : The peak load plants are used to supply power at the peak demand phase. The pumped storage plants and Gas Turbine plants are this type of plants. Their efficiency varies between 60–70%.

(ii) Base load plants: A base load power plant is one that provides a steady flow of power regardless of total power demand by the grid. These plants run at all times through the year except in the case of repairs or scheduled maintenance.

1.4 Advantages of Hydroelectric Plants

The benefits of hydropower plants are manifold as described below:

• The running, operation and maintenance cost of this kind of plants are low.

• After the initial infrastructures are developed the energy is virtually free.

• The plants is totally free of pollution as no conventional fuels are required to be burned.

• The lifetime of generating plants are substantially long.

• Reliability is much more than wind, solar or wave power due to its easy availability and convertibility.

• Water can be stored above the dam ready to cope with peaks in demand.

• The uncertainties that arises due to unscheduled breakdowns are relatively infrequent and short in duration due to the simplicity and flexibility of the instruments.

• Hydro-electric turbine generators can be started and put ‘‘on-line’’ very rapidly.

• It is possible to produce electricity from hydro-electric power plant if flow is continuously available. Benkovic et al. (2013); Panic et al. (2013); Sharma and Awal (2013); Dursun and Cihan (2011) etc. has already discussed the benefits of the HPP in different aspect in their published literatures.

1.5 Disadvantages of Hydro-Electric Plants

But along with the advantages, the disadvantages (Bahadori et al. 2013; Chen et al. 2013; Jensen et al. 2002) of such renewable energy projects are also manifold but lesser than the other sources of infinite and also finite energy.

• The potential of hydro power depend on locations and if properly not selected may cause lots of hostility and absurdity during operational stage of the power plant.

• The dams are very expensive to build. However, many dams are also used for flood control or irrigation, so building costs can be shared.

• The capital cost of electrical instruments along with civil engineering works to be installed and cost of laying transmission lines is generally high.

• The impact on plant life due to the water quality and quantity downstream of hydro power plants are reported.

• The impact on residents and the environment may be unacceptable environmental and social activist if location is not optimally selected.

• Due to increase in water temperature and insertion of excess nitrogen into water at spillways health and migration of fish as well as other aquatic plants get effected.

• Due to the installation of reservoir in the flow paths the siltation rate get altered.

1.6 Types of Dams

A dam is a hydraulic structure of fairly impervious material built across a river to create a reservoir on its upstream side for impounding water for various purposes. A dam and a reservoir are complements of each other. Dams are generally constructed in the mountainous reach of the river where the valley is narrow and the foundation is good. Generally, a hydropower station is also constructed at or near the dam site to develop hydropower. Dams are probably the most important hydraulic structure built on the rivers. These are very huge structure and require huge money, manpower and time to construct.

1.6.1 Classification of Dams:

Based on Function Served:

1- Storage dams
2- Detention dams
3- Diversion dams
4- Debris dams
5- Coffer dams - a temp dam constructed for facilitating construction. It is an enclosure constructed around a site to exclude water so that the construction can be done in dry.

Based on Hydraulic Design

1- Overflow dams
2- Non-overflow dams

Based on Materials of Construction

1- Masonry dam
2- Concrete dam
3- Earth dam
4- Rockfill dam
5- Timber dam
6- Steel dam
7- Combined concrete-cum-earth dam
8- Composite dam.

Based on Rigidity

1- Rigid dams: A rigid dam is quite stiff. It is constructed of stiff materials such as concrete, masonry, steel and timber. These dams deflect and deform very little when subjected to water pressure and other forces.
2- Non-rigid dams: A non-rigid dam is relatively less stiff compared to a rigid dam. The dams constructed of earth and rockfill are non-rigid

dams. There are relatively large settlements and deformations in a non-rigid dam.

Note: Rockfill dams are actually neither fully rigid nor fully nonrigid. These are sometimes classified as semi-rigid dams.

Based on structural action

1- Gravity dams
2- Embankment dams
   2-1- Earth dams
   2-2- Rockfill dams
3- Arch dams
4- Buttress dams

1.6.1.1 Gravity dams.

A gravity dam resists the water pressure and other forces due to its weight (or gravitational forces). Usually it made of cement concrete and straight in plan. Gravity dam is approximately triangular in cross-section, with apex at the top. In the past, the gravity dams were made of stone masonry.

Advantages:

1- Gravity dams are quite strong, stable and durable.
2- Are quite suitable across moderately wide valleys and gorges having steep slopes where earth dams, if constructed, might slip.
3- Can be constructed to very great heights, provided good rock foundations are available.
4- Well adapted for use as an overflow spillway section. Earth dams cannot be used as an overflow section. Even in earth dams, the overflow section is usually a gravity dam.
5- Especially suited to such areas where there is very heavy downpour. The slopes of the earth dams might be washed away in such an area.
6- Maintenance cost of a gravity dam is very low.
7- Does not fail suddenly. There is enough warning of the imminent failure and the valuable property and human life can be saved to some extent.
8- Can be constructed during all types of climatic conditions.
9- sedimentation in the reservoir on the upstream of a gravity dam can be somewhat reduced by operation of deep-set sluices

Disadvantages

1- Gravity dams of great height can be constructed only on sound rock foundations. These cannot be constructed on weak or permeable foundations on which earth dams can be constructed.

2- Initial cost of a gravity dam is usually more than that of an earth dam. At the sites where good earth is available for construction and funds are limited, earth dams are better.
3- Usually take a longer time in construction than earth dams, especially when mechanized plants for batching, mixing and transporting concrete are not available.
4- Require more skilled labour than that in earth dams.
5- subsequent raising is not possible in a gravity dam

Figure 1. 10 Masonry gravity dam – non-overflow

Figure 1. 11 Concrete gravity dam with overflow section

1.6.1.2 Earth dam.

An earth dam is made of earth (or soil) and resists the forces exerted upon it mainly due to shear strength of the soil. It is usually built in wide valleys having flat slopes at flanks (abutments). It can be homogeneous when the height of the dam is not great. Also, it is of zoned sections, with an impervious zone (called core) in the middle and relatively pervious zones (called shells or shoulders) enclosing the impervious zone on both sides. Nowadays majority of dams constructed are of this type. The highest dams of the world are earth dams (Rongunsky dam Rusia, 325 m and Nurek dam, Rusia, 317 m) as well as the largest capacity dams (New Cornelia dam, USA and Tarbela dam, Pakistan).

Figure 1. 12 Earth dam

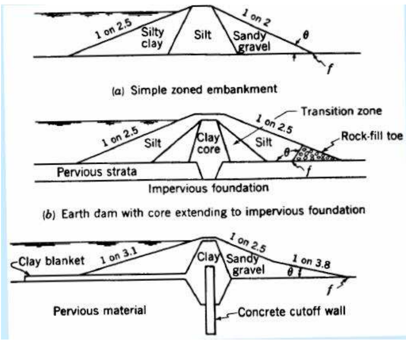


Figure 1. 13 Earth dam construction

Advantages

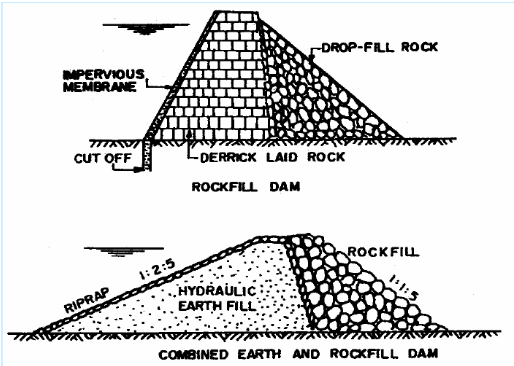
1- Are usually cheaper than gravity dams if suitable earth for construction is available near the site.
2- Can be constructed on almost all types of foundations, provided suitable measures of foundation treatment and seepage control are taken.
3- Can be constructed in a relatively short period.
4- Skilled labour is not required in construction of an earth dam.
5- Can be raised subsequently.
6- Are aesthetically more pleasing than gravity dams.
7- Are more earthquake-resistant than gravity dams.

Disadvantages

1- Are not suitable for narrow gorges with steep slopes.
2- Cannot be designed as an overflow section. A spillway has to be located away from the dam.
3- Cannot be constructed in regions with heavy downpour, as the slopes might be washed away.
4- Maintenance cost of an earth dam is quite high. It requires constant supervision.
5- Sluices cannot be provided in a high earth dam to remove slit.
6- Fails suddenly without any sign of imminent failure. A sudden failure causes havoc and untold miseries.

1.6.1.3 Rockfill dams

A rockfill dam is built of rock fragments and boulders of large size. An impervious membrane (cement concrete or asphaltic concrete or earth core) is placed on the rockfill on the upstream side to reduce the seepage through the dam. A dry rubble cushion is placed between the rockfill and the membrane for the distribution of water load and for providing a support to the membrane. side slopes of rockfill are usually kept equal to the angle of repose of rock (1.4:1 or 1.3:1). Rockfill dams are quite economical when a large quantity of rock is easily available near the site. Aswan high dam (110) in Egypt, while Mica dam (242 m, Canada), and Chicoasen dam (240 m, Maxico) are highest rockfill dams.



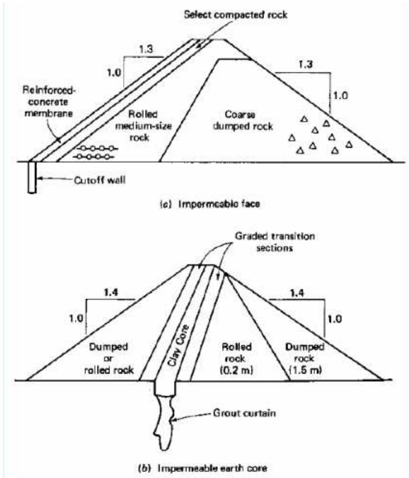


Figure 1. 14 rockfill dam **Figure 1. 15 rockfill dam construction**

Advantages

Rockfill dams have almost the same advantages and disadvantages over gravity dams as discussed for earth dams. Particular advantages and disadvantages over earth dams.

1- Are quite inexpensive if rock fragments are easily available.
2- Can be constructed quite rapidly.
3- Can better withstand the shocks due to earthquake than earth dams.
4- Can be constructed even in adverse climates.

Disadvantages

1- Rockfill dams require more strong foundations than earth dams.
2- Rockfill dams require heavy machines for transporting, dumping and compacting rocks

1.6.1.4 Arch dams

An arch dam is curved in plan, with its convexity towards the upstream side. It transfers the water pressure and other forces mainly to the abutments by arch action. It is quite suitable for narrow canyons with strong flanks which are capable of resisting the thrust produced by the arch action. The section is triangular and is comparatively thinner. It may have a single curvature or double curvature in the vertical plane. These types of dams are subjected to large stresses because of changes in temperature shrinkage of concrete and yielding of abutments.

Figure 1. 16 arch dam

Advantages

1- An arch dam requires less concrete as compared to a gravity dam as the section is thinner.
2- Arch dams are more suited to narrow, V-shaped valley, having very steep slopes.
3- Uplift pressure is not an important factor in the design of an arch dam because the arch dam has less width and the reduction in weight due to uplift does not affect the stability.
4- An arch dam can be constructed on a relatively less strong foundation because a small part of load is transferred to base, whereas in a gravity dam full load is transferred to base.

Disadvantages

1- An arch dam requires good rock in the flanks (abutments) to resist the thrust. If the abutments yield, extra stresses develop which may cause failure.
2- The arch dam requires sophisticated formwork, more skilled labour and richer concrete.
3- The arch dam cannot be constructed in very cold climates because spalling of concrete occurs due to alternate freezing and thawing.
4- The arch dams are more prone to sabotage.
5- The speed of construction is relatively slow.

1.6.1.5 Buttress dams

Buttress dams are of three types: (i) Deck type, (ii) Multiple arch-type, and (iii) Massive-head type.

(i) A deck type buttress dam consists of a sloping deck supported by buttresses. Buttresses are triangular concrete walls which transmit the water pressure from the deck slab to the foundation. Buttresses are compression members. The deck is usually a reinforced concrete slab supported between the buttresses, which are usually equally spaced.

(ii) In a multiple-arch type buttress dam the deck slab is replaced by horizontal arches supported by buttresses. The arches are usually of small span and made of concrete.

(iii) In a massive-head type buttress dam, there is no deck slab. Instead of the deck, the upstream edges of the buttresses are flared to form massive heads which span the distance between the buttresses.

Figure 1. 17 deck type

Figure 1. 18 multiple-arch type

Figure 1. 19 massive-head type

Advantages

1- Uplift/ice pressure is generally not a major factor
2- Can be constructed on relatively weaker foundations.
3- Power house and water treatment plants, etc. can be housed between buttresses.
4- Vertical component of the water pressure on deck prevents the dam against overturning and sliding failures.
5- Can be designed to accommodate moderate movements of oundations without serious damages.
6- Heat dissipation is better in buttress dams.
7- Back of the deck and the foundation between buttresses are accessible for inspection.
8- Can be easily raised subsequently by extending buttresses and deck slabs.

Disadvantages

1- Buttress dams require costlier formwork, reinforcement and more skilled labour. Consequently, the overall cost of construction may be more than that of a gravity dam.
2- Buttress dams are more susceptible to damage and sabotage.
3- Buttress dams cannot be constructed in very cold climates because of spalling of concrete.
4- Because the upstream deck slab is thin, its deterioration may have very serious effect on the stability.

1.6.2 Site Selection for a Dam

A dam is a huge structure requiring a lot of funds. Extreme care shall be taken while selecting the site of a dam. A wrong decision may lead to excessive cost and difficulties in construction and maintenance. Various factors should be considered when selecting the site of a dam.

1- Topography
2- Suitable Foundation
3- Good Site for reservoir – (i) Large storage capacity (ii) Shape of reservoir basin (iii) Water tightness of the reservoir (iv) Good hydrological conditions (v) Deep reservoir (vi) Small submerged area (vii) Low silt inflow (viii) No objectionable minerals
4- Spillway site
5- Availability of materials
6- Accessibility
7- Healthy surroundings
8- Minimum overall cost
9- Other considerations

1.6.3 Selection of Type of Dam

Selection of the most suitable type of dam for a particular site requires a lot of judgment and experience. It is only in exceptional cases that the most suitable type is obvious. Preliminary designs and estimates are usually required for several types of dams before making the final selection on economic basis. The salient features of different types of dams discussed in the preceding sections should be kept in mind while selecting the type of dam. Various factors govern the selection of type of dam.

1- Topography and valley shape
2- Geology and foundation conditions
3- Availability of construction materials
4- Overall cost
5- Spillway size and location
6- Earthquake hazards

7- Climatic conditions.
8- Diversion problems.
9- Environmental considerations
10- Roadway.
11- Length and height of dam.
12- Life of dam.
13- Miscellaneous considerations.

1.7 Hydropower Plant Scheme Layout

Typical components of a hydroelectric plant consist of the following:

1. Structure for water storage and/or diversion, like a dam or a barrage.
2. A head-race water conveying system like a conduit (penstock) or an open channel to transport water from the reservoir or head-water pool up to the turbines.
3. Turbines, coupled to generators.
4. A tail race flow discharging conduit of open channel that conveys the water out of the turbine up to the river.

Although the above components are common for all hydropower development schemes the general arrangement for high and medium head power houses are more or less similar. The low head power plants, which are usually of run-of-power type schemes, have a slightly different arrangement as mentioned in the paragraphs below.

1.7.1 High and medium head development

Usually, there could be two types of power scheme layout:

• Concentrated fall schemes

• Diversion schemes

In the concentrated fall type projects, the powerhouse would be built at the toe of a concrete gravity dam. This type of project development is suitable for medium head projects since a high head project would require an enormous concrete gravity dam, which is generally not adopted. A medium or high head project with an earthfill or rockfill dam may have an isolated or off-stream power house. Here, the water is conveyed to the turbines via penstocks laid under, or by-passing, the dam. Spillways are provided separately to take care of floods. A distinction of such projects is that it consists of a long system of water conduits. Surge tanks are sometimes provided at the end of the conduits to relieve them of water hammer, which is the very high pressure developed by causing the stoppage of flow too suddenly at the turbine end.

In the diversion type of layout, the diversion could be using a canal and a penstock (Figure 18) or a tunnel and a penstock (Figure 19). The former is usually called the Open-Flow Diversion System and the latter Pressure Diversion System.

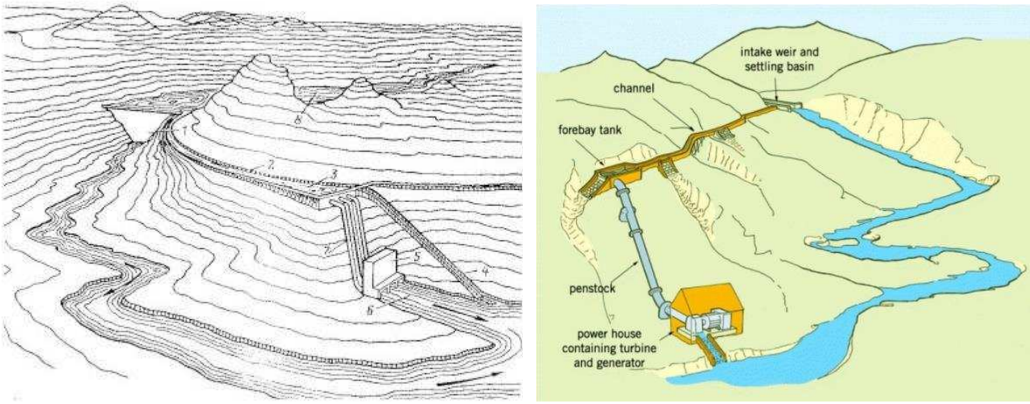


FIGURE 18 HYDROELECTRIC PROJECT BASED ON OPEN FLOW DIVERSION SCHEME
1-DAM .2-INTAKE DIVERSION CONDUIT. 3-HEAD POND. 4- SPILLWAY . 5- POWER HOUSE.
6- TAILVACE. 7-PENSTOCKS. 8-RESERWAIR

Figure 1. 20

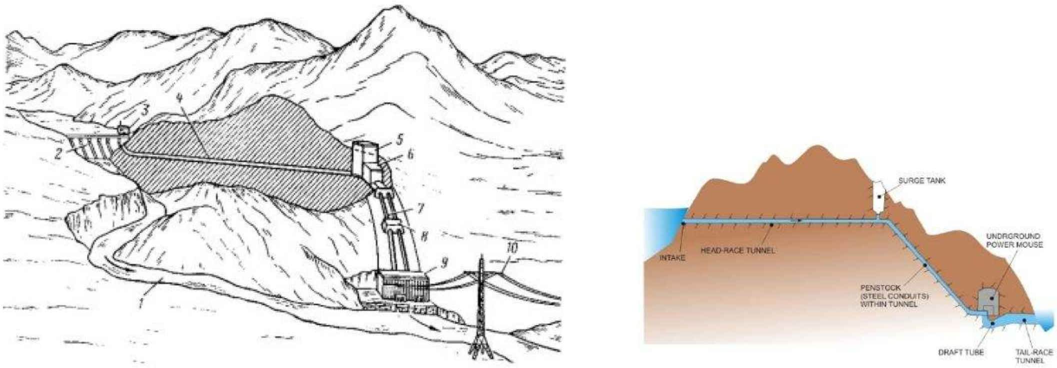


FIGURE 19. HYDROELECTRIC PROJECT USING A PRESSURE DIVERSION SYSTEM
1-WATERCOURSE. 2-DAM. 3- INTAKE STRUCTURES. 4-DIVERSION TUNNEL. 5- SURGE TANK. 6-PENSTOCK FORK HOUSE
7-PENSTOCKS.8- PENSTOCKS SUPPORT.9- POWER HOUSE. 10- POWER LINE

Figure 1. 21

In fact, the combination of open channel and pressure conduit and penstock may be done in a variety of ways shown in Figures 22, 23, and 24.

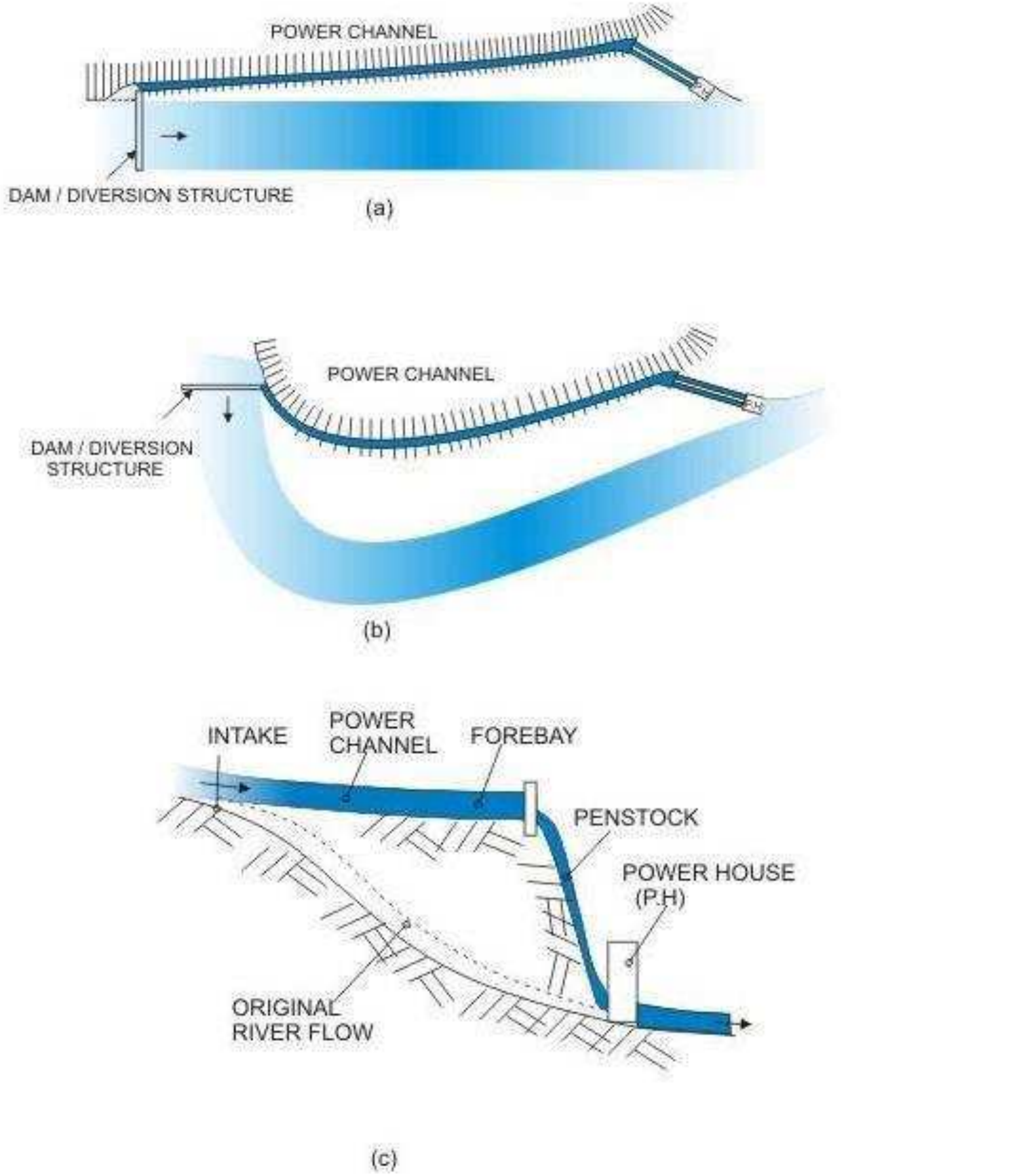


FIGURE 20. DIVERSION HYDRO POWER PROJECT BASED ON OPEN CHANNEL AND PRESSURE FLOW SYSTEMS
(a) LONG CANAL AND SHORT SURFACE PENSTOCK ALONG STRAIGHT RIVER REACH
(b) SAME AS (a) BUT IN A CURVED RIVER REACH
(c) SECTIONAL VIEW ALONG WATER CONDUCTING SYSTEM FOR (a) AND (b)

Figure 1. 22

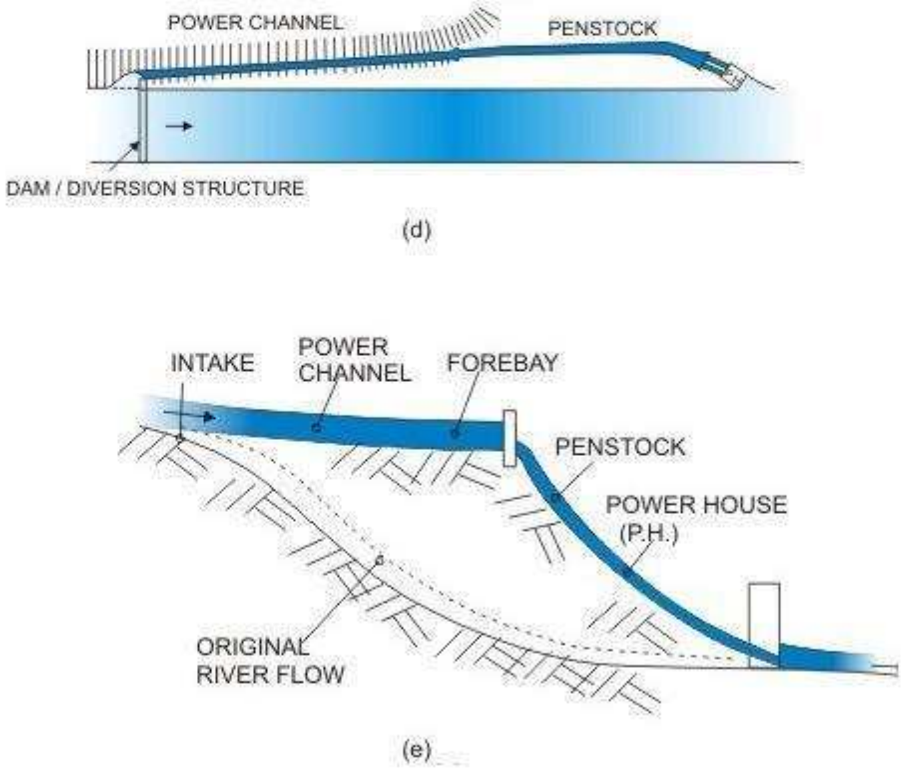


FIGURE 20. DIVERSION HYDRO POWER PROJECT BASED ON OPEN CHANNEL AND PRESSURE FLOW SYSTEMS
(d) SHORT CANAL AND LONG SURFACE PENSTOCK
(e) SECTIONAL VIEW FOR (d)

Figure 1. 23

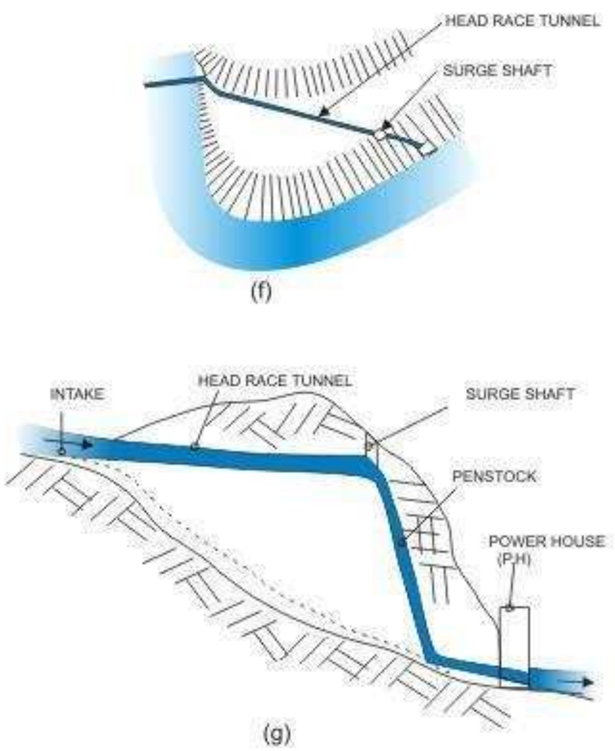


FIGURE 20. DIVERSION HYDRO POWER PROJECT BASED ON OPEN CHANNEL AND PRESSURE FLOW SYSTEMS
(f) HEAD RACE TUNNEL AND PENSTOCK IN A CURVED RIVER REACH
(g) SECTIONAL VIEW FOR (f)

Figure 1. 24

There could be totally underground projects which consist of only pressure system of water conveyance. A variety of such projects are shown in Figure 25. This type of project layout may be termed as underground diversion schemes where even the power house is built underground.

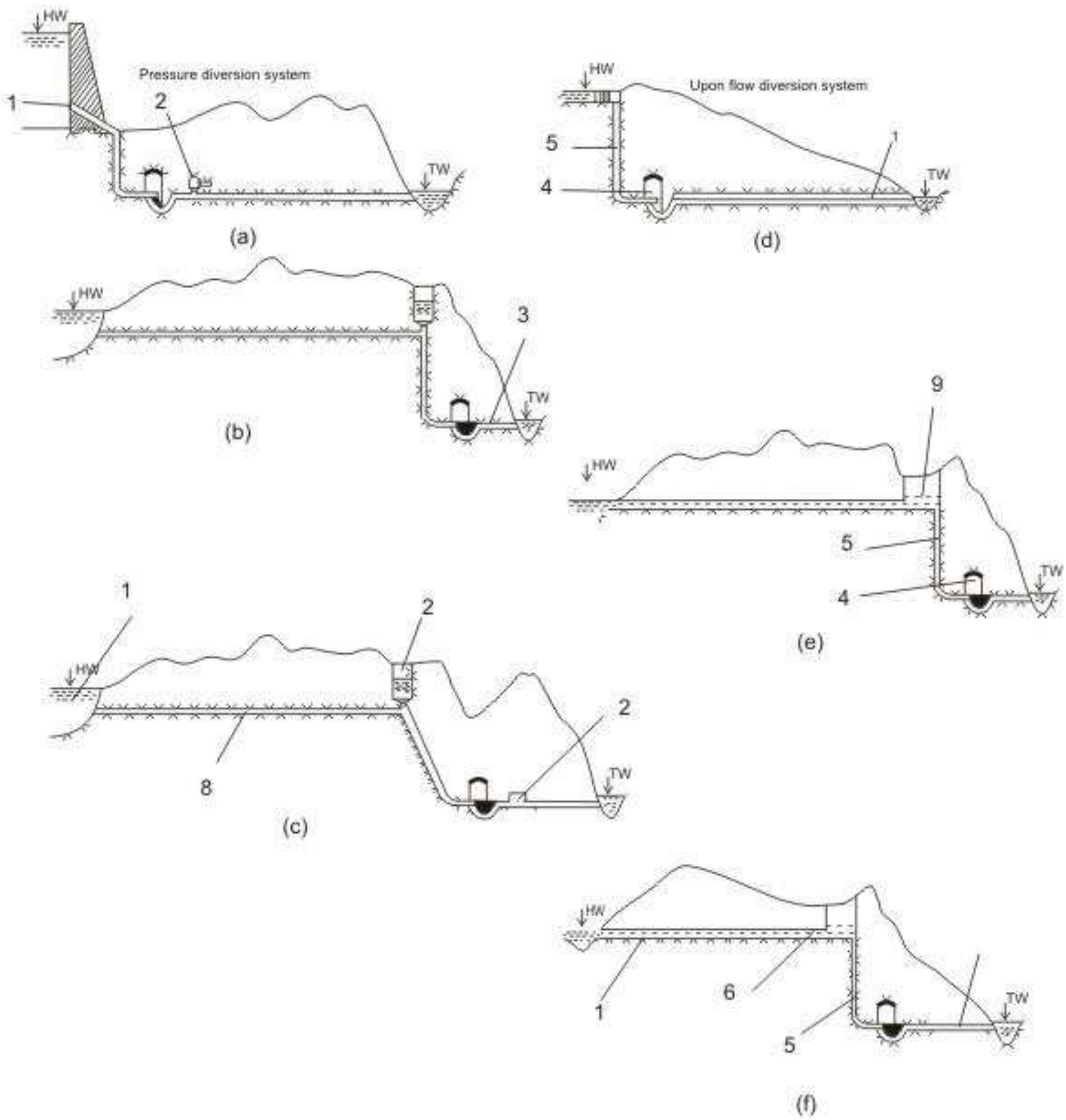


FIGURE 21. Underground project with
(a),(b) and (c) pressure diversion system, and (d),(e) and (f) open flow diversion system;
1-intake structure; 2- surge tank; 3-tailrace pressure tunnel; 4- power house; 5-penstock
6-intake open flow tunnel; 7-tailrace open flow tunnel ;8-intake pressure tunnel; 9--head pond

Figure 1. 25

1.7.2 Low head development

Here too, two types of layouts may be possible:

• In-stream scheme

• Diversion scheme

In the in-stream type of project, the powerhouse would be built as a part of the diversion structure, as shown in Figure 2(a) or a general detailed view in Figure 6. A typical layout of such a power house and its cross section is shown in Figure 22.

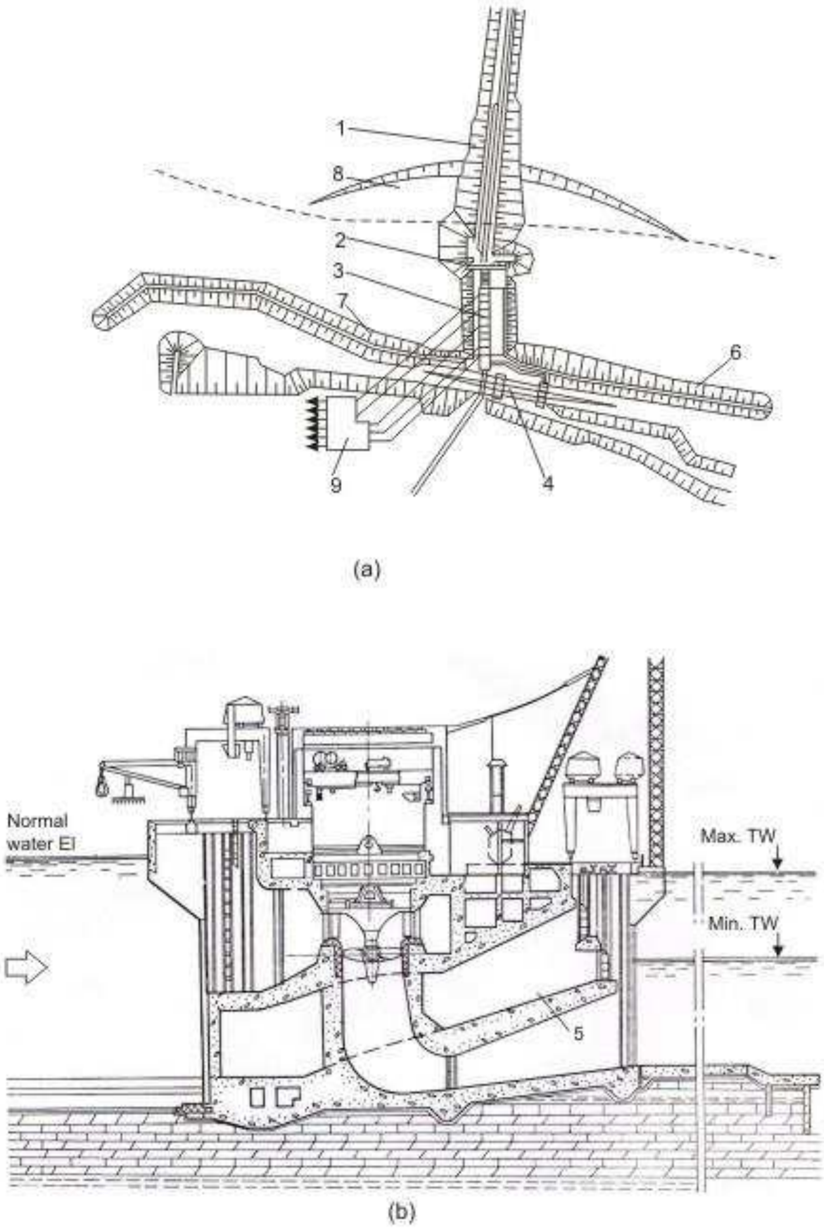


FIGURE 22. A TYPICAL LOW-HEAD IN STREAM POWER HOUSE
(a) plane ;(b) sectional elevation of the power house; 1-earth dam.2-over flow dam
3- power house; 4-lock;5-spillways in power house; 6- navigable canal dike downstream of dam;
7-output dike; 8- left bank clearing; 9- electrical switch yard

Figure 1. 26

In the diversion type of scheme, there has to be a diversion structure as well as a diversion canal, as shown in Figure 2(b). The power house may be located at some convenient point of the canal, that is, at its upstream end, middle, or at the downstream end. The location of the power house depends upon conditions such as hydrological, topographical, geological, environmental economic conditions. The ground water table has to be taken into account.

1.7.3 Position of power houses

As might have been noticed from the layouts, there could be a variety of position for the power house with respect to natural ground level.

IS: 4410(Part10)-1988 differentiates between the following types of power stations, which may be constructed as per site conditions:

1. Surface power station or over ground power station: A power station which is constructed over the ground with necessary open excavation for foundations. Typical examples may be seen from Figs. 7, 11 or 12.

2. Underground power station: A power station located in a cavity in the ground with no part of the structure exposed to outside. A typical example of this type is shown in Figure 23.

3. Semi-underground power station: A power station located partly below the ground level and followed by a tail race.

Figure 1. 27 surface power station

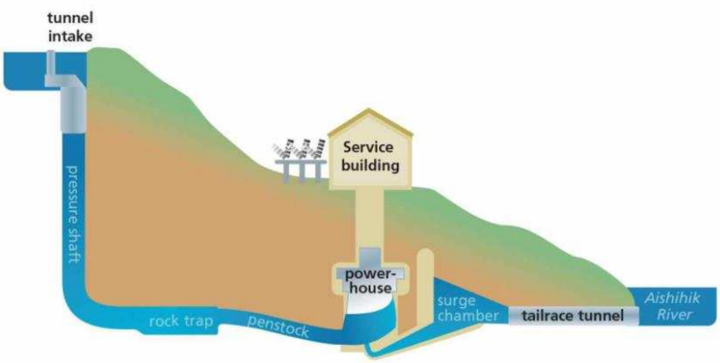


Figure 1. 28 underground power station

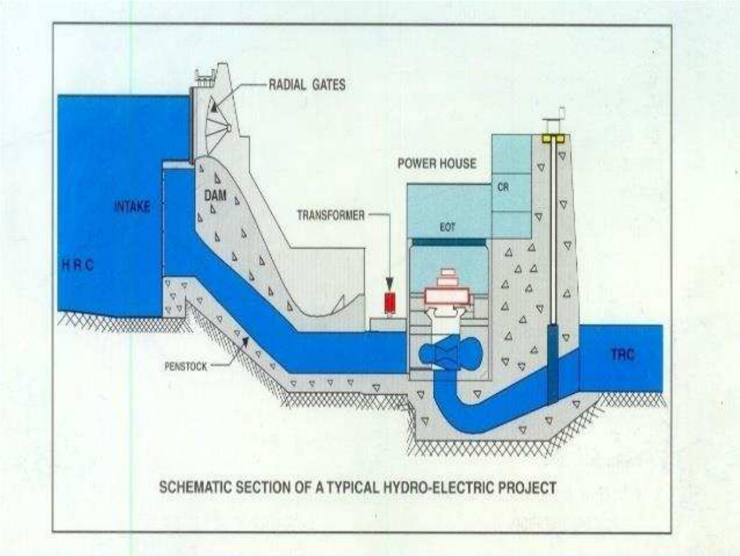


Figure 1. 29 semi-underground power station

1.8 Electrical terms associated with hydropower engineering

Electrical power generated or consumed by any source is usually measured in units of Kilowatt-hour (kWh). The power generated by hydropower plants are normally connected to the national power grid from which the various withdrawals are made at different places, for different purposes. The national power grid also obtains power generated by the non-hydropower generating units like thermal, nuclear, etc. The power consumed at various points from the grid is usually termed as electrical load expressed in Kilo-Watt (KW) or Mega-Watt (MW). The load of a city varies throughout the day and at certain time reaches the highest value (usually in the evening for most Indian cities), called the Peak load or Peak demand. The load for a day at a point of the national power grid may be plotted with time to represent what is known as Daily Load Curve. Some other terms associated with hydropower engineering are as follows:

Load factor

This is the ratio of average load over a certain time period and the maximum load during that time. The period of time could be a day, a week, a month or a year. For example, the daily load factor is the ratio of the average load may be obtained by calculating the total energy consumed during 24 hours (finding the area below the load vs. time graph) and then divided by 24. Load factor is usually expressed as a percentage

$$\textbf{Load factor/day} = \frac{\textbf{\textit{total energy consumed during a day}}\ (\textbf{MWhr})}{\textbf{\textit{maximum load}} \times \textbf{24 hrs}}$$

$$\textbf{Load factor/year} = \frac{\textbf{\textit{total energy consumed during a year}}\ (\textbf{MWhr/year})}{\textbf{\textit{maximum load}} \times \textbf{8,760 hrs}}$$

Installed capacity

For a hydro electric plant, this is the total capacity of all the generating units installed in the power station. However all the units may not run together for all the time.

Capacity factor

This is the ratio of the average output of the hydroelectric plant for a given period of time to the plant installed capacity. The average output of a plant may be obtained for any time period, like a day, a week, a month or a year. The daily average output may be obtained by calculating the total energy produced during 24hours divided by 24. For a hydroelectric plant, the capacity factor normally varies between 0.25 and 0.75.

$$\textbf{capacity factor} = \frac{\textbf{generation (MWhr/year)}}{\textbf{installed capacity (MW)} \times \textbf{8,760 hrs}}$$

Utilization factor

Throughout the day or any given time period, a hydroelectric plant power production goes on varying, depending upon the demand in the power grid and the power necessary to be produced to balance it. The maximum production during the time divided by the installed capacity gives the utilization factor for the plant during that time. The value of utilization factor usually varies from 0.4 to 0.9 for a hydroelectric plant depending upon the plant installed capacity, load factor and storage.

$$\textbf{Utilization factor} = \frac{\textbf{maximum generation (MW)}}{\textbf{installed capacity (MW)}}$$

Firm (primary) power

This is the amount of power that is the minimum produced by a hydro-power plant during a certain period of time. It depends upon whether storage is available or not for the plant since a plant without storage like run-of-river plants would produce power as per the minimum stream flow. For a plant with storage, the minimum power produced is likely to be more since some of the stored water would also be used for power generation when there is low flow in the river.

Secondary Power

This is the power produced by a hydropower plant over and above the firm power.

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CHAPTER 2

CHAPTER 2

PELTON WHEEL

2.1 INTRODUCTION

We emphasize large dynamic turbines that are designed to produce electricity. Most of our discussion concerns hydroturbines that utilize the large elevation change across a dam to generate electricity. There are two basic types of dynamic turbine— impulse and reaction, each of which are discussed in some detail. Comparing the two power-producing dynamic turbines, impulse turbines require a higher head, but can operate with a smaller volume flow rate. Reaction turbines can operate with much less head, but require a higher volume flow rate.

Impulse turbine:

Pelton wheel

Reaction turbine:

1- Francis turbine

2- Kaplan turbine

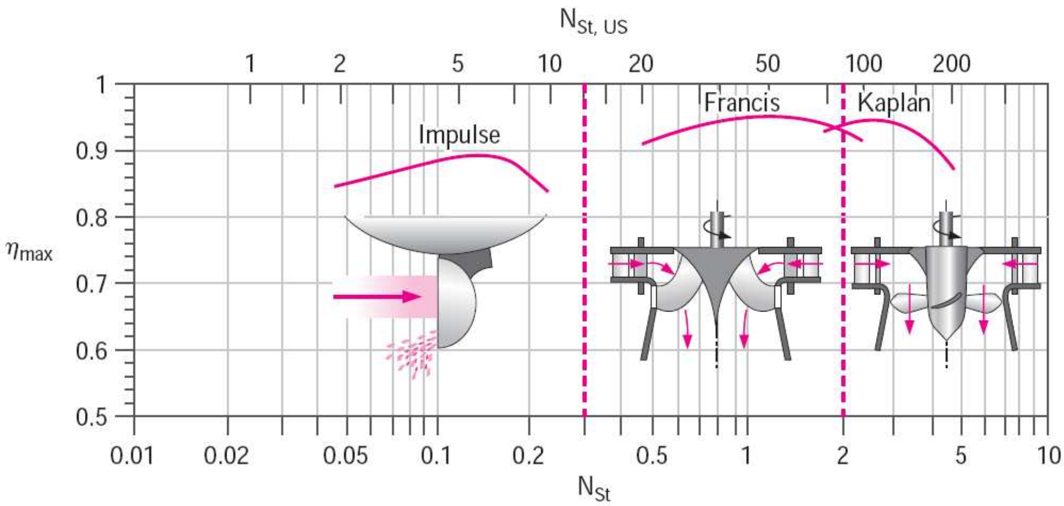


Figure 2. 1 Maximum efficiency as a function of turbine specific speed for the three main types of dynamic turbine.

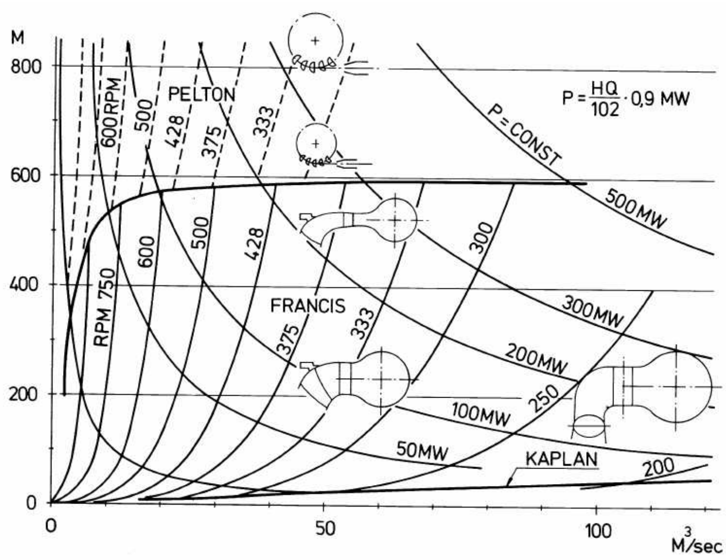


Figure 2. 2 When to use a Pelton turbine

2.2 PELTON WHEEL

In the impulse turbine all the energy of the water is converted into velocity before entering the wheel by expanding through a nozzle or guide vanes .The pressure of the water is atmospheric, hence the wheel must not run full; in which case, it must be placed at the foot of the fall and above the tailrace. The water may be admitted over part of the circumference only or over the whole circumference. An example of impulse turbine is Pelton wheel, Turgo wheel and Banki turbine (cross flow or Ossberger turbine)

Pelton wheel is a special type of axial-flow impulse turbine and is used for very high heads. It is the most efficient type of impulse wheel, having an overall efficiency of 88%.

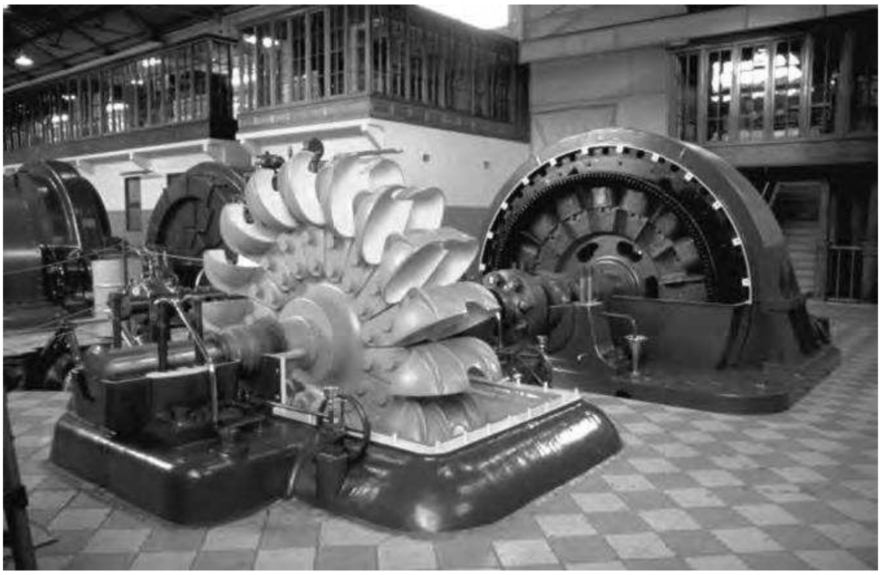


Figure 2. 3 A close-up view of a Pelton wheel showing the detailed design of the buckets; the electrical generator is on the right.

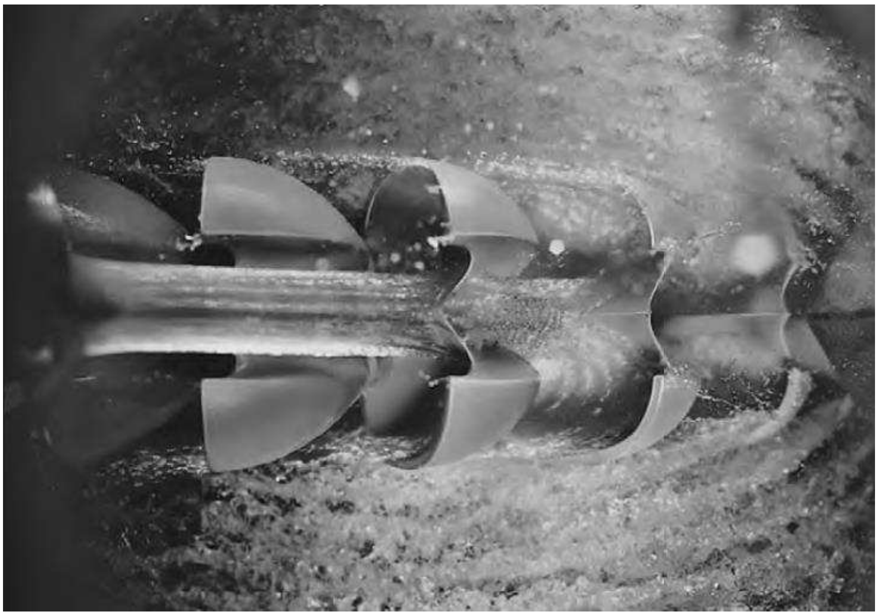


Figure 2. 4 A view from the bottom of an operating Pelton wheel illustrating the splitting and turning of the water jet in the bucket

In such a process, the entire available energy of the water is converted into kinetic energy by passing it through nozzles. The jet impinges on the wheel from one or more nozzles and strikes the blade at the center flowing axially in both directions. The blades are known as buckets and consist of a double hemispherical cup, Fig. (2.5.), fitted with dividing wall (splitter), Plate1. As the water flows axially in both directions, there is no axial thrust on the wheel. Each jet may be deflected backward through an angle of about 165^o as shown in Fig. (2.6). A complete reversal of 180^o would be desirable. This is not possible because the fluid must be thrown to one side to clear the following bucket. The bucket changes the velocity direction of the water, thus a dynamic force is developed.

It should be noted that the flow only partly fills the buckets, and the fluid remains in contact with the atmosphere. Thus, once the jet has produced by the nozzle, the static pressure of the fluid is atmospheric throughout the machine. Figure (2.4) shows a section through a bucket, which is being acted on by a jet. The plane of section is parallel to the axis of the wheel and contains the axis of the jet, Plates 3 a and b.

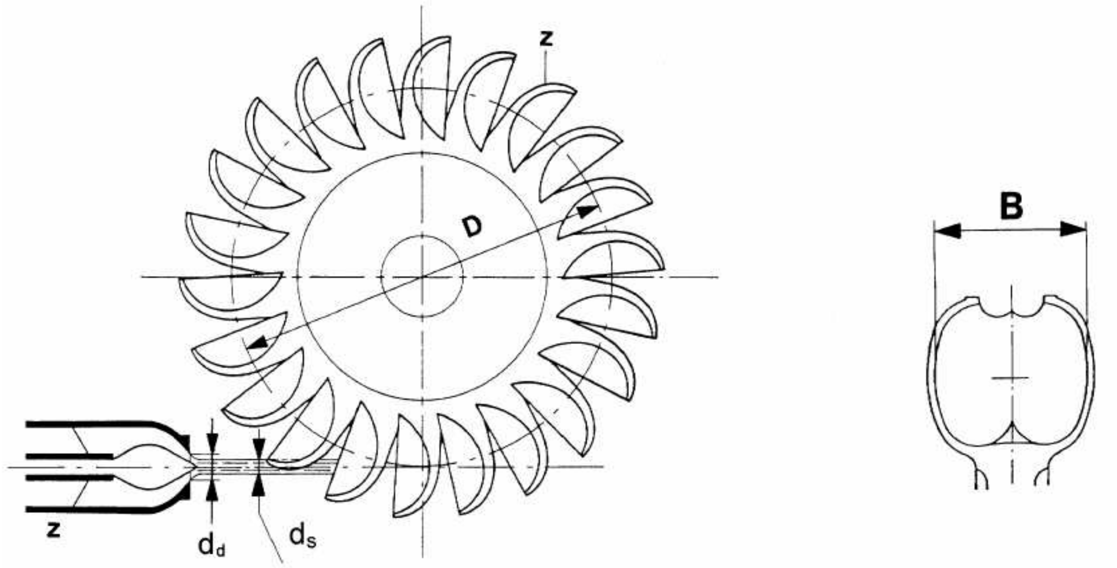


Figure 2. 5 Main dimensions for the Pelton runner

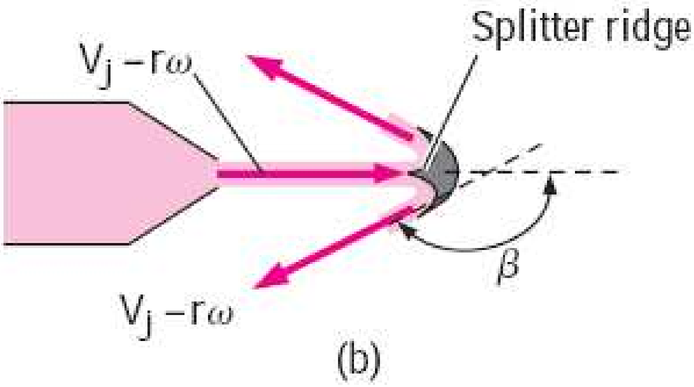


Figure 2. 6 Schematic diagram of a cross section of bucket

2.2.1 Velocity triangles.

The work done may be obtained, as previously explained, by *Euler's* equation for turbine. The total head available at the nozzle is equal to the gross head loss the head losses due to friction in the pipeline leading to the nozzle. If it is equal to H, then the velocity of jet issuing from the nozzle is

$$C_1 = C_V (2g * H)$$

Where C_v is the velocity coefficient and its value is between 0.97 and 0.99. The velocity head of the fluid in the pipeline is normally negligible compared with H.

During the time that any one-bucket is being acted on by the jet the wheel turns through a few degrees and so the direction of the motion of the bucket changes slightly. The effect of this change, however, is very small, and it is sufficient here to regard the direction of the bucket velocity as the same as that of C_1. Since the radius of the jet is small compared with that of the wheel, all the fluid may be assumed to strike the bucket at radius r. It is also assumed that all the fluid leaves the bucket at radius r and that the velocity of the fluid is steady and uniform over sections 1 and 2 where the values C_1 and C_2 are considered.

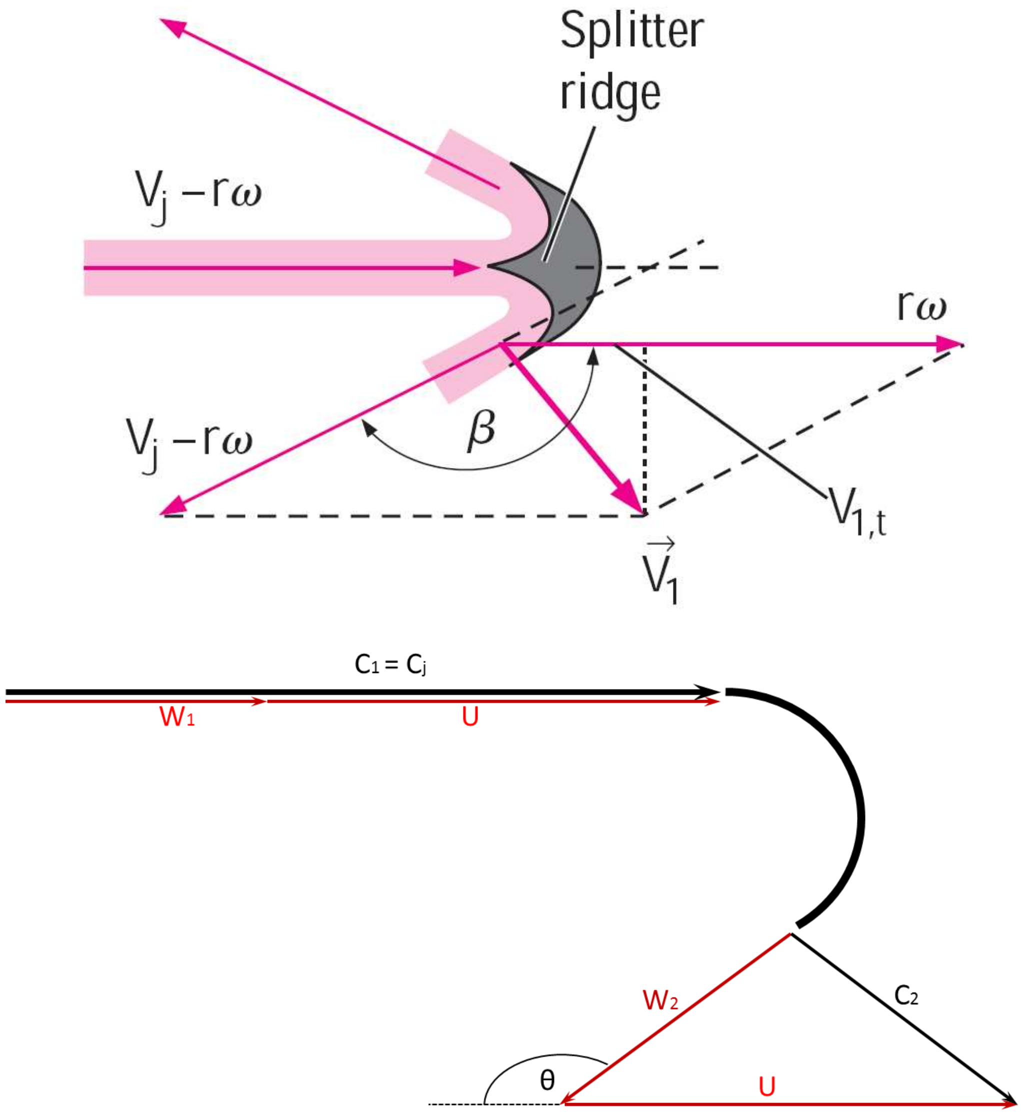


Figure 2. 7 velocity triangles

Since C_1 and U are co-linear, the diagram of velocity vectors at inlet is simply a straight line Fig. (2.7)

$$U_1 = W_{r1}$$

$$C_{U1} = C_1$$

$$W_1 = C_1 - U$$

The relative velocity W_2 with which the fluid leaves the bucket is somewhat less than the initial relative velocity W_1. This is due to the frictional losses as the fluid flows over the buckets and to additional loss as the fluid strikes the splitter ridge, because the ridge cannot have zero thickness.

$$W_2 = K\, W_1$$

Where, K is a fraction slightly less than unity.

2.2.2 Euler's head

As the bucket is symmetrical it is sufficient to consider only that part of the flow, which traverses one side of it. To obtain the velocity triangle at outlet, considering the direction of U as positive.

$$C_{U2} = U - W_2 Cos(\pi - \theta)$$

$$C_{U2} = U + W_2 Cos(\theta)$$

$$\Delta C_U = C_{U1} - C_{U2} = C_1 - \{U + W_2 Cos(\theta)\}$$

$$\Delta C_U = C_1 - \{U + W_2 Cos(\theta)\} = W_1 - W_2 Cos(\theta)$$

$$\Delta C_U = W_1 [1 - K Cos(\theta)]$$

$$H_E = \frac{U_1 C_{U1} - U_2 C_{U2}}{g}$$

$$H_E = \frac{U(C_{U1} - C_{U2})}{g} = \frac{\Delta C_U}{g}$$

So that,

$$H_E = \frac{U}{g}(C_1 - U)[1 - K Cos(\theta)]$$

This equation shows that there is no energy transfer when the blade velocity is either zero or equal to the jet velocity.

2.2.3 Maximum Euler's head

The maximum energy transfer will occur at some intermediate value of the blade velocity.

This is obtained by differentiation with respect to U as follows:

$$\frac{dH_E}{dU} = \frac{(C_1 - 2U)[1 - K Cos(\theta)]}{g} = 0$$

Hence,

$$C_1 - 2U = 0$$

Therefore,

$$U = \frac{1}{2} C_1$$

$$\text{or } \theta = 180^0$$

Substituting this value in equation (4-3) yields

$$H_{E\max} = \frac{C_1^{\ 2}}{4g}[1 - KCos(\theta)]$$

Now the energy input from the nozzle per unit weight is $\frac{C_1^{\ 2}}{2g}$

Thus the maximum efficiency of Pelton Wheel becomes

$$\eta_{\max} = \frac{[1 - KCos(\theta)]}{2}$$

In the ideal case, assuming no friction, K = 1. Also if θ = 180^o, the maximum efficiency becomes 100 percent. In practice however, friction exists and K is in the range of 0.8 to 0.85. Also, the blade angle is usually 165^o, to avoid the interference between the oncoming and out coming jets. Thus the ratio of the wheel velocity to the jet velocity (speed factor) becomes, in practice somewhat smaller than the theoretical, this ratio is about 0.46. Fig. (2.8)

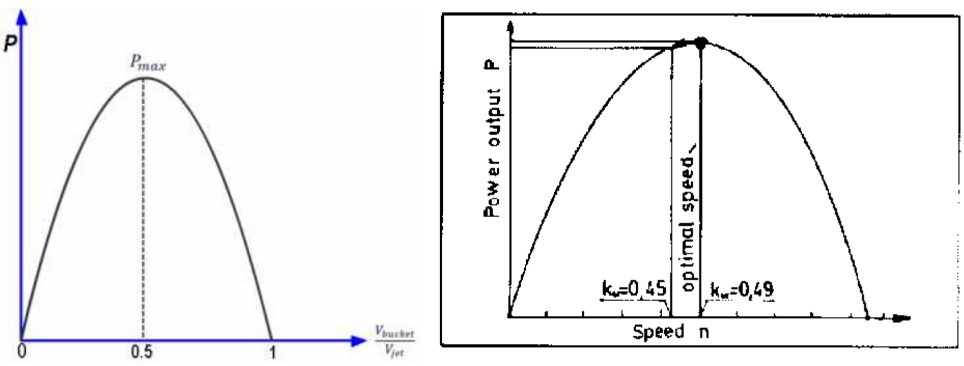


Figure 2. 8 Maximum Euler's Head

2.2.4 Efficiency of Pelton wheel:

The overall efficiency quoted for a machine always refers to the ability of the machine itself to convert fluid energy to useful mechanical energy.

$$\eta_{o_{turbine}} = \frac{P_{sh}}{P_{jet}} = \frac{P_{sh}}{\gamma Q_{th} H_{net}}$$

Thus the required power output P determines the volume rate of flow Q for a certain head H. Since C_1 is already determined, the total cross section area of the jets is given by Q/C_1.

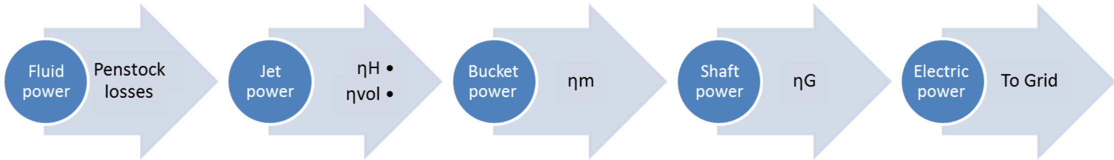


Figure 2. 9 Efficiencies of Pelton wheel

$$\eta_H = \frac{H_E}{H_{net}} \qquad \eta_{vol} = \frac{Q_{act}}{Q_{th}}$$

$$\eta_m = \frac{P_{shaft}}{\gamma Q_{act} H_E}$$

$$\eta_{o_{turbine}} = \frac{P_{sh}}{P_{jet}} = \frac{P_{sh}}{\gamma Q_{th} H_{net}}$$

$$\eta_{o_{plant}} = \frac{P_{elec}}{P_{jet}} = \frac{P_{elec}}{\gamma Q_{th} H_{net}}$$

2.2.4.1 Efficient Operation:

Although the Pelton wheel is efficient and reliable when operating under large heads, it is less suited for smaller heads. This can be explained by that the wheel operates in atmospheric air although housing encloses it. It is therefore essential that the wheel be placed above the tail water race. The head from the nozzle to tail water is wasted. To develop a given output power under a smaller head the rate of flow would need to be greater, with a consequent increase in jet diameter and the increase of wheel diameter. Since, moreover the jet and bucket velocities are reduced as the head is reduced; the machine becomes very bulky and slow running.

A greater rate of flow can be achieved by the use of more jets Fig (2.10).

2.2.5 Design of pelton wheel:

In the design of a Pelton wheel two parameters are of particular importance

a- The ratio of the bucket width to the jet diameter, and

b- The ratio of the wheel diameter to the jet diameter.

2.2.5.1 Design of Buckets

If the bucket width is too small in relation to the jet diameter, the buckets do not smoothly deflect the fluid and in consequence, much energy is dissipated in turbulence and the efficiency drops considerably. On the other hand, if the buckets are too large, the friction on surfaces is unnecessarily high.

The optimum velocity of the bucket width /jet diameter has been found to be 4-5.

The depth of the bucket / jet diameter is 0.9 - 1.2.

2.2.5.2 Number of Buckets

Approximate number of buckets is generally found by using following empirical relations

$$\text{No. of Buckets} \quad Z = 0.5\left(\frac{D}{d}\right) + 15 \quad \text{or} \quad Z = 5.4\sqrt{\frac{D}{d}}$$

Where, D/d is the ratio of wheel diameter to jet diameter.

Small values of D/d involve either too close a spacing of the buckets or two few buckets for the whole jet to be used. There is no upper limit to the ratio, but the larger its value the more bulky is the entire installation, Fig (4.5). In practice a minimum value of about 10 is usually chosen for the ratio of wheel diameter to jet diameter.

2.2.6 Specific Speed and Wheel-jet diameter

The specific speed of a turbine is given by Equation:

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

Results of tests made on single nozzle Pelton turbines show a limited range of specific speed for high efficiency. Generally N_s = 9 - 31 is recommended. The highest efficiency attained is about 88% at approximately Ns = 20. The relationship between Jet ant wheel pitch diameters D affects the specific speed hence the efficiency of the turbine. This may be shown by substituting in Equation 3.9 the following expressions:

i) $P = \frac{\gamma * Q * H * \eta_o}{75} = \frac{\gamma * \pi * d_j^{\,2} * C_1^{\,3} * \eta_o}{4*75*2*g}$

ii) $N = \frac{60 * U}{\pi * D}$

iii) $C_1 = C_V \sqrt{2 * g * H}$

Hence,

$$N_S = \frac{60 * (2g)^{3/4} \sqrt{\gamma}}{2\sqrt{\pi * 75}} \sqrt{\eta_o} C_v^{\,3/2} (\frac{U}{C_1}) \frac{d_j}{D}$$

With standard numerical value of γ,g and C_V

$$N_S \cong 575 \sqrt{\eta_o} (\frac{U}{C_1}) \frac{d_j}{D}$$

Substituting into Equation $\eta_o = 0.88$ and $(\frac{U}{C_1}) = 0.45$ and $N_S = 20$, one obtains the optimum wheel diameter ratio $\frac{D}{d_j} \cong 11$ (approximately)

2.2.7 Number of jets in a Pelton turbine:

Generally, it has a single jet. But if a **s**ingle jet cannot develop the required power and to increase the specific speed, *2* or 4 Jets can be employed, with maximum number 6.

The jets must be equidistant on the outer periphery of the wheel, In this case,

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

Where *n* is the number of jets and HP is the total power of the turbine unit. Sometimes, instead of providing a number of jets to a wheel, two or three wheels are mounted on a common shaft.

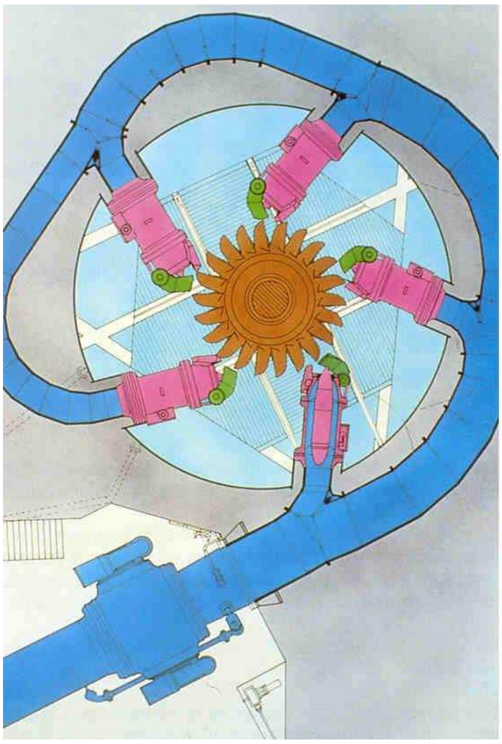


Figure 2. 10 multi-jets Pelton turbine

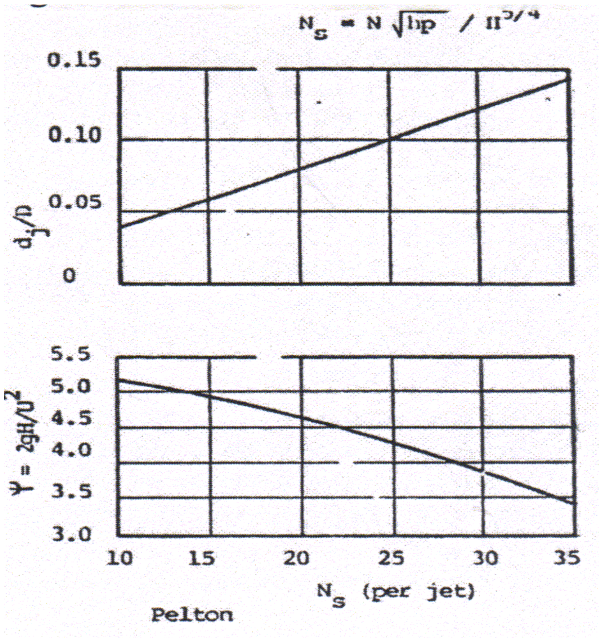


Figure 2. 11 relation between specific speed and jet diameter

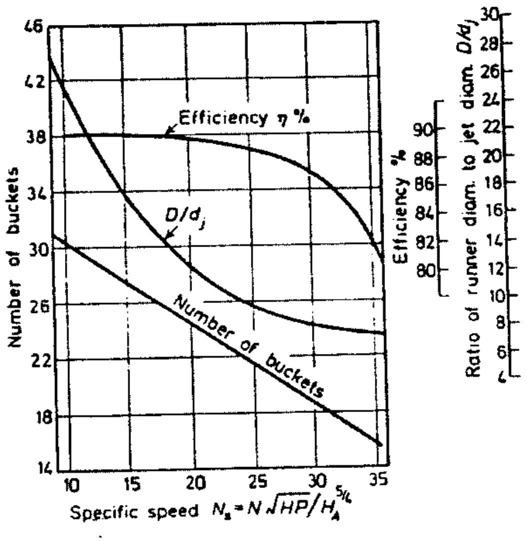


Figure 2. 12 relation between specific speed and number of buckets

2.3 GOVERNING OF PELTON WHEEL:

Since a Pelton wheel is usually employed to drive an electrical generator, it is required that its speed of rotation is constant, regardless of the load. Thus, it must be constant, but for maximum efficiency it is also important that the speed ratio is maintained constant as well.

Since the jet velocity depends only upon the total heads H, which for a given installation is also constant, the velocity ratio may be kept constant provided there is no reduction of head at the nozzle. It follows that any alteration of the load on the turbine must be accompanied by a corresponding alteration of the waterpower but with (U/C_1) remaining constant. Since P = (γ g Q H), it follows that this requirement can only be achieved by alteration in Q such that H is unchanged.

$$Q = AC_1 = AC_V\sqrt{2gH}$$

And therefore, to vary Q, the area of the jet must be changed. This is achieved by any of the following methods operated by the servomotor mechanism:

2.3.1 Spear Regulation:

It consists of a nozzle in which a spear moves too and for by the action of the *servomotor piston* and controls the quantity of water at changing demands, Plate3. This movement causes variation of cross sectional area of jet without change of velocity as shown in Fig. (2.13).

Application:

This method is useful, when the fluctuations in load are small.

Drawback:

When the load falls suddenly, sudden closure of the nozzle causes water hammer in the penstock. Therefore simple spear regulation system is not used in modern turbines where fluctuations in the load ore sudden.

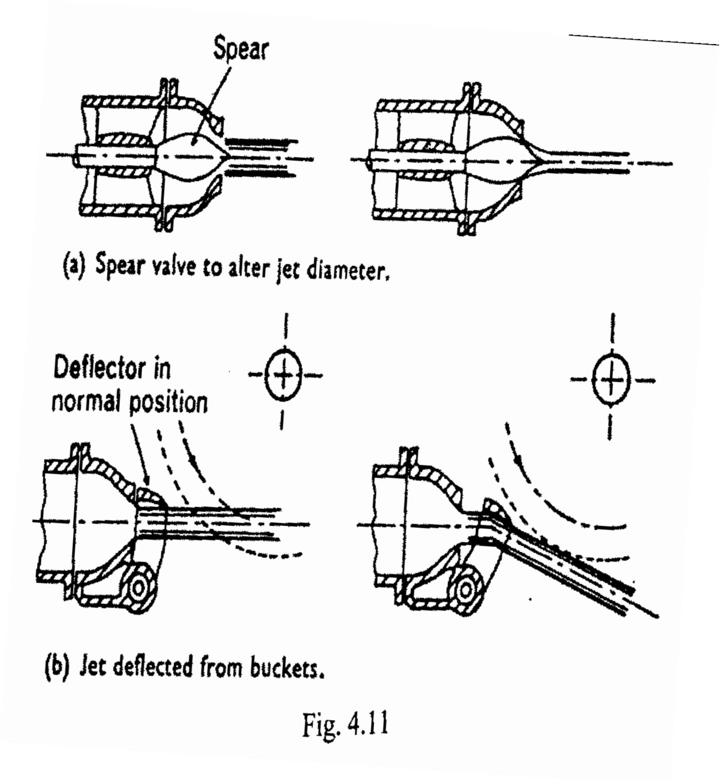


Figure 2. 13 governing of Pelton wheel

2.3.2 Deflector Regulation

It is a plate, pivoted just outside the nozzle and is connected to the oil pressure governor through levers. When the load, drops, it deflects a part of the jet and thus controls the quantity of water striking the buckets.

Drawbacks:

In this system large amount of water goes into waste but there are no chances of water hemmer in penstock. This system is also not mostly used due to large wastage of water and poor speed regulation.

2.3.3 Double Regulation

Figure (2.14) shows this method and is used by all modern Pelton wheel because it has advantages of both spear as well as deflector regulation Systems. If load on a turbine drops suddenly, the speed tends to rise suddenly. In this, system operates the deflector, which comes into action immediately and obstructs a part of water reaching to the bucket. In the mean time, spear is gradually forward to its new position (due to the servomotor) and thus the risk of water hammer is avoided. When the spear has moved to its *new* position to allow the required quantity of water, the deflector moves backward and allows the full jet to strike the buckets through spear nozzle.

As this type of governing controls the speed of turbine and pressure (i.e. water hammer) in the penstock by the combination of spear and jet deflector, this system is known as “Double Regulation”, because it incorporates the use of both the first types of regulation.

Normal Load Running:

The piston in the distribution valve and the actuator occupies their normal positions as shown In Fig. (4.12) In this position as the ports **A** and **B** are closed the oil supplied by the gear pump remains in the middle portion of the distribution va1ve and thus gears of the pump go on rotating without developing further pressure.

Decreasing Load:

As the load on the turbine decreases, speed increases corresponding. As the actuator is connected to the turbine shaft, its speed also increases and balls fly off. Therefore sleeve moves upward, the lever rod inclines and the bell crank lever moves downward. When the bell crank Lever moves downward, the jet deflector will operate and divert whole or part of the jet sway from the buckets. With the downward movement of bell crank lever, the roller on the cam rises.

Distribution Valve Working:

When the lever rod inclines then sleeve control valve rod is pushed down in the distribution valve. Downward movement opens the port B end keeps port k still closed. As soon as port **B** opens, the oil from the middle port of distribution valve goes to the servomotor on the left side of the piston

This oil of high pressure forces the piston to move towards right side, thus forcing the spear to reduce the area of jet and hence the rate of flow. The spear occupies its final position; jet deflector occupies its original position and does not obstruct jet as the speed again becomes normal.

Increasing Load:

When the load on the turbine increases, the speed reduces. This causes the fly balls of the actuator to come down and thus sleeve also moves downward, causing the lever rod to move upward on the other side of the fulcrum. This causes the control valve rod to move upward and to open port **A** but to close port **B**. Thus the oil under pressure will go into the servomotor via port A to the right side of the piston. The piston moves towards left, causing the spear to move away from the nozzle and increases area of flow, which in turn increases the amount of water striking the buckets and. thus brings the speed to come normal. In this case Jet deflector move up but deflector plate is not able to obstruct the jet at normal speed.

2.3.4 Auxiliary nozzle

The auxiliary nozzle arranged at the back of the blades as in Fig. (2.15) is provided for emergency unloading of the wheel.

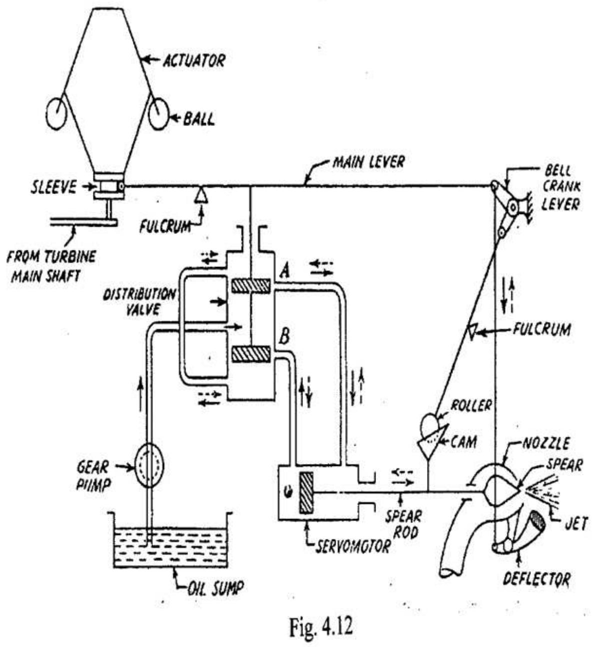


Figure 2. 14 governing of Pelton wheel

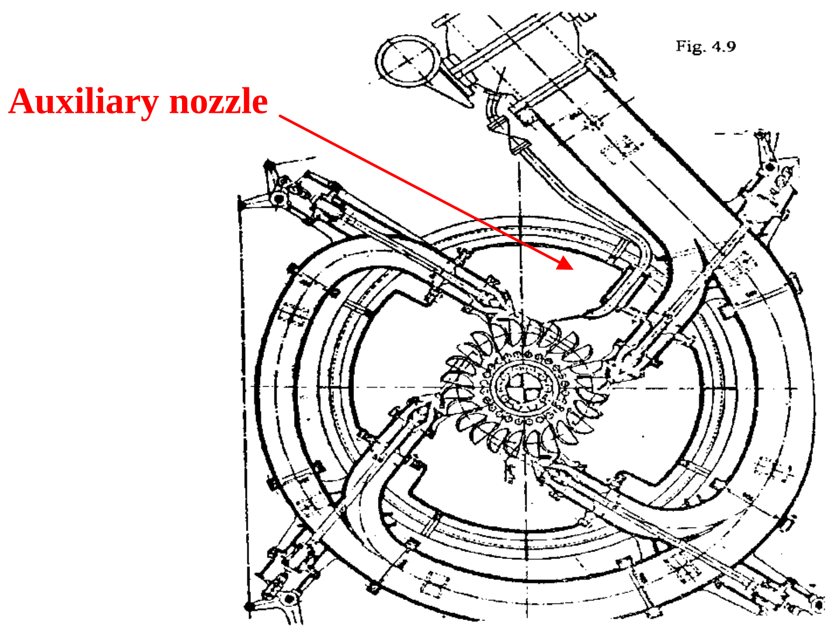


Figure 2. 15 the auxiliary nozzle

2.4 OTHER TYPES OF IMPULSE TURBINES

2.4.1 Turgo Turbine

can handle relatively larger quantities of flow at a given speed and runner diameter by passing the jet obliquely through the runner in a manner similar to a steam turbine. The jet impinges on several buckets continuously whereas only a single bucket per jet is effective at any instant in a Pelton wheel.

As for Pelton Wheel, the Turgo turbine is controlled by the needle valve. Since the needle valve can throttle the flow while maintaining essentia1ly constant jet velocity, the relative velocities at entrance and exit remain unchanged, producing nearly constant efficiency over a wide range *of* power output. Fig. (2.16).

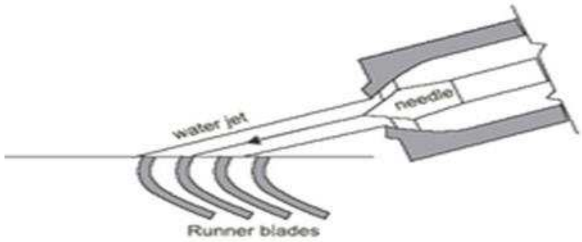


Figure 2. 16 Turgo Turbine

2.4.2 Banki-Mitchell Turbine

Cross flow turbine, illustrated in Fig. (2.17), is a variation on this theme. The water flows from a water jet of rectangular cross-section and passes twice through the turbine blading at right angles to the shaft. The water flows through the runner first from the periphery towards the center end then, after crossing the open center space, from inside outwards. This machine is therefore a turbine with two velocity stages, the water filling only part of the runner at any time

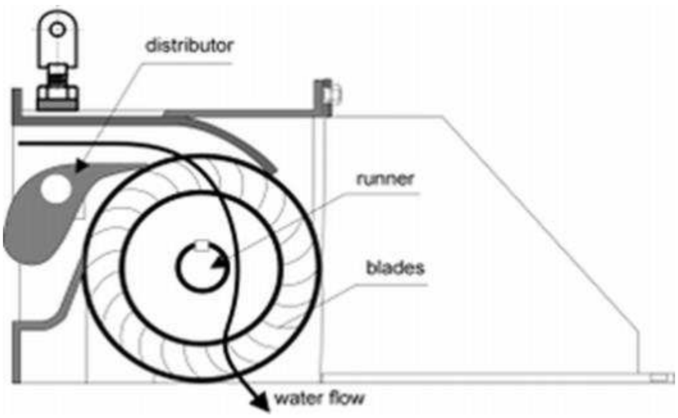


Figure 2. 17 Banki-Mitchell Turbine

Usually the cross-flow turbine is controlled by a guide vane, which can be divided into two separately controlled sections. For most installations the lengths of the two guide vane sections are in the ratio (1/2), allowing for utilization of (1/3), (2/3), or the entire runner, depending on the flow conditions. This contribution provides a relatively flat efficiency curve over the power range of 15 to 100 percent. The long flat peak and the low no load flow are especially important for small power stations, where frequently only a single turbine is installed end the water flow varies between wide limits. Efficiency of a cross flow turbine is shown in Fig. (2.18).

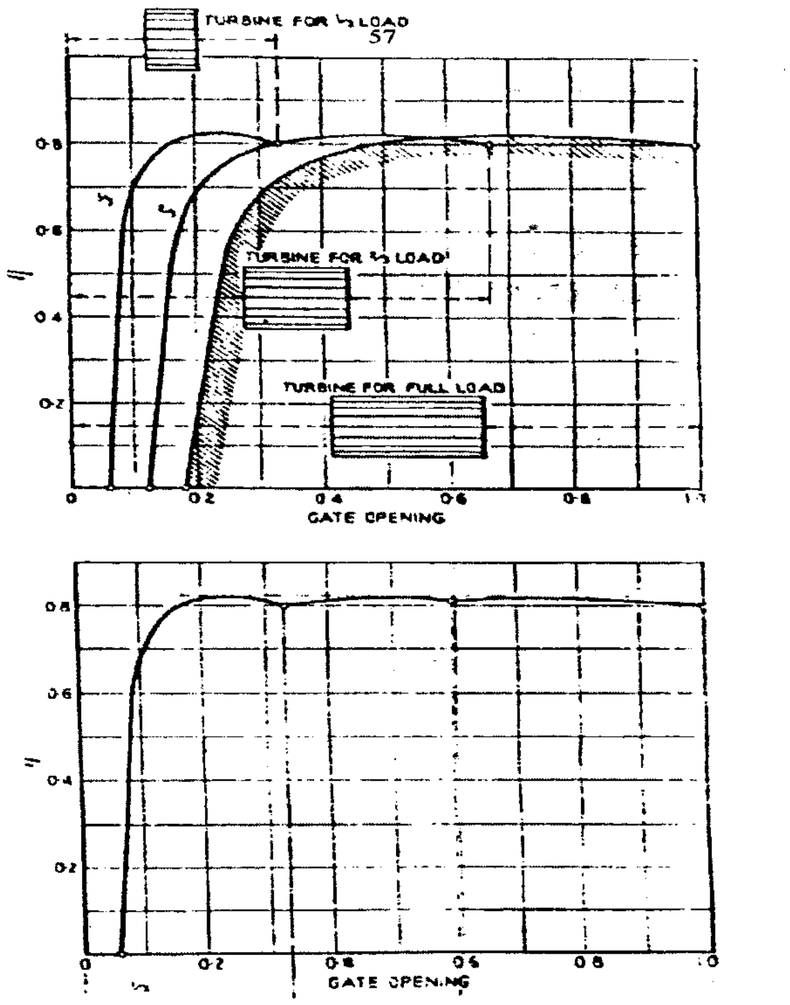


Fig. 4.15

Efficiency curve of a cross-flow turbine at full-load operation (shaded). Two further curves for 1/3 and 2/3 load derived from it by shortening the abscissae. Lower diagram: combining the three curves into a single overall efficiency curve. Ideal cross-flow turbine with divided gate

Figure 2. 18 efficiency of a cross flow turbine.

CHAPTER 3

CHAPTER 3

FRANCIS TURBINE

3.1 INTRODUCTION

Although the Pelton wheel is efficient and reliable when operating under large heads, it is less suited to smaller heads. As it was explained before, this is due to the fact that operating in atmospheric air, the head from nozzle to tail race is wasted. Because of this wasted head, Pelton wheels are usually employed for high heads, i.e. from 200 to 1 km. For lower heads, to avoid bulky and slow running machines, turbines of the reaction types are more suitable.

3.2 Reaction Turbine Operation:

The other main type of energy-producing hydroturbine is the reaction turbine, which consists of fixed guide vanes called stay vanes, adjustable guide vanes called wicket gates, and rotating blades called runner blades Fig. (3.1). Flow enters tangentially at high pressure, is turned toward the

runner by the stay vanes as it moves along the spiral casing or volute, and then passes through the wicket gates with a large tangential velocity component. Momentum is exchanged between the fluid and the runner as the runner rotates, and there is a large pressure drop. Unlike the impulse turbine, the water completely fills the casing of a reaction turbine. For this reason, a reaction turbine generally produces more power than an impulse turbine of the same diameter, net head, and volume flow rate. The angle of the wicket gates is adjustable so as to control the volume flow rate through the runner.

(In most designs the wicket gates can close on each other, cutting off the flow of water into the runner.) At design conditions the flow leaving the wicket gates impinges parallel to the runner blade leading edge (from a rotating frame of reference) to avoid shock losses. Note that in a good design, the number of wicket gates does not share a common denominator with the number of runner blades. Otherwise there would be severe vibration caused by simultaneous impingement of two or more wicket gate wakes onto the leading edges of the runner blades.

For example, in Fig. (3.1) there are 17 runner blades and 20 wicket gates. These are typical numbers for many large reaction hydroturbines, as shown in the photographs in Figs. (3.2) and (3.3). The number of stay vanes and wicket gates is usually the same (there are 20 stay vanes in Fig. (3.1). this is not a problem since neither of them rotates, and unsteady wake interaction is not an issue.

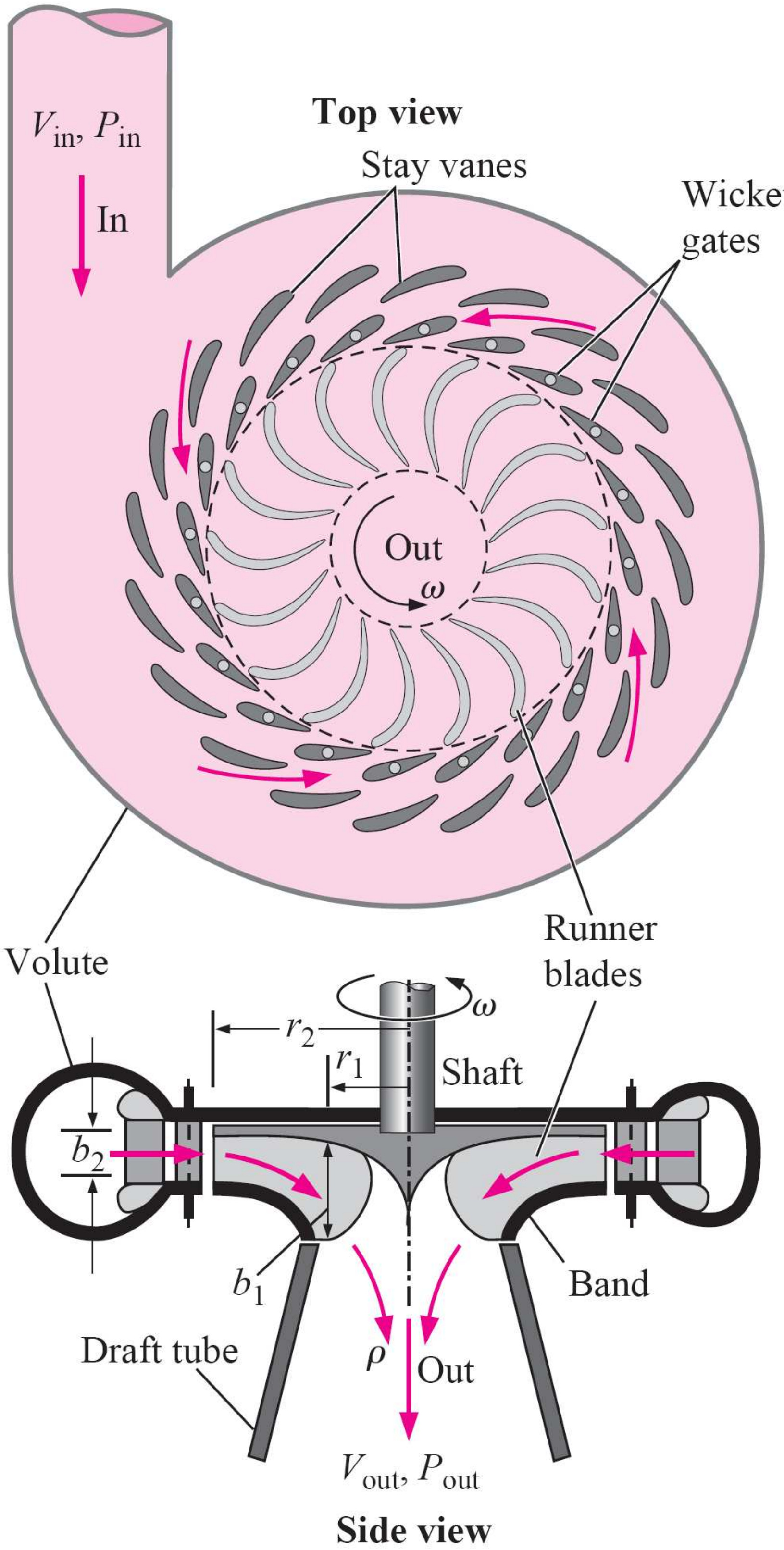


Figure 3. 1 Reaction turbine

Figure 3. 2 The runner of a Francis radial-flow turbine used at the Round Butte hydroelectric power station in Madras

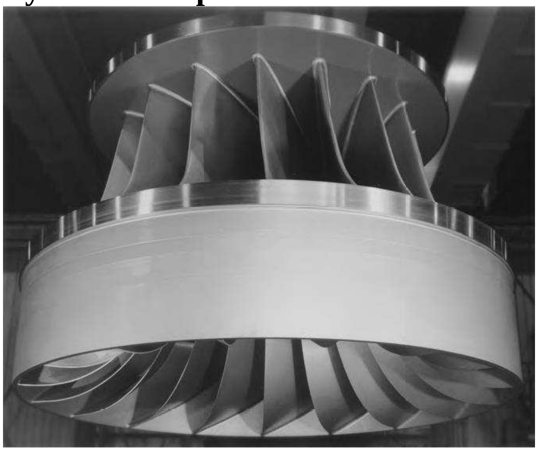


Figure 3. 3 The runner of a Francis mixed-flow turbine used at the Smith Mountain hydroelectric power station in Roanoke

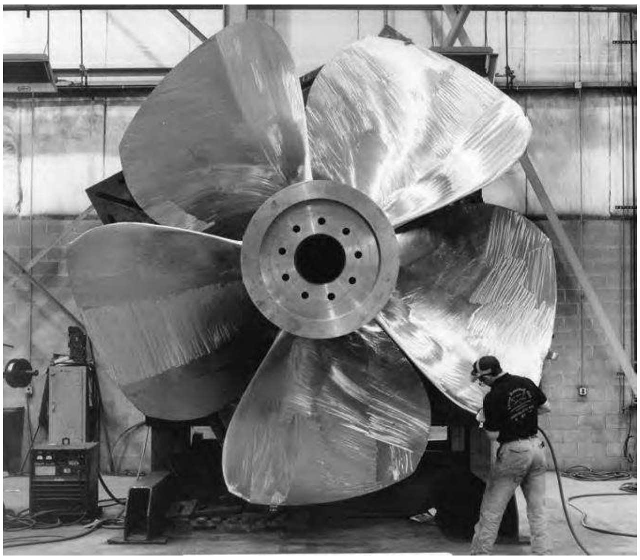


Figure 3. 4 The five-bladed propeller of a Kaplan turbine used at the Warwick hydroelectric power station in Cordele

There are two main types of reaction turbine—Francis and Kaplan. The Francis turbine is somewhat similar in geometry to a centrifugal or turbine. The Francis turbine is named in honor of James B. Francis (1815–1892), who developed the design in the 1840s. In contrast, the Kaplan turbine is somewhat like an axial-flow fan running backward. If you have ever seen a window fan start spinning in the wrong direction when wind blows hard into the window, you can visualize the basic operating principle of a Kaplan turbine. The Kaplan turbine is named in honor of its inventor, Viktor Kaplan (1876–1934). There are actually several subcategories of both Francis and Kaplan turbines, and the terminology used in the hydroturbine field is not always standard.

Recall that we classify dynamic pumps according to the angle at which the flow exits the impeller blade—centrifugal (radial), mixed flow, or axial. In a similar but reversed manner, we classify reaction turbines according to the angle that the flow enters the runner Fig.(3.5). If the flow enters the runner radially as in Fig. (3.5a), the turbine is called a Francis radial-flow turbine (see also Fig. (3.1)). If the flow enters the runner at some angle between radial and axial Fig. (3.5b), the turbine is called a Francis mixed-flow turbine. The latter design is more common. Some hydroturbine engineers use the term “Francis turbine” only when there is a band on the runner as in Fig.(3.5b). Francis turbines are most suited for heads that lie between the high heads of Pelton wheel turbines and the low heads of Kaplan turbines.

A typical large Francis turbine may have 16 or more runner blades and can achieve a turbine efficiency of 90 to 95 percent. If the runner has no band, and flow enters the runner partially turned, it is called a propeller mixed-flow turbine or simply a mixed-flow turbine Fig. (3.5c). Finally, if the flow is turned completely axially before entering the runner Fig. (3.5d), the turbine is called an axial-flow turbine. The runners of an axial-flow turbine typically have only three to eight blades, a lot fewer than Francis turbines. Of these there are two types: Kaplan turbines and propeller turbines.

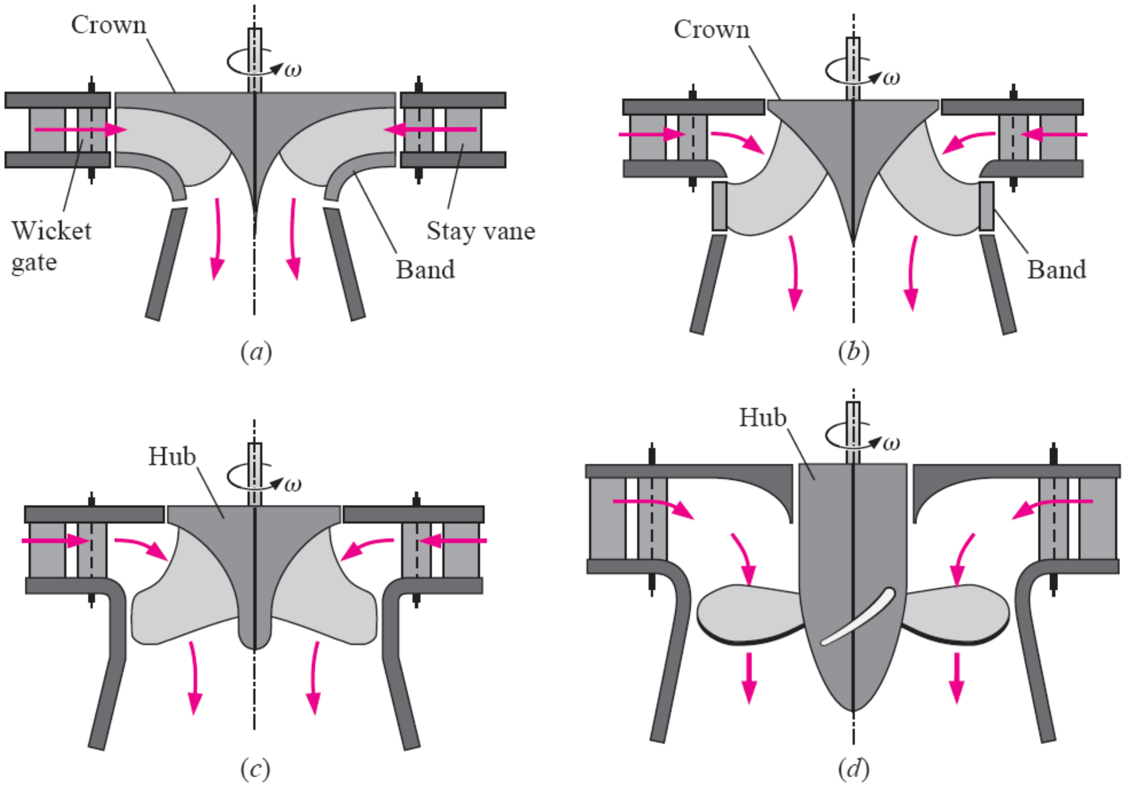


Figure 3. 5 The distinguishing characteristics of reaction turbines

(a) Francis radial flow. **(c) Propeller mixed flow.**

(b) Francis mixed flow. **(d) Propeller axial flow.**

Kaplan turbines are called **double regulated** because the flow rate is controlled in two ways—by turning the wicket gates and by adjusting the pitch on the runner blades. Propeller turbines are nearly identical to Kaplan turbines except that the blades are fixed (pitch is not adjustable), and the flow rate is regulated only by the wicket gates (**single regulated**). Compared to the Pelton and Francis turbines, Kaplan turbines and propeller turbines are most suited for low head, high volume flow rate conditions. Their efficiencies rival those of Francis turbines and may be as high as 94 percent. Figure (3.2) is a photograph of the radial-flow runner of a Francis radial flow turbine. The workers are shown to give you an idea of how large the runners are in a hydroelectric power plant. Figure (3.3) is a photograph of the mixed-flow runner of a Francis turbine, and Fig. (3.4) is a photograph of the axial-flow propeller of a Kaplan turbine. The view is from the inlet (top).

We sketch in Fig. (3.6) a typical hydroelectric dam that utilizes Francis reaction turbines to generate electricity. The overall or gross head H_{gross} is defined as the elevation difference between the reservoir surface upstream of the dam and the surface of the water exiting the dam,

$$H_{\text{gross}} = z_A - z_E.$$

If there were no irreversible losses anywhere in the system, the maximum amount of power that could be generated per turbine would be of course, there are irreversible losses throughout the system, so the power actually produced is lower than the ideal power given by the equation:

$$\textit{Ideal power production:} \qquad \dot{W}_{\text{ideal}} = \rho g \dot{V} H_{\text{gross}}$$

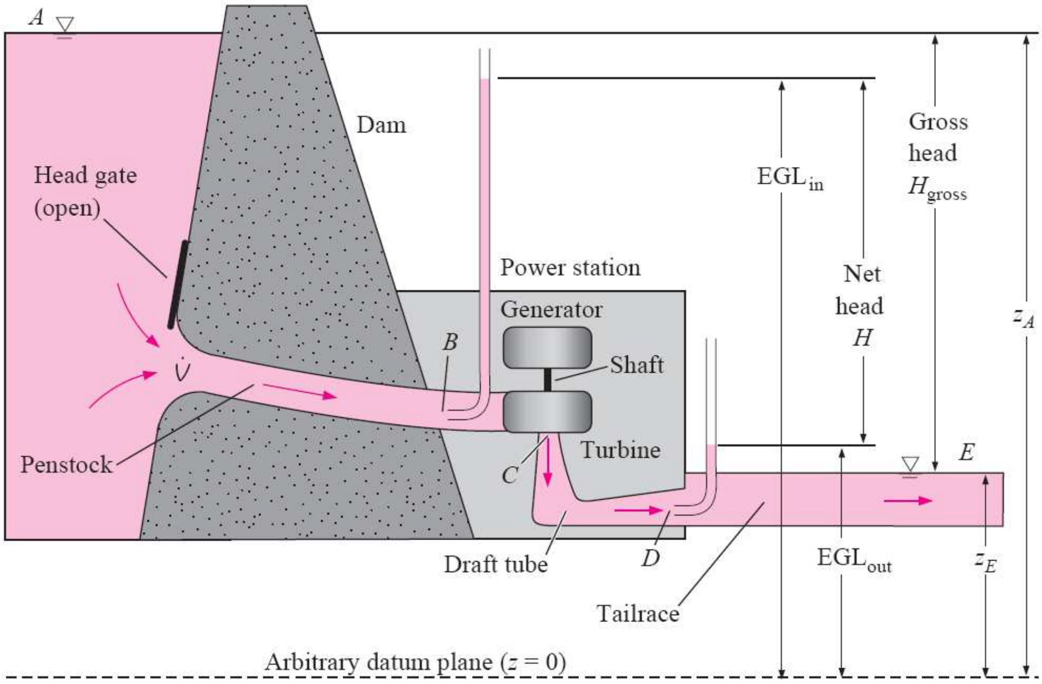


Figure 3. 6 Typical setup and terminology for a hydroelectric plant

We follow the flow of water through the whole system of Fig. (3.6), defining terms and discussing losses along the way. We start at point A upstream of the dam where the water is still, at atmospheric pressure, and at its highest elevation, z_A. Water flows at volume flow rate $V^{\cdot}$ through a large tube through the dam called the penstock. Flow to the penstock can be cut off by closing a large gate valve called a head gate at the penstock inlet. If we were to insert a Pitot probe at point B at the end of the penstock just before the turbine, as illustrated in Fig. (3.6), the water in the tube would rise to a column height equal to the energy grade line EGL_{in} at the inlet of the turbine. This column height is lower than the water level at point A, due to irreversible losses in the penstock and its inlet. The flow then passes through the turbine, which is connected by a shaft to the electric generator. Note that the electric generator itself has irreversible losses. From a fluid mechanics perspective, however, we are interested only in the losses through the turbine and downstream of the turbine.

After passing through the turbine runner, the exiting fluid (point C) still has appreciable kinetic energy, and perhaps swirl. To recover some of this kinetic energy (which would otherwise be wasted), the flow enters an expanding area diffuser called a draft tube, which turns the flow horizontally and slows down the flow speed, while increasing the pressure prior to discharge into the downstream water, called the tailrace. If we were to imagine another Pitot probe at point D (the exit of the draft tube), the water in the tube would rise to a column height equal to the energy grade line labeled EGL_{out} in Fig. (3.6). since, the draft tube is considered to be an integral part of the turbine assembly, the net head across the turbine is specified as the difference between EGL_{in} and EGL_{out},

Net head for a hydraulic turbine: $$H = \mathrm{EGL_{in}} - \mathrm{EGL_{out}}$$

In words,

The net head of a turbine is defined as the difference between the energy grade line just upstream of the turbine and the energy grade line at the exit of the draft tube.

3.3 Francis Turbine

This type of machine covers tremendous ranges of Power, Rates of Mass Flow Rates and Rotational Speeds. Francis Turbines normally operate with heads in the range of 30-500 m producing hundreds of megawatts. The efficiency of large Francis turbines has gradually risen over the years and now is about 95%.

Operation:

The Fluid enters a spiral casing, called a volute or scroll case, which completely surrounds the runner. The cross-sectional area of the volute decreases along the fluid path in such a way as to keep the fluid velocity constant in magnitude. From the volute the fluid passes between stationary guide vanes mounted all around the periphery of the runner. The function of these guide vanes is to direct the fluid on to the runner at the angle appropriate to the design and can be turned around pivots. These gates are also termed *Wicket Gates*. In its passage through the runner, the runner blades deflect the fluid so that its angular momentum is changed.

From the center of the runner the fluid is turned into the axial direction and flows to the waste via the draft tube. The lower end of the draft tube must, under all conditions of operation be submerged below the level of the water in the Tail race, that is the channel which carries the used water away. Only in this way it can be ensured that a hydraulic turbine is full of water. Large turbines may also have a “stay ring” or fixed vanes outside the ring of the guide vanes. Their main function is to act as columns helping to support the weight of the electrical generator above the turbine. They are so shaped so as to conform to the streamlines of the flow approaching the guide vanes.

3.3.1 Velocity triangles

Consider an inward flow Francis turbine represented diagrammatically in Fig. (3.1). A section of the runner guide vane ring, showing the blades, vanes and velocity triangles, is given in Fig. (3.7)

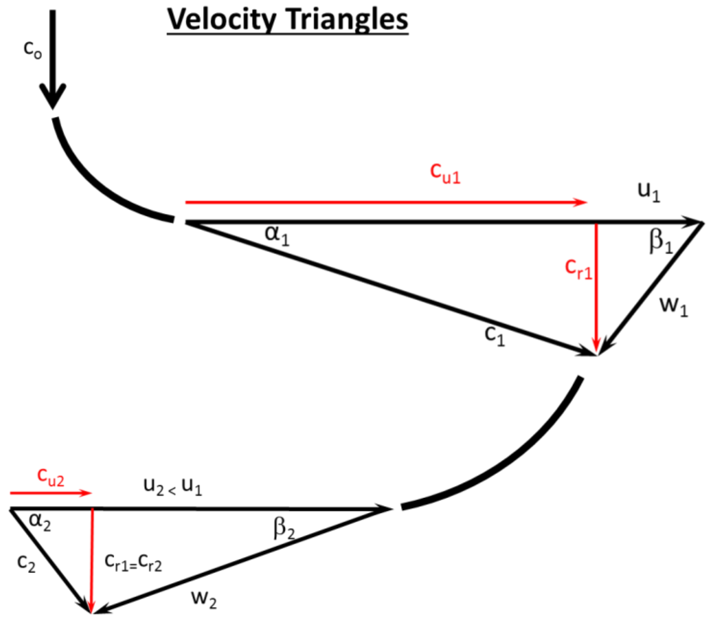


Figure 3. 7 velocity triangles

The total head available to the machine is H and the water velocity on entering the guide vanes is C_o. the velocity leaving the guide vanes is C_1 and is related to C_o by the continuity equation:

$$C_O A_O = C_{r1} A_1$$

But

$$C_{r1} = C_1 * \mathrm{Sin}(\alpha_1) \text{ , so that}$$

$$C_O A_O = A_1 C_1 * \mathrm{Sin}(\alpha_1)$$

The direction of C_1 is governed by the Guide vane angle α_1. It is chosen in such a way that the relative velocity meet the runner blade tangentially, i.e. it makes an angle β_1 with the tangent at blade inlet.

3.3.1.1 Types of velocity triangles

The condition of radial flow at inlet may be achieved by making the inlet blade angle β_1 equal 90°, such that the absolute tangential velocity component at inlet C_{u1} is equal the blade peripheral velocity U_1 as shown in Fig. (3.8) from the inlet velocity diagram. The Euler's head equal:

$$H_E = \frac{U_1^{\,2} - U_2 C_{u2}}{g}$$

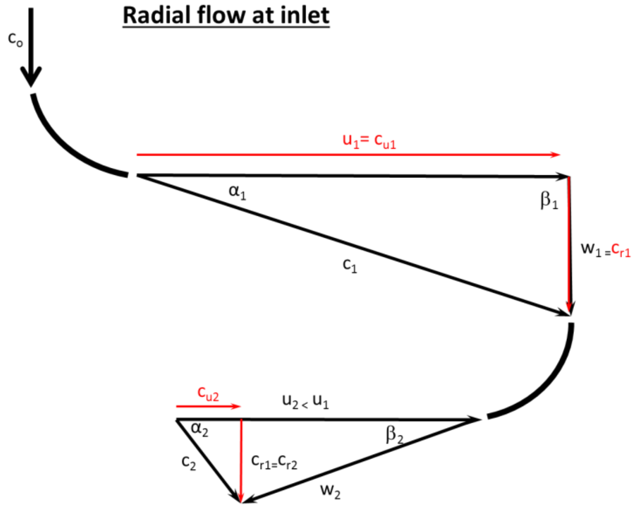


Figure 3. 8 radial flow at inlet

The energy transferred H_E is given by Euler's Equation, which for the maximum energy transfer condition secured when $C_{U2} = 0$, takes the form

$$H_E = \frac{U_1 C_{u1}}{g}$$

The condition of no whirl component at outlet may be achieved by making the outlet blade angle β_2, such that the absolute velocity at outlet C_2 is radial as shown in Fig. (3.9) from the outlet velocity diagram.

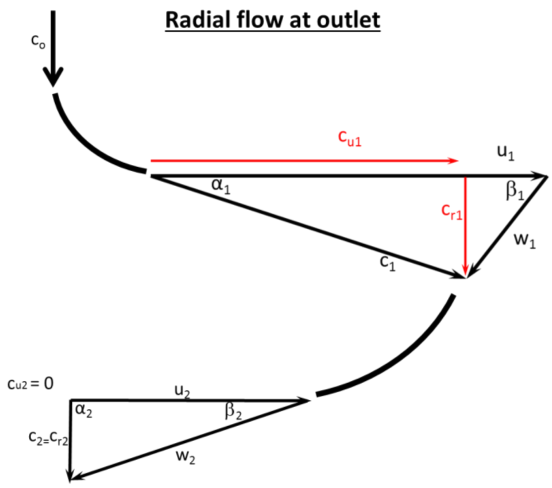


Figure 3. 9 radial flow at outlet

The condition of both radial flow at inlet and outlet as shown in Fig. (3.10) from the inlet and outlet velocity diagrams. The Euler's head equal:

$$H_E = \frac{U_1^{\;2}}{g}$$

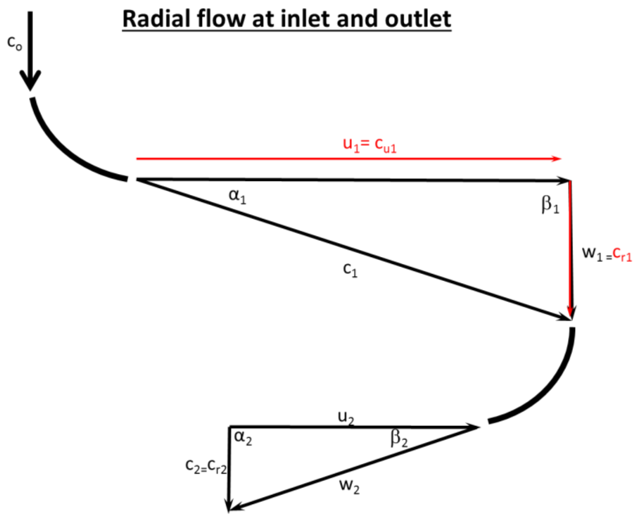
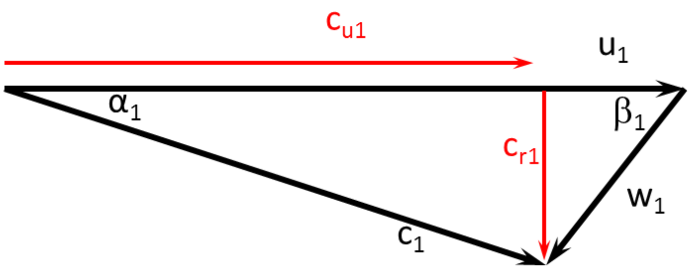


Figure 3. 10 radial flow at inlet and outlet

3.3.1.2 Inlet velocity triangle



$$\tan\beta_1 = \frac{C_{r1}}{U_1 - C_{u1}}$$

$$U_1 = C_{r1}\cot\beta_1 + C_{u1}$$

$$C_{u1} = C_{r1}\cot\alpha_1$$

Eliminating C_{u1} from the two equations:

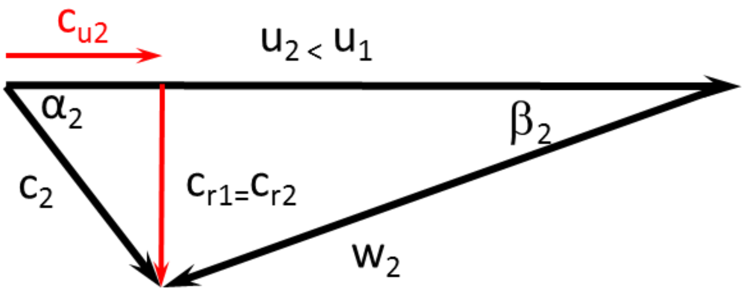
$$U_1 = C_{r1}(\cot\beta_1 + \cot\alpha_1)$$

Therefore,

$$\cot\beta_1 = \frac{U_1}{C_{r1}} - \cot\alpha_1$$

$$\frac{U_1}{C_{r1}} = \cot\beta_1 + \cot\alpha_1$$

3.3.1.3 Outlet velocity triangle



$$\tan\beta_2 = \frac{C_{r2}}{U_2 - C_{u2}}$$

$$U_2 = C_{r2}\cot\beta_2 + C_{u2}$$

$$C_{u2} = C_{r2}\cot\alpha_2$$

Eliminating C_{u1} from the two equations:

$$U_2 = C_{r2}(\cot\beta_2 + \cot\alpha_2)$$

Therefore,

$$\cot\beta_2 = \frac{U_2}{C_{r2}} - \cot\alpha_2$$

$$\frac{U_2}{C_{r2}} = \cot\beta_2 + \cot\alpha_2$$

3.3.2 Net head

The total energy at the inlet to the runner consists of the velocity head $C_1^2/2g$ and pressure head H_1. In the runner, the fluid energy is decreased by H_E, which is transferred to the runner. Water leaves the impeller with kinetic energy $C_2^2/2g$. Thus, the following energy equation holds:

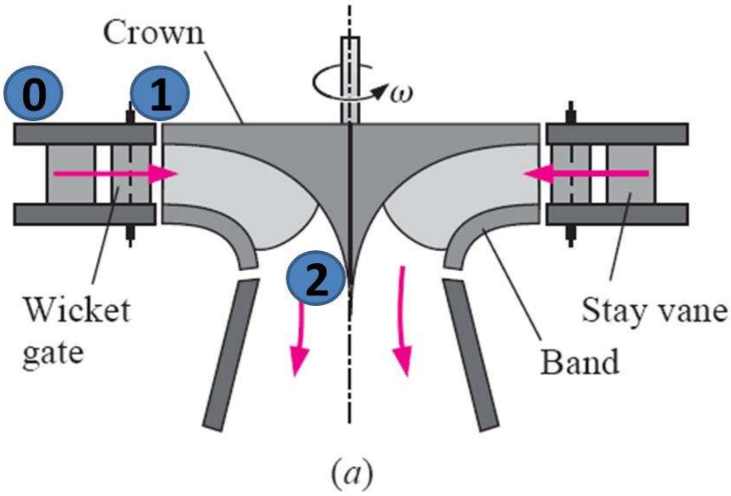


Figure 3. 11 net head and Euler's head

At point (0): the total energy at the volute inlet

$$H_{net} = \frac{P_0}{\gamma} + \cancel{\frac{C_0^2}{2g}} + z_0$$

At point (1): The total energy at the inlet to the runner consists of the velocity head and pressure head H_1.

$$H_{net} = \frac{P_1}{\gamma} + \frac{C_1^2}{2g} + z_1 + h_{L0\to1}$$

3.3.3 Euler's head

At point (2): at runner exit, the fluid energy is decreased by H_E, which is transferred to the runner. Water leaves the impeller with kinetic energy

$$H_{net} = H_E + \cancel{\frac{P_2}{\gamma}} + \frac{C_2^2}{2g} + \cancel{z_2} + h_{L0\to2}$$

In which h_{L0-1} is the loss of head in the guide vane ring and h_{L0-2} is the loss in the whole turbine, including entry, guide vanes and runner.

The energy transferred H_E is given by Euler's Equation, which for the maximum energy transfer condition secured, takes the form

$$H_E = \frac{U_1 C_{u1} - U_2 C_{u2}}{g}$$

3.3.4 Work and Efficiency

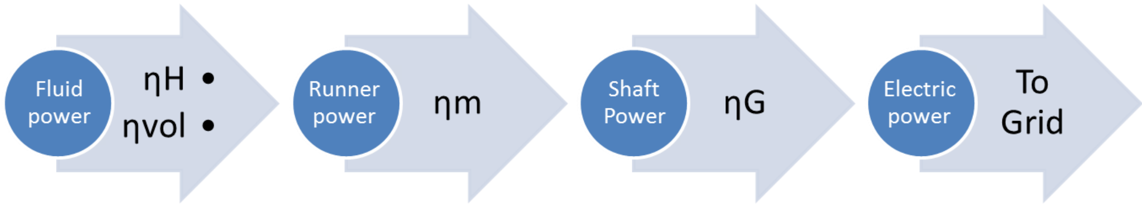


Figure 3. 12 efficiency of Francis turbine

$$\eta_H = \frac{H_E}{H_{net}} \qquad \eta_m = \frac{P_{sh}}{\gamma Q_{act} H_E}$$

$$\eta_{vol} = \frac{Q_{act}}{Q_{th}} \qquad Q_{act} = 2\pi r_1 b_1 k_1 c_{r1} = 2\pi r_2 b_2 k_2 c_{r2}$$

$$\eta_{o\ turbine} = \frac{P_{sh}}{\gamma Q H_{net}} = \eta_H \times \eta_{vol} \times \eta_m$$

$$\eta_{o\ plant} = \frac{P_{elec}}{\gamma Q H_{net}} = \eta_H \times \eta_{vol} \times \eta_m \times \eta_G$$

b_1=*runner width (breadth) at inlet*

b_2=*runner width (breadth) at outlet*

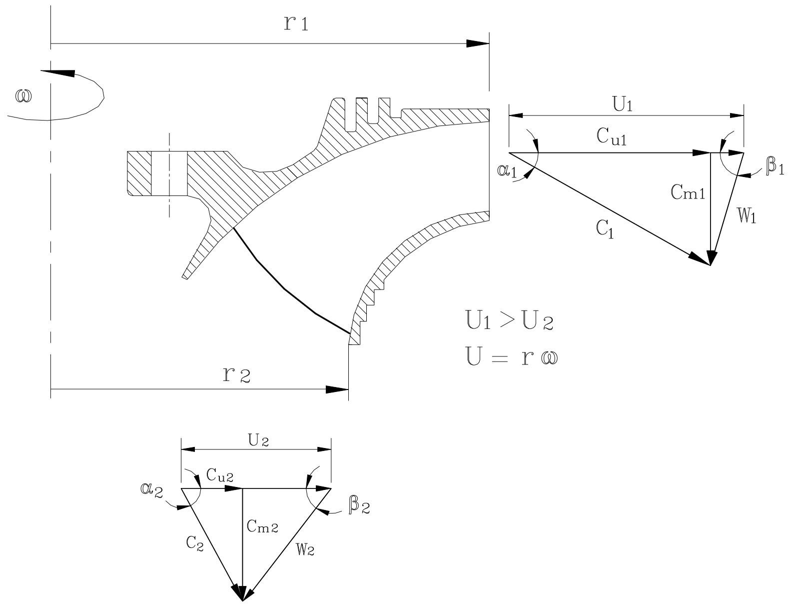


Figure 3. 13 runner blade dimensions

3.3.5 Losses in Francis Turbines

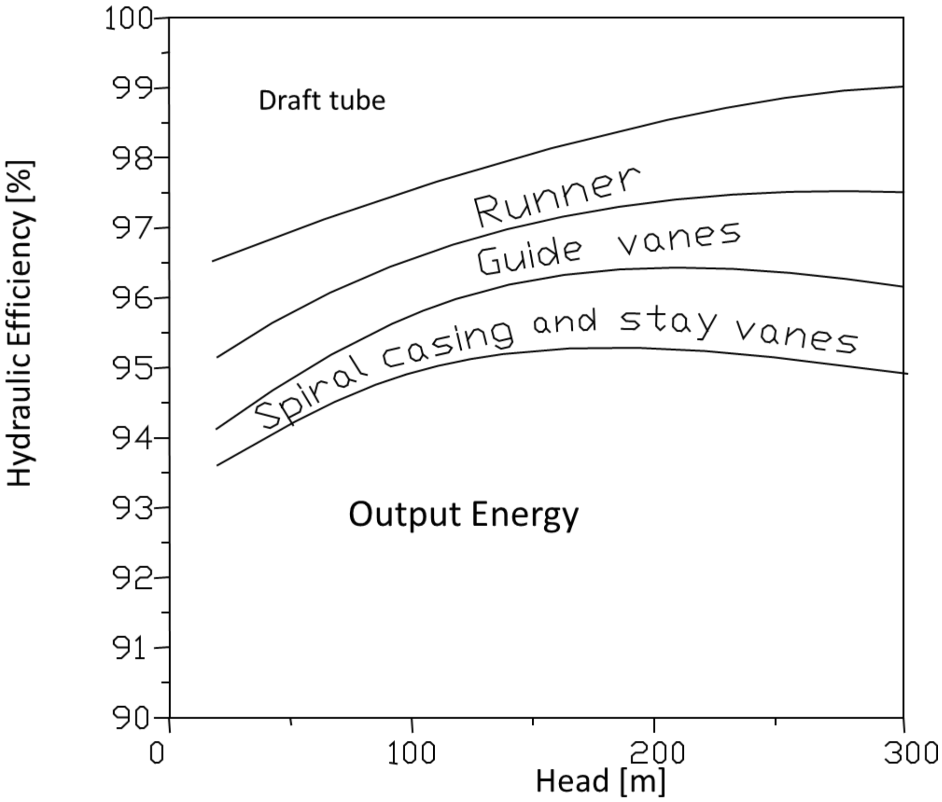


Figure 3. 14 losses in Francis turbine

The relationship between the runner speed and the spouting speed $\sqrt{2gH}$ for the Francis turbines not so rigidly defined as for the Pelton Wheel. In practice, the Speed ratio $U_2 / \sqrt{2gH}$ is contained within the limits 0.6 to 0.9.

We have to mention that the simple expressions derived here do not apply exactly to the flow considered as whole. This is due to that, many machines are so constructed that uniformity of conditions at inlet and outlet is impossible to achieve. In mixed flow machines the fluid leaves the rotor at various radii. Even Francis turbines nowadays usually have some "mixed flow" at the outlet; more ever, the inlet and outlet edges of the blades are not always parallel to the axis of rotation. Although the expressions involve the assumption that the velocities at inlet and outlet are uniform around the circumference; in practice this condition is not fulfilled even for a runner in which all the flow is in the plane of rotation. Individual particles of the fluid may have different velocities. Since guide vanes and runner blades are both limited in number, the directions taken by individual particles may differ appreciably from that indicated by the velocity diagram. Even the average direction of the relative velocity may differ from that of the blade it is supposed to follow. Thus the velocity diagrams and the expressions based on them should be regarded only as a first approximation to the truth.

In spite of these defects, however the simple theory is useful in indication how the performance of the machine varies with changes in the operating conditions, and in what way the design of the machine should be altered to modify its characteristics.

3.3.6 Speed Regulation

The speed regulation of a turbine is an important and complicated problem. The magnitude of the problem varies with size, type of machine and installation, type of electrical load, and whether or not the plant is tied into an electrical grid. Note that runaway or no-load speed can be higher than design speed by factors as high as 2.6. This is an important design consideration for all rotating parts, including the generator.

The speed of a turbine has to be controlled to a value that matches the generator characteristics and the grid frequency:

$$n = 120 f / N\text{p}$$

where *n* is turbine speed in rpm, *f* is the required grid frequency in Hz, and *Np* is the number of poles in the generator. Typically, *Np* is in multiples of 4. There is a tendency to select higher speed generators to minimize weight and cost. However, consideration has to be given to speed regulation.

Regulation of speed is normally accomplished through flow control. Adequate control requires sufficient rotational inertia of the rotating parts. When load is rejected, power is absorbed, accelerating the flywheel; and when load is applied, some additional power is available from deceleration of the flywheel.

3.3.7 Governing Francis Turbine

An inward – flow turbine has the valuable feature of inherent stability, that is, it is to some extent self-governing. This is because a centrifugal head like that in a forced vortex is developed in the fluid that rotates with the runner. The centrifugal head balances part of the supply head. If for any reason the rotational speed of the runner falls, the centrifugal head also falls, with the result that a higher rate of flow through the machine is possible and the speed raises again, the converse action results from an increase in speed.

Considering the alternative form of Euler's Equation for turbines,

$$W = \frac{1}{2g}[(C_1^2 - C_2^2) + (U_1^2 - U_2^2) - (w_1^2 - w_2^2)]$$

For a turbine, that is, a machine in which work is done by the fluid, this expression must be positive. This is most easily achieved by the inward flow arrangement. Then $U_1 > U_2$ and, since the flow passages decrease rather than increase in cross-sectional area, w_2 usually exceeds w_1.The contributions of the second and third brackets to the work done by the fluid are thus positive. By appropriate design, however, an outward flow turbine although seldom desirable, is possible.

Governing of the turbine means that to allow the speed of the machine to be maintained even when the power demand on the shaft varies. The guide blades of a Francis turbine are pivoted and connected by levers and links to the regulating ring. The regulating ring is attached with two regulating rods connected to the regulating lever as shown in Fig (4.20), this regulating lever is connected with the regulating shaft, which is operated by the piston of servomotor.

3.3.7.1 Load Decrease:

When load on the turbine decreases, speed tends to increase, this moves fly balls of the actuator away and thus raises the sleeve. The main lever on the other side of the fulcrum pushes down the control valve rod and opens the port **B**. with the opening of the port **B**, oil under the pressure enters the servomotor from left and pushes the piston to move towards the right. The previous oil at the right side of the servomotor is thus pushed back into the oil sump through port A and upper part of the distributing valve. When the piston of the servo-motor moves towards the right, regulating ring is rotated to decrease the passage between the guide vanes by changing guide vane angles, thus the quantity of water reaching the runner blades is reduced. Therefore gradually speeds come to normal and then actuator, main lever and distribution valve attain their normal position.

3.3.7.2 Load Increase:

Similarly when the load on the turbine increases exactly reversed action takes place. Regulating ring is moved in the reversed direction to increase the passage between the guide blades thus allowing more water to meet the increased load demand

3.3.7.3 Water Hammer Pre-Caution:

Sudden reduction in the passage between the guide blades may cause water hammer, which can be prevented by providing a relief valve near the turbine, which diverts the water directly to the tailrace. Thus it functions similar to that of jet deflector as in Pelton wheel, thus double regulation (speed and pressure) is also performed in Francis turbine.

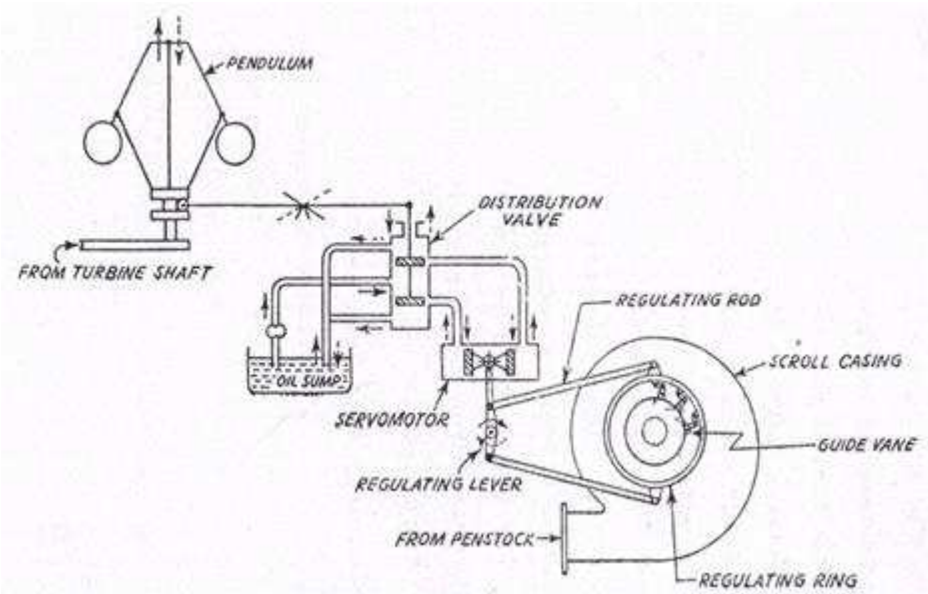


Figure 3. 15 Governing of Francis turbine

CHAPTER 4

CHAPTER 4

KAPLAN TURBINE

4.1 INTRODUCTION

The power developed by a turbine is proportional to a product of the total head available and the flow rate. Therefore, the power required from a turbine may (within limits) be obtained by a desired combination of these two quantities.

4.2 Criteria for Improvising Axial Turbines:

For a Pelton wheel, in order to achieve high jet velocities, it is necessary that the total head is large and, consequently, the flow rate is usually small. However, the Pelton wheel becomes unsuitable if the head available is small, so that in order to achieve the desired power the quantity has to be greater. A Francis-type radial turbine is then used. Its proportions depend upon the flow rate, which must be increased, the blade passages become shorter but wider, and a mixed flow type turbine results. "*If the process is carried further, an axial flow turbine is obtained because the maximum flow rate may be passed through when the flow is parallel to the axis*".

4.3 Classification:

Axial propeller turbines are generally either of fixed blade or Kaplan (adjustable blade) variety.

4.4 Construction & Operation:

(A) The Classical propeller Turbine: illustrated in Fig. (4.21), is a vertical axis machine with a scroll case and a radial wicket gate configuration that is very similar to the flow inlet for a Francis turbine. The flow enters radially inward and makes a right-angle turn before entering the runner in an axial direction the runner usually has four or six blades and closely resembles ship's propeller. Apart from frictional effects, the flow approaching the runner blades is that of a free vortex (whirl velocity inversely proportional to radius) whereas the velocity of the blades them selves is directly proportional to radius. To cater for the differing relation between the fluid velocity and the blade velocity as the radius increase, the blades are twisted, as shown in Fig. (4.22), where the angle with the axis being greater at the tip than at the hub.

(B) The Kaplan Turbine: The blade angles may be fixed if the available head and the load are both fairly constant, but where these quantities may

vary, a runner is used on which the blades may be turned about their own axes while the machine is running, Plate 6. When both guide vane angle and runner blade angle may thus be varied, a high efficiency can be maintained over a wide range of operating conditions. Such turbine is known as a Kaplan turbine after its inventor, the Austrian engineer Viktor Kaplan (1876-1934) Fig. (4.23) shows the efficiency curve of a Kaplan turbine with fixed blades plotted to the same scale on a base representing the output or gate opening both turbines having the same specific speed.

Strength Issue: The runner blades must be long in order to accommodate the large flow rate and, consequently considerations of strength required to transmit the tremendous torques involved impose the necessity for large blade chords, Thus pitch/chord ratios of about 1 to 1.5 are used and, hence, the number of blades is small, usually 4,5 or 6, Plate 8.

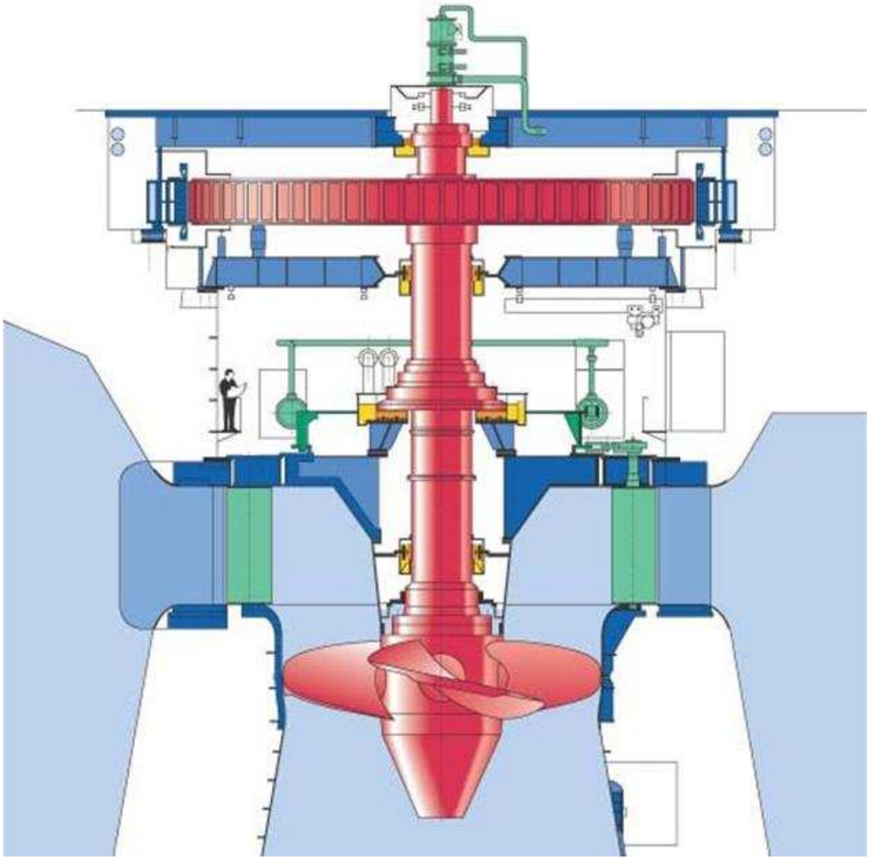


Figure 4. 1 Kaplan turbine configuration

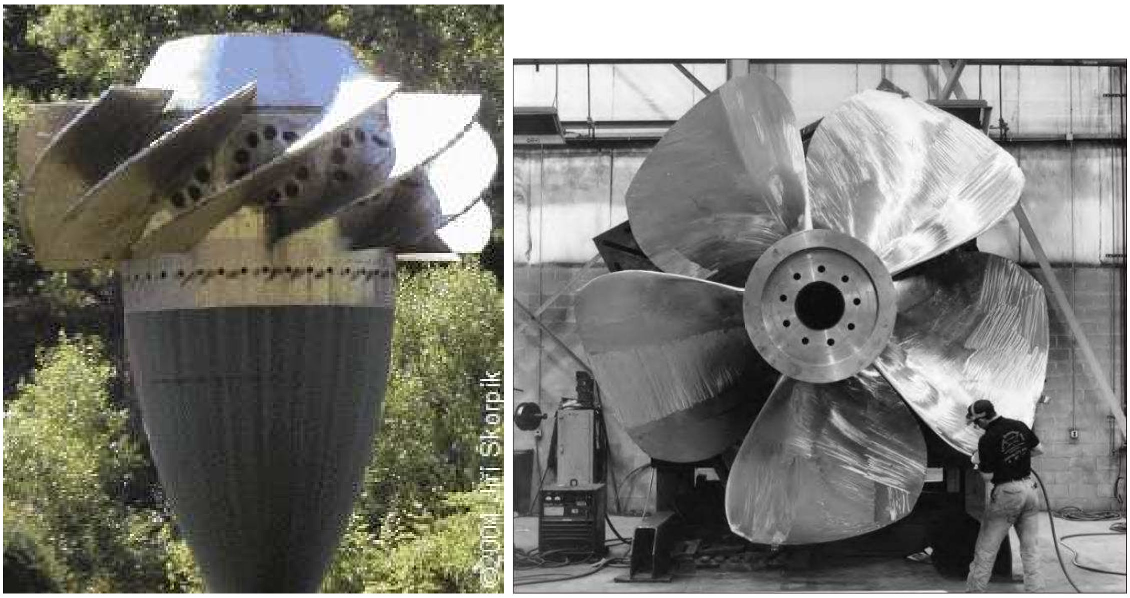


Figure 4. 2 Kaplan turbine

4.5 Governing the Kaplan Turbines

As explained in the previous paragraph that Kaplan turbine has guide vanes as well as runner vanes adjustments. Fig (4.24) gives views of the runner and guide blades fully open in the left hand view and completely closed in the right hand side.

A governor is required to operate guide vanes as well as runner vanes simultaneously. Similarly to that of Francis turbine explained before, this governor employs a servomotor for the guide vanes, but it is also provided with another one for the rotor blades. *Both the servomotor distribution valves actuate runner vanes and guide vanes simultaneously in such a way that water pass through them without shock at all load conditions.* The mechanism operating runner vanes is located inside a hollow coupling fitted between the turbine and generator shaft as shown in Fig (4.25).

The Servomotor consists of a cylinder with piston working under oil pressure (supplied by distribution valve) on either side. This piston is connected to an operating rod which moves up and down and passes through the turbine shaft, which is made hollow for the propose. The operating rod actuates the crosshead which caries number of arms each connected with runner blades through small cranks. These later are connected to the runner blades with pivots casted integrally with the blades this mechanism is enclosed in the runner hub leaving blades outside.

4.6 Velocity Triangles

Consider the velocity triangles of an axial flow turbine as shown in Figure (4.3); the velocity of flow is axial at inlet and outlet and, of course, remains the same. The whirl velocity is tangential. The blade velocity at inlet and outlet is the same.

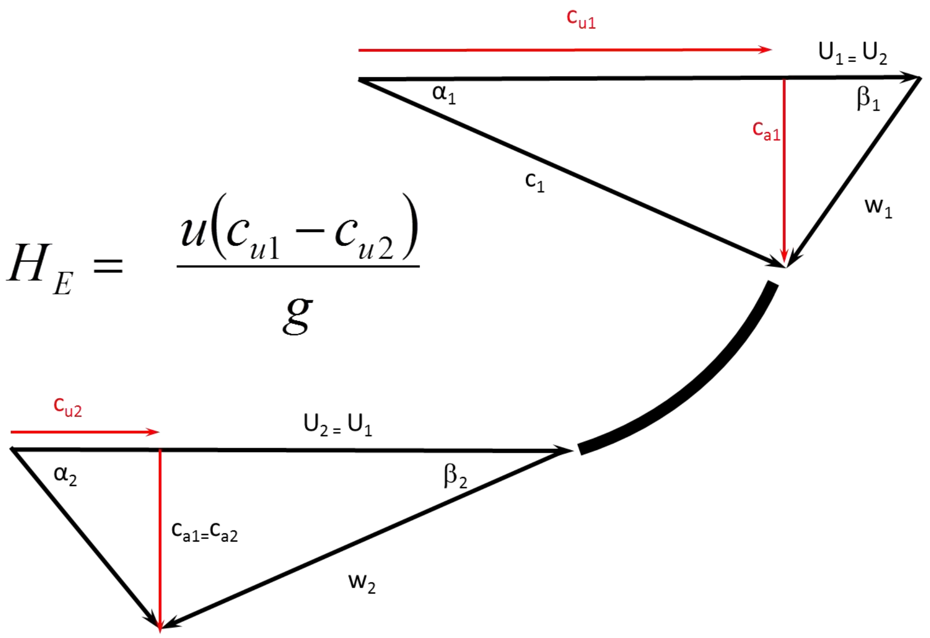


Figure 4. 3 Kaplan turbinevelocity triangles

Euler's equation will be

$$H_E = \frac{u(c_{u1} - c_{u2})}{g}$$

Consider the velocity triangles of an axial flow turbine as shown in Figure (4.4);

1- The velocity of flow is axial at inlet and outlet and, of course, remains the same.

2- The whirl velocity is tangential.

3- The blade velocity at inlet and outlet is the same.

4- The absolute velocity at outlet is axial

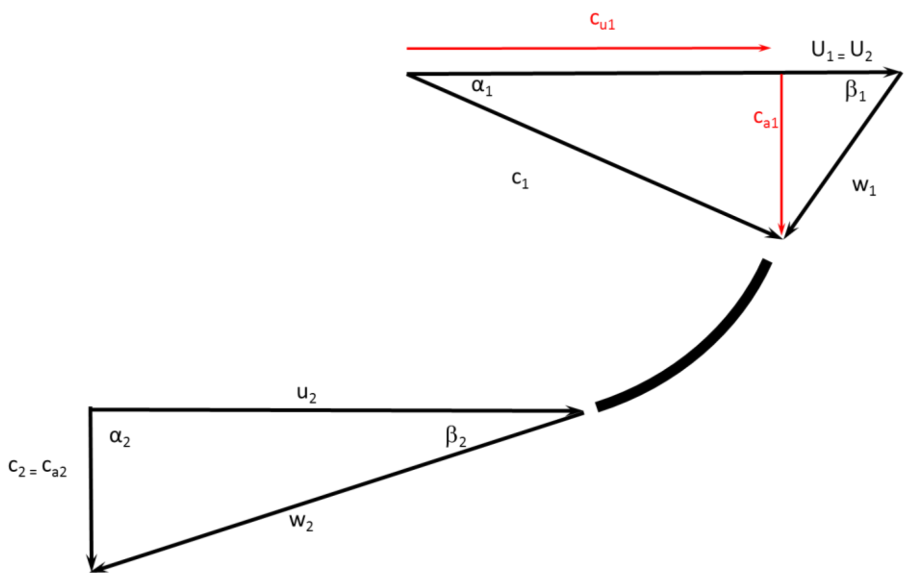


Figure 4. 4 velocity triangles for absolute velocity at outlet is axial

Euler's equation will be

$$H_E = \frac{uc_{u1}}{g}$$

In which:

$$U_1 - C_{u1} = C_a \cot \beta_1$$

4.7 Twisted blade

Since H_E should be the same at the blade tip and at the hub, but U is greater at the tip, it follows that C_{u1} must be reduced similarly, the velocity of flow C_a should remain constant along the blade and therefore (cot β_1) must be increased towards the tip of the blade. Thus β_1 has to be reduced, and consequently the blade must be twisted so that it makes a greater angle with the axis at the tip than it does at the hub as shown in figure (4.5).

$$H_E(const) = \frac{u\uparrow\uparrow \times c_{u1}\downarrow\downarrow}{g}$$

$$u\uparrow + c_{u1}\downarrow \Rightarrow (u - c_{u1})\uparrow\uparrow$$

$$\left(u - c_{u1}\right)\uparrow\uparrow \Rightarrow \cot\beta_1 \uparrow\uparrow \Rightarrow \beta_1 \downarrow\downarrow$$

$$\left(\beta_{tip}\right) \prec \left(\beta_{hub}\right)$$

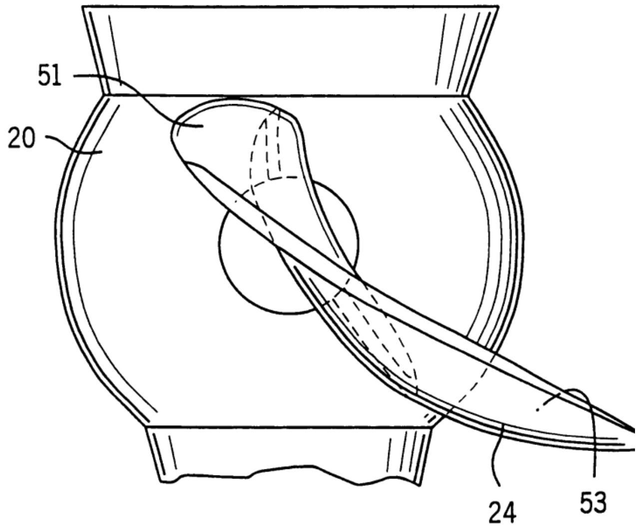


Figure 4. 5 twisted blade

However, this condition, known as the "Free *Vortex* "design, is not always met. It is then necessary to apply Euler's Equation to an element dr and integrate from R_1 to R_2 as follows and β_2 = f(r) must be known the discharge

$$Q = A\, C_a = \pi\, C_a\, (R_2^2 - R_1^2).$$

4.8 Isolated Blade:

The major assumption underlying the considerations of the previous section was that of an infinite number of blades in the runner.

In practice, of course, the number of blades is finite and their spacing depends on a particular inner design and therefore, may vary considerably. The distance between the adjacent blades, **S** is known as **pitch**, whereas the ratio of blade chord to the pitch

$$\delta = C / S$$

where,

$$S = \frac{2\pi . r}{z}$$

$$c = \text{chord}$$

δ is called **blade solidity** and is a measure of the closeness of blades. If the blades are very far apart, they must be treated as bodies in an external flow, provided their mutual interference is negligible.

In this section we will consider the blades far apart so that $S \rightarrow \infty$ and hence, $\delta = 0$. For such case, an approximate solution may be based on the isolated blade element theory.

The blade is divided into elements of ΔL width and is put an angle of incidence α with respect to the mean relative inflow velocity W_m shown in Fig (4.6). The lift L experienced by the element is perpendicular to W_m and the drag D is parallel to W_m. The resultant force R may be resolved into two perpendicular forces:

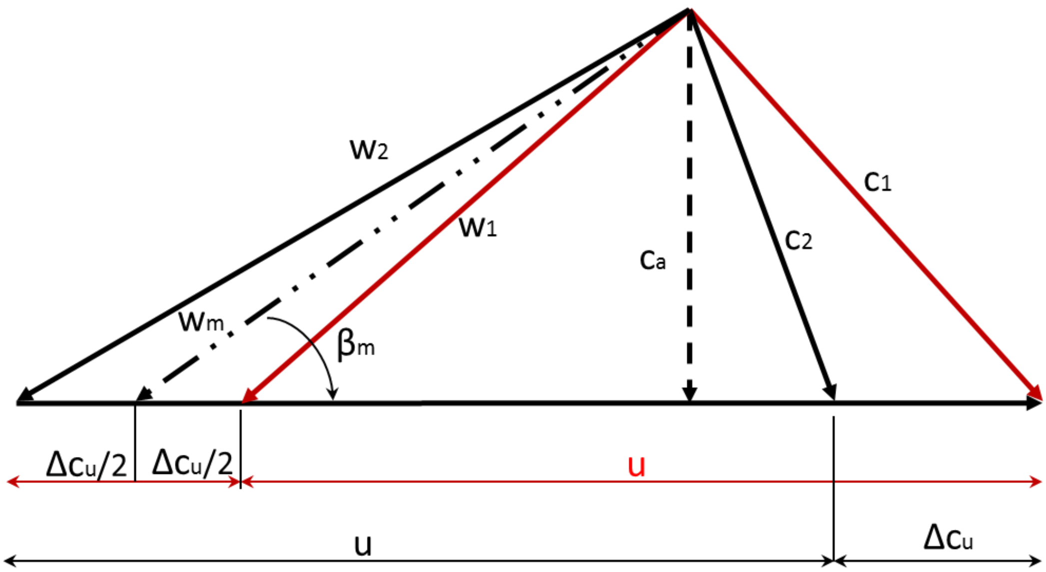


Figure 4. 6 isolated blade velocity triangles

Axial Force

$$F_a = L\cos\beta + D\sin\beta$$

Tangential Force

$$F_t = L\sin\beta - D\cos\beta$$

if, by definition,

$$L = \frac{1}{2}\rho w_m^2 C_L . C . \Delta L$$

where C is the Aerofoil chord and C_L is the *Lift Coefficient*

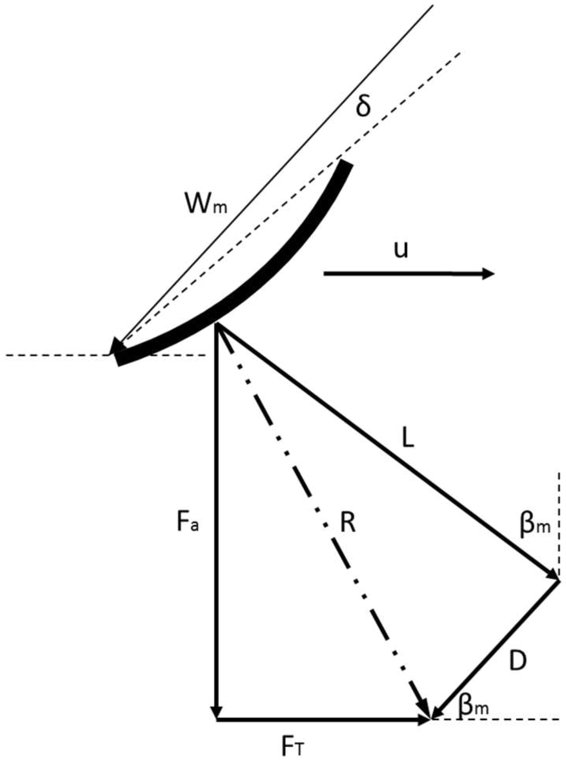


Figure 4. 7 forces applied on isolated blade

also

$$D = \frac{1}{2}\rho w_m^2 C_D.C.\Delta L$$

Where C_L is the *Drag Coefficient*

The forces become,

$$F_a = \frac{1}{2}\rho w_m^2 [C_L \cos\beta + C_D \sin\beta].C.\Delta L$$

and

$$F_t = \frac{1}{2}\rho w_m^2 [C_L \sin\beta - C_D \cos\beta].C.\Delta L$$

Equating the axial force F_a to the force produced by the fluid pressure, we get

$$F_a = \frac{2\pi.r}{z}(P_1 - P_2).\Delta L$$

where z is the number of the blades

$$P_1 - P_2 = \frac{1}{2}\rho w_m^2 [C_L \cos\beta + C_D \sin\beta].\left(\frac{C}{S}\right)$$

Where,

$$S = \frac{2\pi .r}{z}$$

Equating the tangential force to the force due to the change of momentum:

$$F_t = \frac{2\pi .r}{z} \Delta L \rho C_a . \Delta C_U$$

Then,

$$\Delta C_U = \frac{w_m^2 [C_L \sin\beta - C_D \cos\beta]}{2(S/C).C_a}$$

from the velocity diagram

$$\tan\beta = \frac{C_a}{U - (0.5)[C_{U1} + C_{U2}]}$$

and from Bernoulli equation

$$\frac{P_1 - P_2}{\gamma} = \frac{\Delta C_U}{g} . \frac{C_a}{\tan\beta} + h_F$$

and we get by substitution that,

$$h_F = \frac{W_m^2}{2g} . \frac{1}{(S/C)} . \frac{C_D}{\sin\beta}$$

The blade element efficiency is defined as the ratio of the net energy transferred to the turbine H to the total fluid energy change across the rotor (H + h_F)

$$\eta_B = \frac{H}{H + h_F}$$

4.9 Work and Efficiiency

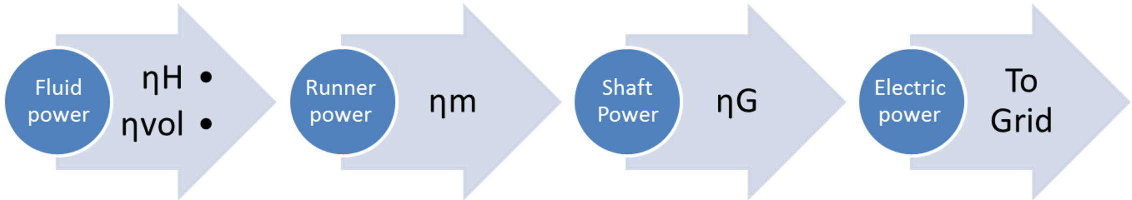


Figure 4. 8 work and efficiency of Kaplan turbine

$$\eta_H = \frac{H_E}{H_{net}}$$

$$\eta_m = \frac{P_{sh}}{\gamma Q_{act} H_E}$$

$$\eta_{vol} = \frac{Q_{act}}{Q_{th}}$$

$$Q_{act} = \frac{\boldsymbol{\pi}}{4}\left(d_o^2 - d_i^2\right) C_a k$$

4.10 Draft tube

An important element of a turbine installation is the draft tube, which is used to recover as much of the velocity head as possible as it leaves the runner. The recovery of the velocity head is very important:

a- At high specific speeds the velocity head at exit from the runner may be 15% of the net head of the installation.

b- At low Specific Speeds the velocity head is of the order of 3 to 4 % of the net head.

Actually, an efficient recovery of velocity head at runner exit results in a

1- Decrease of the exit diameter D2 of the turbine

2- And thus has a direct bearing on the cost of the turbine.

3- Besides, the reduction of kinetic energy of the water exiting the turbine runner, within limits a well - designed draft tube will permit installation of the turbine above the tail water elevation without losing any head.

Different designs of a draft tube are common, ranging from straight conical diffuser to configurations with bends and bifurcations. Some typical shapes and relative dimensions are shown in figure, where the dimensions are given in terms of the runner diameter.

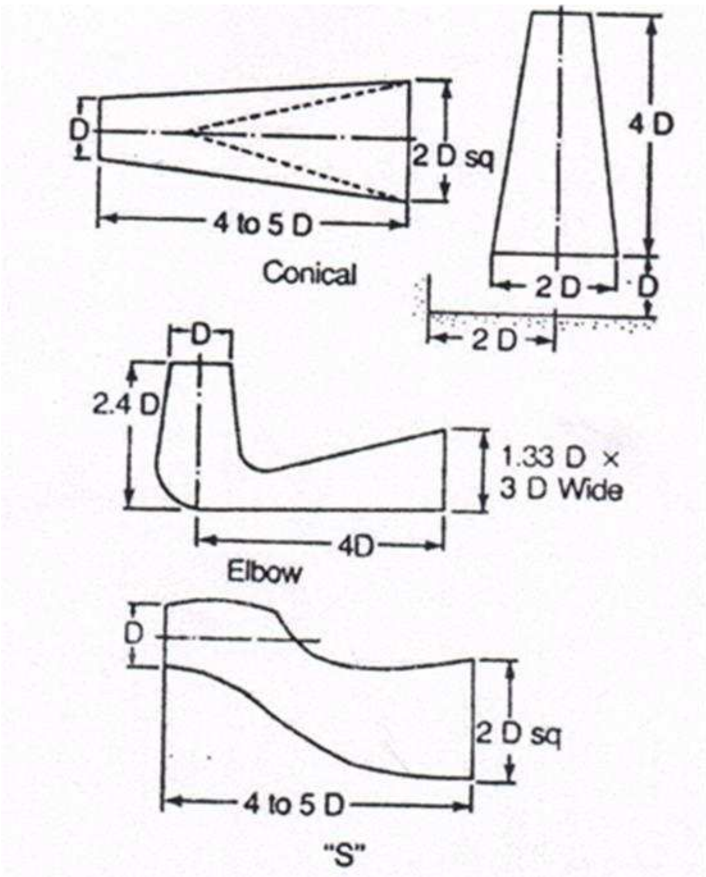


Figure 4. 9 typical draft tube dimensions

4.10.1 Work produced without draft tube

Applying the energy equation between 1 and 2 which are runner inlet and outlet respectively without using draft tube

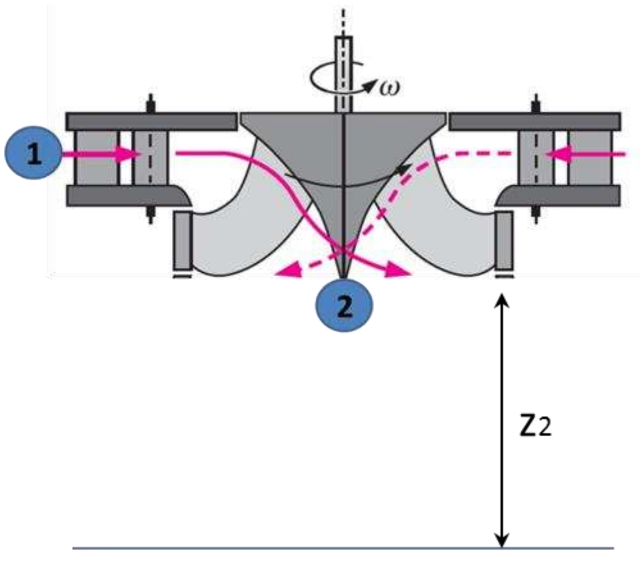


Figure 4. 10 work produced without draft tube

$$H_{net} = \frac{P_1}{\rho.g} + \frac{C_1^2}{2.g} + z_1 = \frac{\cancel{P_{atm}}}{\cancel{\rho.g}} + \frac{C_2^2}{2.g} + z_2 + H_E + h_{L(1-2)}$$

Where z is the height of the turbine to the tailrace, $h_{L(1\text{-}2)}$ is the friction losses inside the turbine runner.

then

$$W_{1\to2} = H_{E1\to2} = H_{net} - z_2 - \frac{C_2^2}{2.g} - h_{L(1-2)}$$

4.10.2 Work produced with draft tube

If using a draft tube, a similar relation can be obtained between inlet of runner 1 and exit of the draft tube 3.

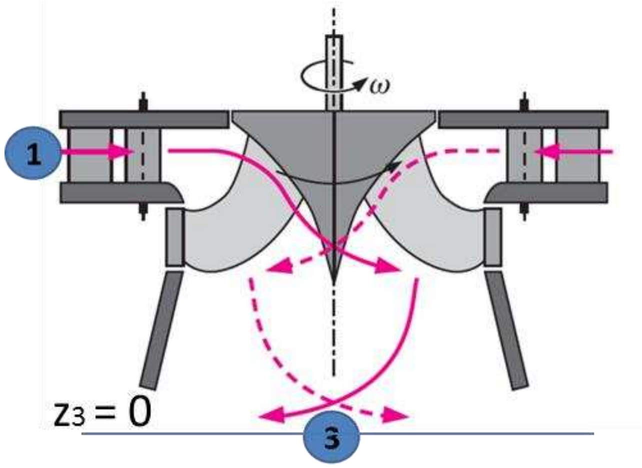


Figure 4. 11 work produced with draft tube

$$H_{net} = \frac{P_1}{\rho.g} + \frac{C_1^2}{2.g} + z_1 = \frac{\cancel{P_{atm}}}{\cancel{\rho.g}} + \frac{C_3^2}{2.g} + \cancel{z_3} + H_E + h_{L(1-3)}$$

Where in this case z = 0 since the turbine discharge is from the draft tube which is at the datum level.

Thus work produced is equal to

$$W_{1\to3} = H_{E1\to3} = H_{net} - \frac{C_3^2}{2.g} - h_{L(1-3)}$$

The difference in the work produced in the two cases:

$$\Delta W = z + \frac{(C_2^2 - C_3^2)}{2.g} - h_{L(2-3)}$$

This represents the saved energy by using the draft tube.

Applying the energy equation between inlet and outlet of the draft tube:

$$\frac{P_2}{\rho.g}+\frac{C_2^2}{2.g}+z_2=\frac{P_{atm}}{\rho.g}+\frac{C_3^2}{2.g}+z_3+h_{L(2-3)}$$

Height of the Draft Tube

$$z=z_2-z_3$$

$$\frac{P_2}{\rho.g}(\text{vacuum})=-\left[z+\frac{(C_2^2-C_3^2)}{2.g}-h_{L(2-3)}\right]$$

This means that a large negative head is created at the inlet diameter.

The efficiency of the draft tube may be defined as:

$$\eta_{D.T}=\frac{\left[\frac{(C_2^2-C_3^2)}{2.g}-h_{L(2-3)}\right]}{C_2^2/2.g}=\frac{C_2^2-C_3^2-(2.g.h_{L(2-3)})}{C_2^2}$$

C_3 must be smaller than C_2 to obtain a high efficiency.

So, pressure at runner exit in terms of draft tube efficiency:

$$\frac{P_2}{\rho.g}(\text{vacuum})=-z_2-\eta_{DT}\times\frac{c_2^2}{2g}$$

Small divergence angles require long diffusers to achieve the area necessary to reduce the exit velocity. Long diffusers increase constriction costs, therefore the angle in some cases may be increased up to about 15° from the typical optimum value of 7°. In addition to the increased loss through a large angle diffuser, flow separation can lead to unstable flow. Flow instability is to be avoided, as it has an adverse effect on turbine performance.

For some type of turbine installations, such as a vertical axis- turbine, the flow must be turned through a 90° after leaving the turbine. This is accomplished by adding an elbow between the turbine and the draft tube performance and requires careful design, figure (4.12).

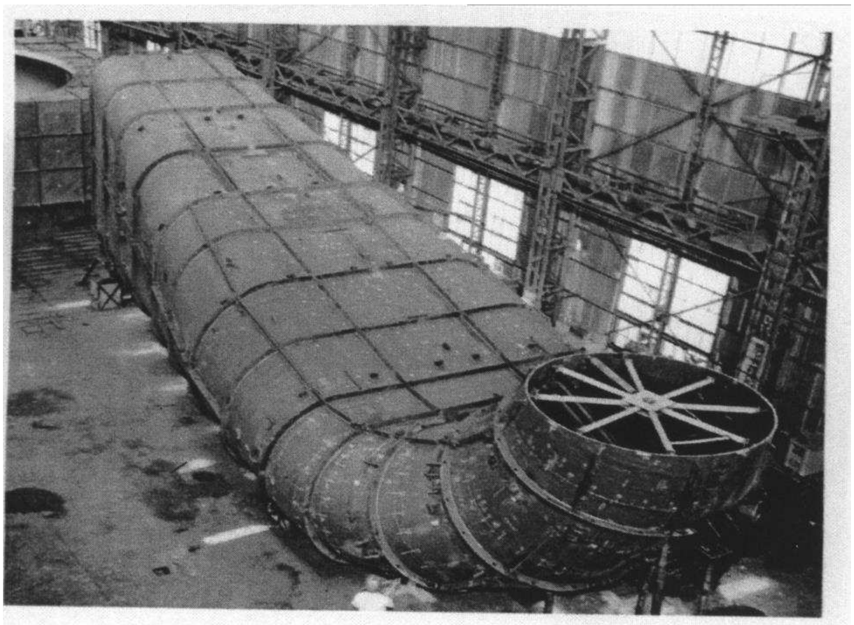


Figure 4. 12 draft tube

4.11 Cavitation

For steady flow Bernoulli's equation shows that if the velocity along a streamline increases then the pressure must decrease. In a liquid, however, the pressure cannot fall below the vapor pressure at the temperature concerned. If at any point the vapor pressure is reached, the liquid boils and small bubbles of vapor form in large numbers.

These bubbles are carried along by the flow and on reaching a point where the pressure is higher they suddenly collapse as the vapor condenses to liquid again. A cavity results and the surrounding fluid rushes in to fill it. The liquid moving from all directions collides at the center of the cavity thus giving rise to very high pressures (up to 1 GPa).

Any solid surface in the vicinity is also subjected to those intense pressures, because even if the cavities are not actually at the solid surface, the pressures are propagated from the cavities by pressure waves similar to those encountered in water hammer. This alternate formation and collapse of vapor bubbles may be repeated with a frequency of many thousands times of a second. The intense pressures, even though acting for only a very brief time over a tiny area, can cause severe damage to the surface.

The material ultimately fails by fatigue, aided perhaps by corrosion, and so the surface becomes badly scored and pitted, parts of the surface may even be torn away completely

Not only is cavitation destructive, it may also disturb the flow that the efficiency of the machine is impaired. Everything possible therefore should be done to eliminate cavitation in fluid machinery, that is, to ensure that at every point the pressure of the fluid is above the vapor pressure. When the liquid has air in solution this is released as the pressure falls and so air cavitation also occurs. Although air cavitation is less damaging than vapor cavitation to surfaces, it has a similar effect on the efficiency of the machine.

Since cavitation begins when the pressure reaches to a low value, it is likely to occur at points where the velocity or the elevation is combined. In a reaction turbine the point of minimum pressure is usually at the outlet end of a runner blade, on the leading side.

Due to the shape of pressure distribution around blade profile, there is a point on the rotor blade near to the exit where negative pressure is higher than that at exit of the rotor P2. Denoting this extra pressure by the term "dynamic depression Pd",

$$\frac{P_d}{\rho.g} = -\lambda \frac{w_2^2}{2g}$$

Where

λ is the coefficient of local dynamic depression

W_2 is relative velocity at runner exit

Maximum negative pressure in the whole system

$$\frac{P_{\min}}{\rho.g} = \frac{P_2}{\rho g} + \frac{P_d}{\rho g}$$

where

$$\frac{P_2}{\rho.g} = -z_2 - \eta_{DT} \times \frac{c_2^2}{2g}$$

$$\frac{P_{\min}}{\rho.g}(\text{vacuum}) = -\left[\eta_{DT} \times \frac{c_2^2}{2g} + z + \lambda \frac{w_2^2}{2g}\right]$$

In order to avoid cavitation the maximum negative pressure must be lesss than $P_{atm} - P_{vp} = P_{vp}$ (vacuum) as shown in figure (4.13).

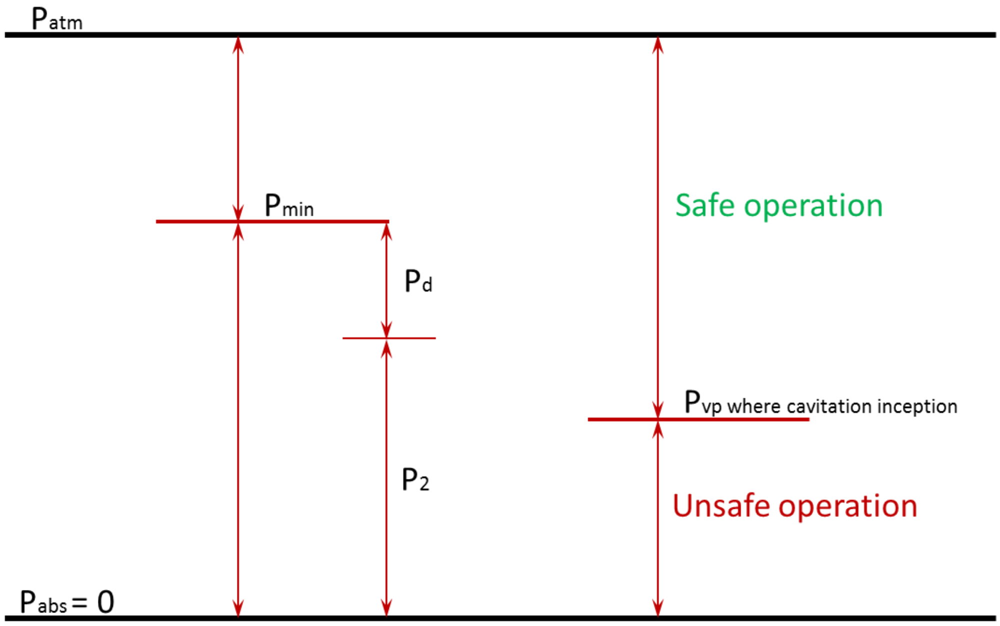


Figure 4. 13 minimum and vapor pressure for cavitation inception

4.11.1 Thoma’s cavitation factor

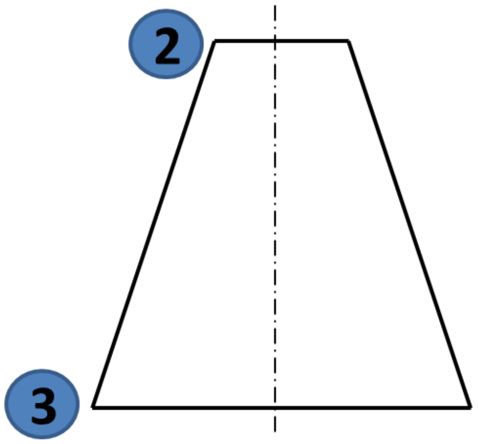


Figure 4. 14 draft tube

$$\frac{P_{\min}}{\gamma}+\frac{C_2^2}{2.g}+z_2=\cancel{\frac{P_3}{\rho.g}}+\cancel{\frac{C_3^2}{2g}}+\cancel{z_3}+h_{L1\to 3}$$

Where,

$$\frac{P_3}{\gamma}=\frac{P_{atm}}{\gamma}=0$$

$$\frac{C_3^2}{2g}=neglect$$

$$\frac{P_{\min}}{\gamma}+z_2=-\left(\frac{C_2^2}{2.g}+h_{L2\to 3}\right)\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

$$\frac{P_{\min}}{\gamma}+z_2=-\left[\eta_{DT}\times\frac{c_2^2}{2g}+\lambda\frac{w_2^2}{2g}\right]\ldots\ldots\ldots\ldots(2)$$

Thoma assumed that

$$\frac{P_{\min}}{\gamma}+z_2=-\sigma\times H_{net}$$

$$\sigma=\frac{-\frac{P_{\min}}{\gamma}-z_2}{H_{net}}=\frac{\left(\frac{P_{atm}-P_{\min_{abs}}}{\gamma}\right)_{vacuum}-z_2}{H_{net}}$$

When $P_{\min}=P_v$

$$\sigma_{cr}=\frac{-\frac{P_{\min}}{\gamma}-z_2}{H_{net}}=\frac{\left(\frac{P_{atm}-P_{vp_{abs}}}{\gamma}\right)_{vacuum}-z_2}{H_{net}}$$

For safety against cavitation

$$\sigma_{cr\,site}\succ\sigma_{catalouge}$$

$$\sigma_{cr\,site}=\frac{\left(\frac{P_{atm}-P_{vp_{abs}}}{\gamma}\right)_{vacuum}-z_2}{H_{net}}\qquad\qquad\sigma_{catalouge}=\frac{\left(\frac{P_{atm}-P_{\min_{abs}}}{\gamma}\right)_{vacuum}-z_2}{H_{net}}$$

4.11.2 Net Positive Suction Head

$$z_{cr\,site}=\left(\frac{P_{atm}-P_{vp_{abs}}}{\gamma}\right)-\sigma_{cr_{site}}\times H_{net}$$

$$z_{cr\,site}=\left(\frac{P_{atm}-P_{vp_{abs}}}{\gamma}\right)-NPSH$$

$$NPSH=\sigma_{cr_{site}}\times H_{net}$$

$$NPSH = \left(\frac{P_{atm} - P_{vp_{abs}}}{\gamma} \right) - z_{cr\ site}$$

For safety against cavitation the actual installation height (Z) must be less than the calculated critical height (Z_{cr})

$$z_{actual_{site}} \prec z_{cr_{site}}$$

4.12 Other Propeller Turbines:

The classical design does not take full advantage of the geometric properties of an axial flow runner. The flow enters the scroll case in a horizontal direction, issues radially inward from the guide case, where it forms a vortex and discharges into the draft tube in a vertical direction with very little whirl component remaining. The flow must then again be turned through 90o to discharge into the tail water in a horizontal direction.

From a design point of view, this is less than desirable for many reasons. The flow field entering the runner is highly complex, and it is difficult to design the proper pitch distribution from hub to tip for minimal shock losses. There are additional losses in the elbow, and the tortuous flow path required from the inlet to the outlet requires additional civil works.

More modern designs take full advantage of the axial-flow runner, these include the tube, bulb, and the *Straflo* types illustrated in Fig. (4.27). The flow enters and exits the turbine with minor changes in direction. In tube type, an externally mounted generator is driven by a shaft, which passes through the flow passage either upstream or downstream of the runner.

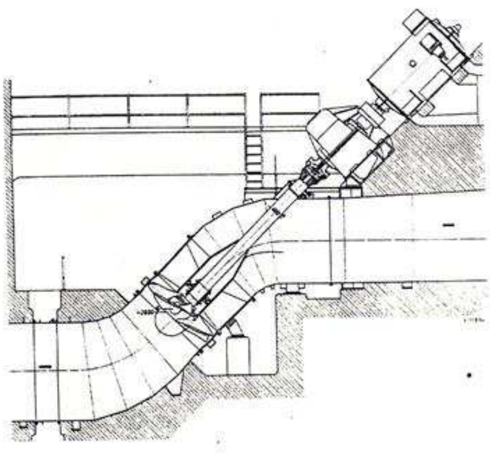


Figure 4. 15 Straflo type axial-flow runner

In bulb turbine, generator is housed within the bulb, Plate 8. Smaller units are available in which an externally mounted generator is driven by a right-angle drive which is housed within the bulb.