**CHAPTER 2: Hydraulic Turbines**

**2.1 Introduction**

Hydraulic turbines are power-producing turbomachines, using water as the fluid. The water has to be available at a reasonable height or head, in fairly large quantities so that some economically feasible power projects may be developed. As was seen in Chapter 1, the power available in the water is proportional to the product of the flow rate and the head (*P* = *wQH*). In a project site, the available flow rate depends on the rainfall in the region, the extent of the catchment area, and the possibility of storage of water (natural or built-up). The available head is a characteristic of the topography of the project site. A schematic layout of a typical hydroelectric power project is shown in Fig. 2.1.



**Figure 2.1** Schematic layout of a typical hydroelectric power project. (1) Dam; (2) reservoir fed from rainfall in catchment; (3) trash gate; (4) forebay (to take care of daily fluctuations); (5) surge tank; (6) valve house; (7) penstocks; (8) powerhouse; (9) turbines; (10) tailrace.

The type of turbine to be employed for the power project depends on the head available and the round-the-year uniform flow rate that is possible at the site of the project. This chapter aims at identifying the different types of turbines, based on the characteristics of the modern-day turbines, their suitability for a given project site, the constructional details, and the design of each type of turbine.

**2.2 Classification of Hydraulic Turbines**

Hydraulic turbines are classified based on several criteria. Some prominent classification details are shown in Table 2.1.

**Table 2.1:** Classification of hydraulic turbines



Apart from the criteria of classification, it can be seen that basically there are three types of turbines: Pelton, Francis, and Kaplan turbines, named after their designers.

**1.** The *Pelton turbine* is an impulse turbine, with tangential flow, for high-head applications.

**2.** The *Francis turbine* is a reaction turbine, with radial or mixed flow, for medium head applications.

**3.** The *Kaplan turbine* is a reaction turbine, with axial flow, for low-head applications.

**2.2.1 Selection of Hydraulic Turbines**

The power projects, where hydraulic turbines are to be installed, are generally huge projects, involving very high investments on head works and machinery. Because of the wide variation of the two basic data, namely, the head and the flow rate, each project requires some unique design. Therefore, the selection and design of a particular type of turbine must be undertaken with some discretion, so as to have the highest possible efficiency of the turbine.

There are two approaches to decide the type of turbine suitable for a given project site, with specified head and flow rate:

**1.** One criterion is the head (meters of water) available. Shown in Fig. 2.2 is a scale that indicates the head and the corresponding suitable turbine.



**Figure 2.2** Selection of turbines on the basis of head.

It may be observed that there are some ranges of overlaps, such as the 50-75 m or 150-250 m stretches. In these ranges, the turbine can be selected by the criterion of the specific speed, mentioned in the next paragraph, to include the effect of the available flow rates also.

**2.** Another criterion is the specific speed corresponding to the site data. The head, *H* (m of water), and the flow rate, *Q* (m3/s), are taken as data. An overall efficiency ηof the order of 0.85 or 0.88 can be assumed. Then, the power *P* is given by

*P* = (*w Q H* η/ 1000) (kW)

Now, the speed of the turbine *N* rpm has to be selected that must be one of the synchronous speeds (*N* = 3000/*p*, *p* = 1, 2, 3, …, to have a frequency of 50 Hz of electrical power supply) because turbines drive the alternators. With this, the specific speed can be calculated by



Figure 2.3 indicates a scale of specific speeds and the corresponding suitable turbines.



**Figure 2.3** Selection of turbines on the basis of specific speed.

It may be noted here that even on this scale, there are overlaps. Also, the limits of the specific speed to select any particular type of turbine are, however, not very sharp. There are many other considerations, such as cost factors, control factors, applications, etc., and the calculated specific speed can be logically manipulated.

If the calculated specific speed happens to be like 400, then one can assume two equal-sized turbines with the available flow rate equally divided between the two. The new specific speed for each of the turbine becomes



This new specific speed is brought in the range 60−300 to select a Francis turbine. This method of crossing over the range (from Kaplan to two or more units of Francis, or from Francis to multiple-jet Pelton) has to be evaluated with alternate plans and other possible cost factors.

Selection of any type of turbine for a site is by any of the above two approaches. However, the selection also has to focus on the further steps of the design. As an illustration, on the stretch of the Francis turbine in Fig. 2.3, another scale of the diameter ratio *D*2/*D*1 is also given. This indicates that the ratio *D*2/*D*1 varies with specific speeds. As the specific speed increases, the Francis turbine tends to be nearer to the axial flow machine.

The name “Francis turbine” therefore is not an all-time standard design. Just like the diameter ratios, all the comparative and absolute dimensions have to be determined for a given project. The guidelines start from the values of the specific speed.

It has to be mentioned here that Figs. 2.2 and 2.3 are derived from Fig. 1.13, where the different types of turbines are compared for their suitability for the situation dictated by the specific speed. The aim is to maximize the efficiency in a given set of data for a project site.



**Figure 1.13** Characterization of turbomachines by the specific speeds.

**3.3 Pelton Turbine**

The Pelton turbine belongs to the range of the low specific speeds (5 to 70) and the range of high heads (150 m of water and above). It is an impulse-type turbine.

**3.3.1 Constructional Details of Pelton Turbine**

A Pelton turbine setup is shown in Fig. 3.4. Some of the terms used are the shaft, the rotor, the nozzle, the jet, etc., which are shown in the figure. The blades or vanes of the rotor in the case of Pelton turbine are the “Pelton double cups” or “buckets,” as shown in Fig. 3.5. These double cups are mounted on the periphery of a circular disk and together they form the rotor of the Pelton turbine.



**Figure 3.4** Schematic layout of a Pelton turbine.



**Figure 3.5** Pelton double cup.

The water, supplied from the head-works to the power house through the penstocks (steel pipes), is led to these buckets in the form of a high-speed jet issued from a nozzle (Fig. 3.6). The kinetic energy of water jet is transferred to the series of the buckets (and to the rotor) that come in succession in the line of the jet as the rotor rotates. The jet gets divided into two equal halves by the jet splitter of the double cup, with each half striking the cups on either side.

The potential energy of the water at the head-works is converted into the kinetic energy of the jet of water in the nozzle, with a nozzle efficiency *η*n. This efficiency is of the order of 0.98. A concentric spear inside the nozzle controls the rate of flow of water coming out of the nozzle. The movement of the spear is controlled by a servomotor of the governor that is intended to maintain a uniform speed of the turbine.



**Figure 3.6** Nozzle of Pelton turbine.

In multi-jet Pelton turbines, water is led around the rotor into the identical nozzles equally spaced around the periphery of the rotor. The spear assemblies are also identical in all the nozzles. Their movement is controlled by the same source, so that all the jets are equally controlled.

Whenever a Pelton turbine is required to be stopped, a shut-off valve in the supply mains has to be closed. But this should not be done suddenly, as otherwise water hammer is likely to occur in the pipes. A deflector can be actuated so that the jet can be deflected from the Pelton cups, in the opposite direction, so that it can act, in the meantime, as a brake jet. This jet is in the opposite direction to the main jet, with water striking the backside of the cups. These details are shown in Fig. 3.7. A separate brake jet can also be arranged in the direction opposite to the main jet.



**Figure 3.7** Deflector assembly and brake jet.

Pelton turbines can be designed with horizontal or vertical shaft arrangements. In a horizontal setup, the turbine wheel can be between two journal bearings. The design can also be in the over-hung form, with bearings on one side and its shaft coupled to the generator shaft. In vertical installations, the supports are the thrust bearings, the turbine wheel being at the bottom level.

**2.3.2 Analysis of the Pelton Turbine**

The gross head at the project site, *H*g, is the difference of water levels between the reservoir and tailrace. The head available at the power house or nozzles is *H*. The difference (*H*g - *H* ) is due to the topography of the land and the losses in the penstocks, valves, height of nozzles above the tailrace, etc.

The velocity of the water jet at the outlet of the nozzle or inlet to the rotor cups is

 (2.1)

where *c*v is the coefficient of velocity of the nozzle, with a value of the order of 0.96-0.98. The velocity of the Pelton cups is



where *D* is the pitch diameter (m) and *N* is the rotational speed (rpm) of the rotor shaft. The centerline of the water jet is tangential to this pitch circle (Fig. 2.8). The jet interacts with the Pelton cups over a stretch of the travel, from position A to position B in Fig. 2.8. The mean position is shown in the cross-section in Fig. 2.9. The outlet is a little divergent so that the water leaves the buckets without splashing over the back of the next bucket.



**Figure 2.8** Jet striking Pelton buckets.



**Figure 2.9** Jet is split between the two cups.

The water takes a turn of about 165° from *β*1 = 0° to *β*2 = 165°. The velocity triangles corresponding to the upper bucket of Fig. 2.9 are shown in Fig. 2.10. It may be noted that the inlet velocity triangle is reduced as a straight line, as shown. When the blade friction factor is *c*b we have





**Figure 2.10** Velocity triangles for the Pelton cup at the inlet and outlet.

With reference to inlet velocity triangle of Fig. 2.10, we have

Vr1 = V1 – U (2.2 a)

And Vu1 = V1 (2.2 b)

Also, with reference to the outlet velocity triangle, we have

Vu2 = U – Vr2 cos β2  (2.3 a)

Using Eq. (2.2a) and Eq. (2.2b) in Eq. (2.3a), we get

Vu2 = U – cb x Vr1 cos β2

Vu2 = U – cb (V1 – U) cos β2 (2.3 b)

The specific work *W* is given by



 W = (U/V1) (1- U/V1) (1 + cb cos β2) V12 (2.4 a)

 W = ϕ (1 – ϕ) (1 + cb cos β2) V12 (2.4 b)

where *ϕ = U*/*V*1 is the speed ratio. For a given installation, *c*b, *β*2, and *V*1 are constants. The specific work *W* is maximized when *dW*/*dϕ* is taken as zero, resulting in *ϕ =* 0.5. Hence

Wmax = 0.25 (1 + cb cos β2) V12 (2.5)

The hydraulic efficiency *η*h = *W*/(*V*1 2/2) works out to a maximum value of

 *η*hmax = *Wmax* / (*V*1 2/2)

ηhmax = (1 + cb cos β2) / 2 (2.6)

**2.3.3 Efficiencies of Pelton turbine**

The hydraulic efficiency is obtained as in Eq. (2.6) in the form ηh = (1 + cb cos β2) / 2

This expression holds for the defining equation



where the hydraulic efficiency is defined as the ratio of the power of rotor to the power available in the inlet stream of water (*η*h *= P*r/*P*), including the effect of exit losses, fluid friction on the blade surface, etc.

One can recall the definition of volumetric efficiency *η*v as

 (2.7)

The reduction in the volume flow rate in a turbine (Δ*Q*) was earlier attributed to the leakage. In the case of Pelton turbine, this loss can also be attributed to the “ineffective” volume flow rate that is in the outer layers of water in the jet, which may not be as effective as the core of the jet in exerting the force on the buckets.

The mechanical efficiency (*η*m) has also been defined as

 (2.8)

The overall efficiency is then given by

 (2.9)

There are quite a few factors that affect the above efficiencies, individually and collectively. The surface finish of the blades, the friction factor *c*b , the sand content of minute sizes in the water jet, the speed, the smoothness of the bearings, the disk friction of the rotor, and the magnitudes of all these in relation to the total power contribute to the final value of the overall efficiency. It is common to use a value of approximately 0.85−0.88 for the overall efficiency.

**2.3.4 Design Parameters of Pelton Turbine**

Although the speed ratio, *ϕ =* 0.5 is a theoretical value, in practice, *ϕ* is taken as about 0.45 or 0.46, so that

U = (0.45 to 0.46) V1 (m/s) (2.10)

With this, the values of *U* and *N* (as assumed earlier) are known in the equation *U* = π *DN*/60, from which we can calculate

D = 60 U / π N (m) (2.11)

Now, from the specific speed, as decided by the site data, the number of jets is determined. The total available volume flow rate is equally divided between the jets. The flow rate per jet is Qj = Q / n , n = 1, 2, 3, … (2.12)

Now,

Volume flow rate in a jet = Area of jet x Velocity of jet

Qj = ( π d2 /4) V1

And therefore

 (2.13)

For the Pelton wheel, the jet diameter is an important parameter to decide the geometrical proportions of the Pelton double cups. These proportions are so fixed that almost the entire jet is usefully employed in striking the cups at the middle splitter line to generate the maximum possible torque. The notch at the tip of the bucket helps in this aspect. If the notch were not present, the cups would come in contact with the jet, starting from the tip, where a sizable quantity of the jet would disperse over and around the tip with a lot of losses as shown in Fig. 2.11.



**Figure 2.11** Losses in the jet when buckets do not have notches.



**Figure 2.12** Proportions of Pelton cups.

The geometrical features of the Pelton double cup are shown in Fig. 2.12. These features are the length (*L*), breadth (*B*), and depth (*T* ) of the double cup. These parameters are optimized in terms of the jet diameter *d*, and are specified by the following equations:

*L*= 2.3*d* to 2.8*d* (2.14)

*B*= 2.8*d* to 3.2*d* (2.15)

*T*= 0.6*d* to 0.9*d* (2.16)

The width of the notch, *N*, in Fig. 2.12 is approximated as 2-5 (mm) more than the jet diameter *d*. The number of buckets is also optimized and is given as

 (2.17)

**2.4 Francis Turbine**

The Francis turbine is a reaction turbine suitable for a medium range of specific speeds (60−300) and a medium range of heads (50−250 m). The Francis turbine is designed as a radial flow machine in the range of specific speeds of 60−120. As the specific speed corresponding to the data of the project site keeps on increasing, the design shifts to a mixed flow machine and then to almost an axial flow machine.

**2.4.1 Constructional Features of Francis Turbine**

An installation of the Francis turbine is shown in Fig. 2.13. Water from the penstock pipe enters an outer spiral casing that may be fabricated out of steel plates or cast in concrete with a lining of steel plates. This casing is arranged around a ring of guide vanes, and its area of cross-section diminishes progressively giving a uniform distribution of water at a continuous outlet, inward toward a number of guide vanes, around the rotor.



**Figure 2.13** Schematic layout of a Francis turbine.

The guide vanes have airfoil shapes so that the passages between them act like nozzles that convert a part of the pressure energy of water into kinetic energy. The water coming from the casing is directed on to the rotor vanes. Each guide vane has its own axis about which it can swing, so as to vary the area of flow of water. The swinging of all the guide vanes (about their individual axes) is controlled by a governor-actuated regulator ring, so that the flow of water can be controlled. The details are shown in Fig. 2.14.



**Figure 2.14** Control of guide vanes.

This control is only to the extent of maintaining constant speed over the fluctuation of loads over the turbine. The water from the guide vanes enters the rotor, with both kinetic energy and pressure energy. The rotor vanes absorb these energies and the water is discharged to a component known as a draft tube (Fig. 2.13).

As a reaction turbine installation, the casing, guide-vane ring, and rotor of the Francis turbine run full without the water coming in contact with the atmosphere. At the exit of the rotor, that is, in the draft tube also, the water is not open to atmosphere.

The draft tube is a slightly divergent tube, connecting the outlet of the runner to the tailrace level. The draft tube also runs full. The water column in the draft tube is under sub-atmospheric pressure and effectively saves the head that otherwise would have been lost, when the turbine is installed at a higher level than the tailrace. Because of the divergent portion, a part of the exit kinetic energy is also recovered.

Francis turbine installations can be designed either with horizontal shafts or vertical shafts. With horizontal shafts, the draft tubes have to be provided with a bend that reduces the effectiveness of the draft tube in recovering the lost head. The vertical shaft installations have the turbine rotor at the lowest level so that the axial discharge from the runner becomes vertically downward. This becomes a very good feature of the vertical draft tube, with better efficiencies.

Francis turbines with huge capacities are usually designed with vertical shafts. Some dams are built primarily for the purpose of irrigation. The water level in the dams may reach such levels that it is possible to use that head while letting the water out to the canals through the turbines in power houses at the bottom of the dams. Generally, Francis turbines become the most suitable choices for such “dam power houses.”

**2.4.2 Analysis of the Francis Turbine**

Velocity triangles were discussed in detail in Chapter 3 with an understanding that the planes of the triangles were either perpendicular to the axis (for radial flow machines) or parallel to the axis (for axial flow machines). In continuation with the same discussion, for the mixed flow pattern of the Francis turbine, the plane of the inlet velocity triangle is perpendicular to the axis (radially inward flow), but the outlet velocity triangle is on a plane that shifts its orientation (from perpendicular to) parallel to the axis (axial discharge flow). This agrees well with the expression for the specific work (*W*) because the components of velocities responsible for the energy transfer are the whirl components, *V*u1 and *V*u2, whether in a radial flow pattern or an axial flow pattern. With this in mind, a typical set of velocity triangles is shown in Fig. 2.15. The orientation of the rotor blades is also shown in Fig. 2.15.

The range of specific speeds corresponding to the design of Francis turbine is stated as 60−300, and this is a wide range. The details shown in Fig. 2.15 apply to the lower stretch of the specific speeds 60−120 in this wide range. As the specific speed increases, the shapes of blades of the runner also change. The changing phases of the blades and the applicable velocity triangles are shown in Fig. 2.16, to cover the whole range of the specific speeds. Figure 2.16 may be considered as continuation of Fig. 2.15.

The velocity triangles at the outlet of the blades in all the cases of Fig. 2.16 have *α*2=90°, namely axial discharge, as in Fig. 2.15. In all the cases, *V*u2 = 0. This results in the specific work *W* being equal to *U*1 *V*u1. The input to the rotor is in both kinetic and potential forms of energies, adding up to a total of *gH*.

Therefore, the hydraulic efficiency is given by

$η\_{h}= \frac{U\_{1}V\_{u1}}{g H}$ (2.18)



**Figure 2.15** Francis runner with velocity triangles.



**Figure 2.16** Francis runner with velocity triangles. Runner and velocity triangles for specific speed (a) *Ns* = 120-180 and (b) *Ns* = 180-300.

**2.4.3 Efficiencies of Francis turbine**

The hydraulic efficiency of the Francis turbine is obtained in the form

$η\_{h}= \frac{U\_{1}V\_{u1}}{g H}$

which is same as Eq. (2.18). The expressions for volumetric efficiency [Eq. (2.7)], mechanical efficiency [Eq. (2.8)] and overall efficiency [Eq. (2.9)], which were detailed for the Pelton turbine in Section 2.3.2, are very much valid for Francis turbine also, because, these expressions are independent of the (a) geometrical features and the (b) mechanism of energy transfer of these turbines. Also, the reasons for the various losses in the turbines are largely of same nature. The loss due to the leakage in the clearances in Francis turbine is one difference that exists between Pelton and Francis turbines. The numerical values of the efficiencies also approximately match between the machines.

**2.4.4 Design Parameters of Francis Turbine**

The design parameters of Francis turbine are discussed in the following sub-sections.

***2.4.4.1 Volute or Spiral Casing***

The casing is the outer conduit of the Francis turbine assembly. The penstock pipe is connected to the inlet of this casing (through a stop valve). The cross-section at the inlet is circular with the diameter equal to that of the penstock pipe. The cross-sectional area of the casing has to gradually decrease. At the same time, the height of the inlet to the guide vanes, from the inner side of the casing, has to remain invariant. Hence, the cross section gets into oval shape, gradually reducing the area to almost zero, when it reaches back through 360°.

At the inlet, the area *A* = (*πd*2)/4. At any angle *θ* (refer Fig. 2.17), we have

$$A\_{0}= \frac{π d^{2}}{4} x \frac{\left(360- θ\right)}{360°}$$

where *θ* is measured at the central axis, starting from the line of commencing the decrease of area. The overall purpose is to reduce the eddies and to effect a smooth and uniform distribution of water all around the rotor.



**Figure 2.17** Spiral casing with decreasing area (*B* = *B*o + 5 mm).

***2.4.4.2 Guide Vanes***

As explained earlier, the guide vanes swing about their own individual axes. In their “full open” position, let their tail ends be on a diameter *D*0. For the purpose of the design procedure, the following two assumptions are made:

**1.** The guide vanes also rotate about the axis of the shaft, like the runner does, at the speed of the runner, *N* rpm.

**2.** There is a reference velocity,$\sqrt{2 gH }$, of the water if the entire head energy were to be converted into kinetic energy without loss.

(Both the above are only assumptions; neither the guide vanes rotate around the runner nor the potential energy of water is totally converted into velocity in the Francis turbine.) With these assumptions, the hypothetical tangential velocity of the guide-vane tip is

$$U\_{0}= \frac{π D\_{0 }N}{60}$$

Also $U\_{0}= ϕ\_{0}\sqrt{2 g H}$ (2.19)

So $D\_{0}= \frac{60 ϕ\_{0 } \sqrt{2 g H}}{π N}$ (2.20)

Here *ϕ*0 is a hypothetical “speed ratio” or a simple coefficient, varying between 0.7 at *N*s= 60 and 1.31 at *N*s= 300. The length of the guide vanes, *L*0, is taken as

 $L\_{0}=0.3 D\_{0}$ (2.21)

The height of the guide vanes, *B*0, is calculated from the equation of flow rate:

Volume flow rate (m3/s ) = Flow area around the rotor (m2) x Flow velocity (m/s)

$$=>Q= π D\_{0} B\_{0} V\_{f0}= π D\_{0} B\_{0} ψ\_{0} \sqrt{2 g H}$$

So $B\_{0}= \frac{Q}{π D\_{0} ψ\_{0 }\sqrt{2 g H}}$ (2.22)

where 𝛙0 is the flow coefficient, varying from 0.15 at *N*s = 60 to 0.35 at *N*s = 300.

The number of guide vanes is decided as a thumb rule. Too few numbers do not serve the purpose of uniform distribution of water. Too many numbers may give rise to some loss, with added costs. Depending on the diameter *D*0, as determined above, the number of guide vanes varies from 8 to 24 (8, 10, 12, 14, 16, 20, 24) as *D*0 varies from 20 cm to 2 m. (Numbers chosen are even to facilitate the shipping of the component parts in segments.)

***2.4.4.3 Rotor***

The rotor diameter at the inlet (namely, the outer diameter) and the tangential velocity of the blades at the inlet are related, as usual, by the relation

$$U\_{1}= \frac{π D\_{1 }N}{60}$$

The velocity *U*1 is taken as $U\_{1}= ϕ\_{1}\sqrt{2 g H}$ (2.23)

where *ϕ*1 is the speed ratio with respect to the reference velocity $\sqrt{2 g H}$. This speed ratio has values ranging from 0.62 to 0.82 over the range of specific speeds 60−300. Equating both the expressions for *U*1 we get

 $D\_{1}= \frac{60 ϕ\_{1 } \sqrt{2 g H}}{π N}$ (2.24)

The outlet diameter *D*2 of the runner (inner diameter) is calculated from the ratio *D*2/*D*1, varying from 0.5 to 1, with the specific speed range of 60−300. The height *B*1 of the blades at the inlet is taken as equal to the height of the guide vanes *B*0. The number of blades on the runner is generally within ±1 of that of the guide vanes. The blade angles of the runner are also chosen over a range of values, depending on the specific speed. These values are indicated in Figs. 2.15 and 2.16. It may be noted that the values of *α*1 and *β*1 shown in Figs. 2.15 and 2.16 are the values at full load on the turbine.