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HYDRAULIC TURBINES

List of the various advantages and disadvantages of hydropower plants

Table 9.2 Features of Hydroelectric Power Plants

Advantages	Disadvantages
Technology is relatively simple and proven. High efficiency. Long useful life. No thermal phenomena apart from those in bearings and generator.	Number of favourable sites limited and available only in some countries. Problems with cavitation and water hammer.
Small operating, maintenance, and replacement costs.	High initial cost especially for low head plants compared with thermal power plants.
No air pollution. No thermal pollution of water.	Inundation of the reservoirs and displacement of the population. Loss of arable land. Facilitates sedimentation upstream and erosion downstream of a barrage.

9.2 HYDRAULIC TURBINES

The efficiency of a hydraulic turbine can be defined as the work developed by the rotor in unit time divided by the difference in hydraulic energy between inlet and outlet of the turbine in unit time.

The efficiencies of the three principal types of hydraulic turbine just mentioned are shown in Figure as functions of the power specific speed

$$\Omega_{sp} = \frac{\Omega\sqrt{P/\rho}}{(gH_E)^{\frac{5}{4}}},$$

where P is the power delivered by the shaft, ρ is the density of water, HE is the effective head at turbine entry, and Ω is the rotational speed in radians per second.

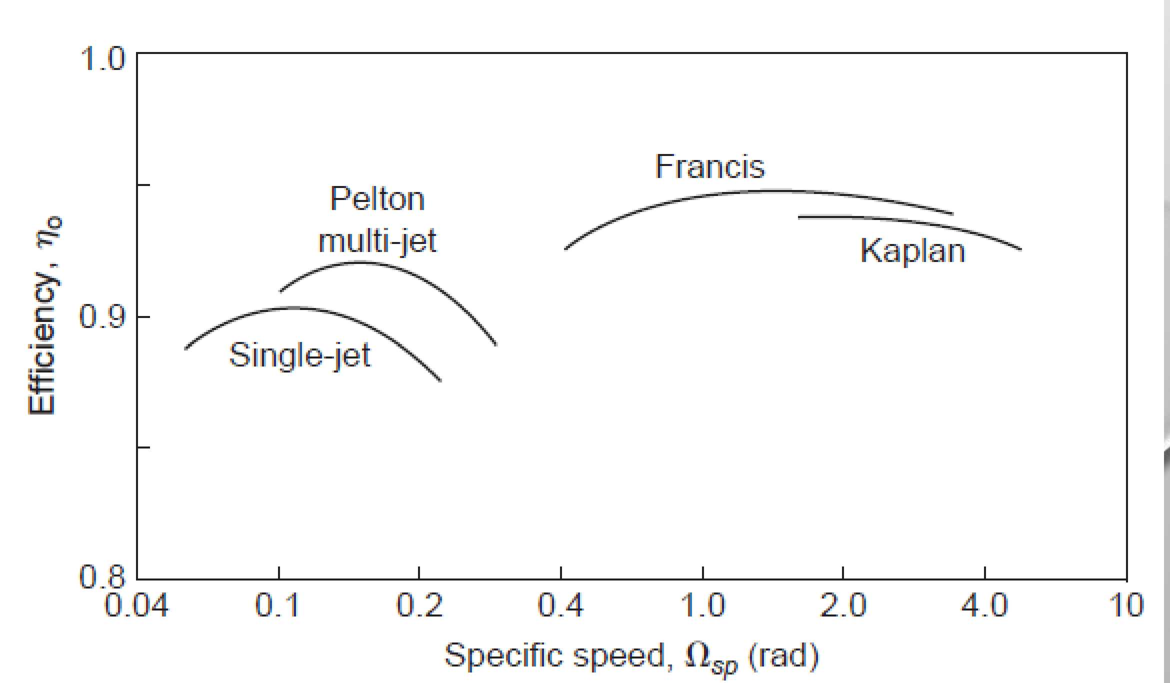


Table 9.3 summarizes the normal operating ranges for the specific speed, the effective head, the maximum power and best efficiency for each type of turbine.

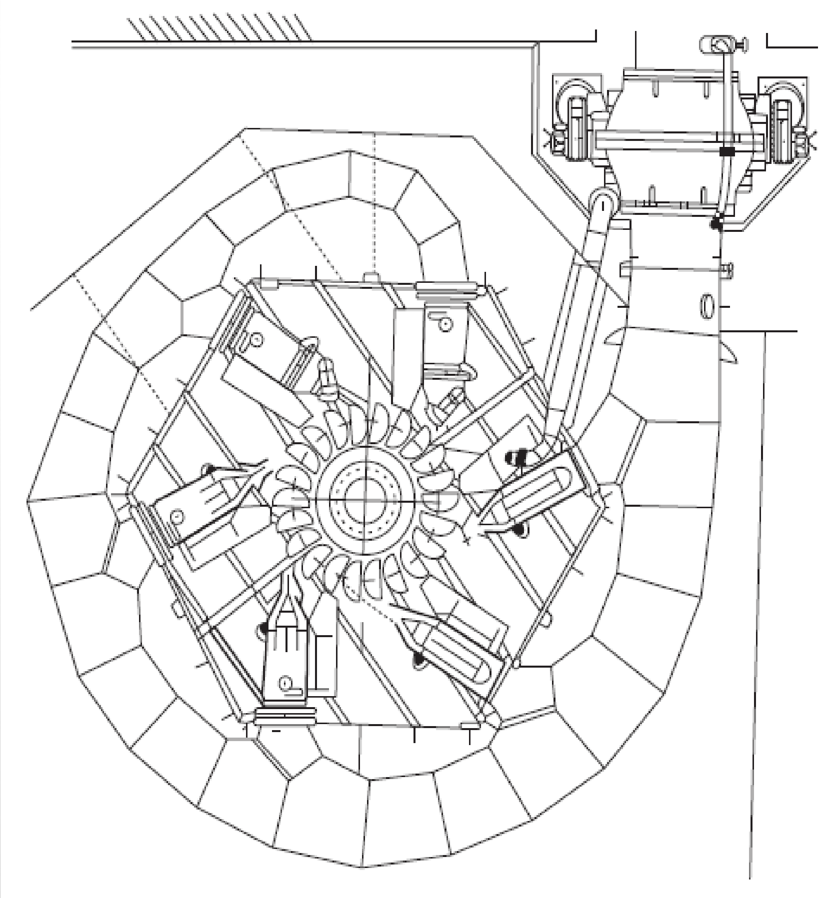
Table 9.3 Operating Ranges of Hydraulic Turbines

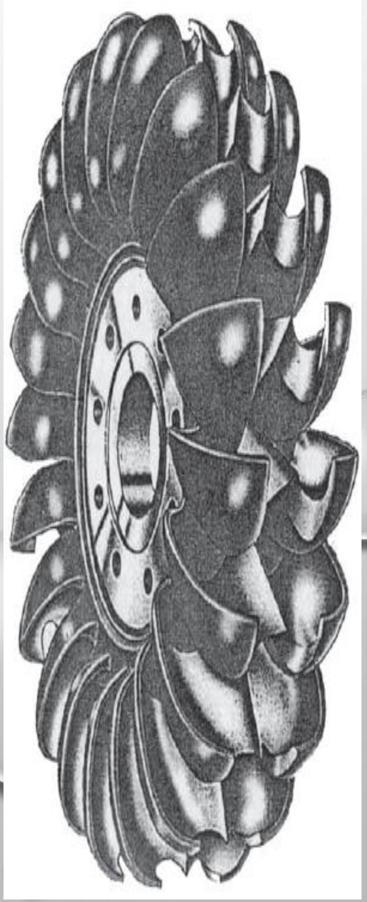
	Pelton Turbine	Francis Turbine	Kaplan Turbine
Specific speed (rad)	0.05–0.4	0.4–2.2	1.8–5.0
Head (m)	100–1770	20–900	6–70
Maximum power (MW)	500	800	300
Optimum efficiency (%)	90	95	94
Regulation method	Needle valve and deflector plate	Stagger angle of guide vanes	Stagger angle of rotor blades

Note: Values shown in the table are only a rough guide and are subject to change.

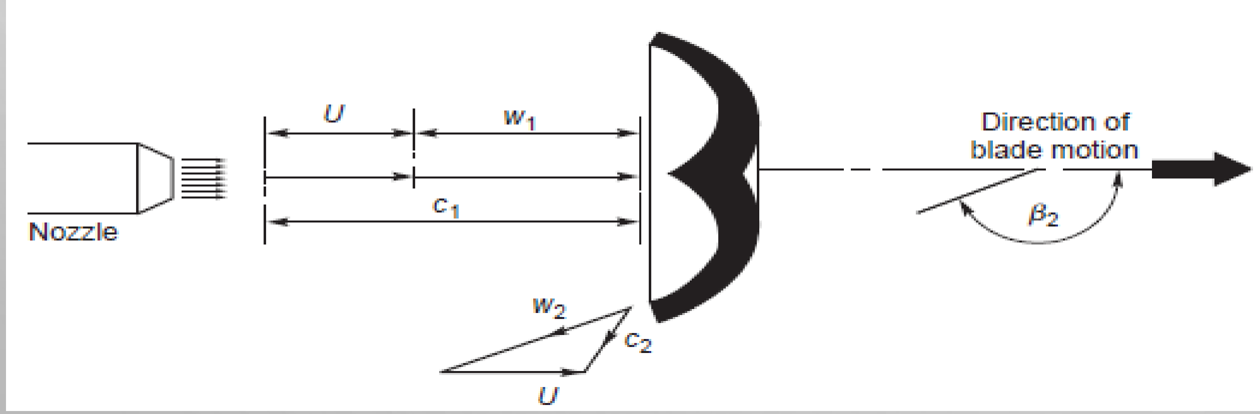
9.3 THE PELTON TURBINE

This is the only hydraulic turbine of the impulse type now in common use. It is an efficient machine and it is particularly suited to high head applications. The rotor consists of a circular disc with a number of blades (usually called buckets) spaced around the periphery. One or more nozzles are mounted in such a way that each nozzle directs its jet along a tangent to the circle through the centers of the buckets. A “splitter” or ridge splits the oncoming jet into two equal streams so that, after flowing round the inner surface of the bucket, the two streams depart from the bucket in a direction nearly opposite to that of the incoming jet. Figure 9.3 shows the runner of a Pelton turbine and Figure 9.4 shows a six-jet vertical axis Pelton turbine.





Considering one jet impinging on a bucket, the appropriate velocity diagram is shown in Figure 9.5. The jet velocity at entry is c_1 and the blade speed is U so that the relative velocity at entry is $w_1 = c_1 - U$. At exit from the bucket one half of the jet stream flows as shown in the velocity diagram, leaving with a relative velocity w_2 and at an angle β_2 to the original direction of flow. From the velocity diagram the much smaller absolute exit velocity c_2 can be determined. From Euler's turbine equation, eqn. (1.18c), the specific work done by the water is



$$\Delta W = U_1 c_{\theta 1} - U_2 c_{\theta 2}.$$

For the Pelton turbine, $U_1 = U_2 = U$, $c_{\theta 1} = c_1$ so we get

$$\Delta W = U[U + w_1 - (U + w_2 \cos\beta_2)] = U(w_1 - w_2 \cos\beta_2),$$

in which the value of $c_{\theta 2} < 0$, as defined in Figure 9.5, i.e., $c_{\theta 2} = U + w_2 \cos\beta_2$.

The effect of friction on the fluid flowing inside the bucket will cause the relative velocity at outlet to be less than the value at inlet. Writing $w_2 = kw_1$, where $k < 1$,

$$\Delta W = Uw_1(1 - k\cos\beta_2) = U(c_1 - U)(1 - k\cos\beta_2). \tag{9.2}$$

An efficiency η_R for the runner can be defined as the specific work done ΔW divided by the incoming kinetic energy, i.e.,

$$\eta_R = \Delta W \Big/ \left(\frac{1}{2}c_1^2\right) = 2U(c_1 - U)(1 - k\cos\beta_2)/c_1^2. \tag{9.3}$$

Therefore,

$$\eta_R = 2\nu(1 - \nu)(1 - k\cos\beta_2), \tag{9.4}$$

where the blade speed to jet speed ratio, $\nu = U/c_1$.

To find the optimum efficiency, differentiate eqn. (9.4) with respect to the blade speed ratio, i.e.,

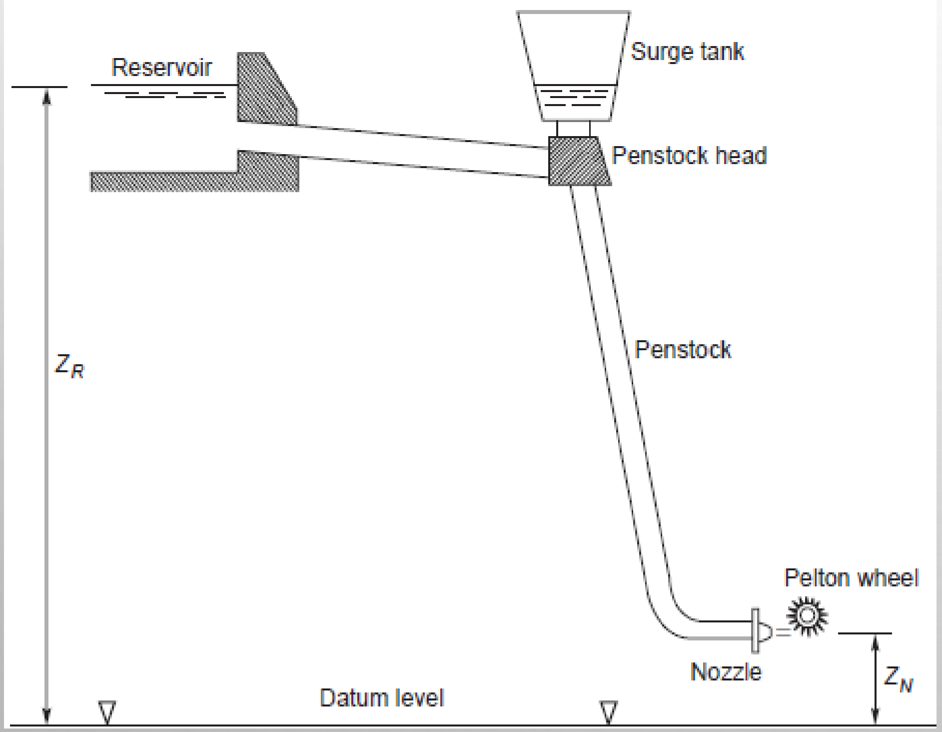
$$\frac{d\eta_R}{d\nu} = 2\frac{d}{d\nu}(\nu - \nu^2)(1 - k\cos\beta_2) = 2(1 - 2\nu)(1 - k\cos\beta_2) = 0.$$

Therefore, the maximum efficiency of the runner occurs when $\nu = 0.5$, i.e., $U = c_1/2$. Hence,

$$\eta_{R\,\max} = (1 - k\cos\beta_2). \tag{9.5}$$

A Simple Hydroelectric Scheme

The layout of a Pelton turbine hydroelectric scheme is shown in Figure 9.7. The water is delivered from a constant level reservoir at an elevation z_R (above sea level) and flows via a pressure tunnel to the penstock head, down the penstock to the turbine nozzles emerging onto the buckets as a high speed jet. To reduce the deleterious effects of large pressure surges, a *surge tank* is connected to the flow close to the penstock head, which acts so as to damp out transients. The elevation of the nozzles is z_N and the gross head, $H_G = z_R - z_N$.



Sizing the Penstock

It is shown in elementary textbooks on fluid mechanics, e.g., Shames (1992) and Douglas, Gasiorek, and Swaffield (1995), that the loss in head with incompressible, steady, turbulent flow in pipes of circular cross-section is given by Darcy's equation:

$$H_f = \frac{2flV^2}{gd}, \qquad (9.6)$$

where f is the friction factor, l is the length of the pipe, d is the pipe diameter, and V is the mass average velocity of the flow in the pipe. It is assumed, of course, that the pipe is running full. The value of the friction factor has been determined for various conditions of flow and pipe surface roughness and the results are usually presented in what is called a *Moody diagram*. This diagram gives values of f as a function of pipe Reynolds number for varying levels of relative roughness of the pipe wall.

The penstock (the pipeline bringing the water to the turbine) is long and of large diameter and this can add significantly to the total cost of a hydroelectric power scheme. Using Darcy's equation it is easy to calculate a suitable pipe diameter for such a scheme if the friction factor is known and an estimate can be made of the allowable head loss. Logically, this head loss would be determined on the basis of the cost of materials etc., needed for a large diameter pipe and compared with the value of the useful energy lost from having too small a pipe. A commonly used compromise for the loss in head in the supply pipes is to allow $H_f \leq 0.1\ H_G$.

From eqn. (9.6), substituting for the velocity, $V = 4Q/(\pi d^2)$, we get

$$H_f = \left(\frac{32fl}{\pi^2 g}\right)\frac{Q^2}{d^5}. \qquad (9.7)$$

Energy Losses in the Pelton Turbine

The effective head, H_E (or delivered head), at entry to the turbine is the gross head minus the friction head loss, H_f,

$$H_E = H_G - H_f = z_R - z_N - H_f$$

and the spouting (or ideal) velocity, c_o, is

$$c_o = \sqrt{2gH_E}.$$

The pipeline friction loss H_f is regarded as an external loss and is not usually included in the losses attributed to the turbine itself. The performance and efficiency of the turbine are, in effect, measured against the total head, H_E, as shown in the following.

The main energy losses of the turbine occur in

(i) the nozzles due to fluid friction;
(ii) converting the kinetic energy of the jet into mechanical energy of the runner;
(iii) external effects (bearing friction and windage).

Each of these energy losses are now considered in turn.

For *item* (*i*) let the loss in head in the nozzles be ΔH_N. Thus, the available head is

$$H_E - \Delta H_N = c_1^2/(2g), \tag{9.8}$$

where c_1 is the actual velocity of the jet at nozzle exit. The nozzle efficiency is defined by

$$\eta_N = \frac{\text{energy at nozzle exit}}{\text{energy at nozzle inlet}} = \frac{c_1^2}{2\,gH_E}. \tag{9.9a}$$

This efficiency is usually very close to 100% as the flow is accelerating through the nozzle. An often-used alternative to η_N is the nozzle velocity coefficient K_N defined by

$$K_N = \frac{\text{actual velocity at nozzle exit}}{\text{spouting velocity}} = \frac{c_1}{c_0},$$

i.e.,

$$\eta_N = K_N^2 = \frac{c_1^2}{c_0^2}. \tag{9.9b}$$

For *item* (*ii*) the loss in energy is already described in eqn. (9.2) and the runner efficiency η_R by eqns. (9.3) and (9.4). The turbine hydraulic efficiency η_h is defined as the specific work done by the rotor, ΔW, divided by the specific energy available at entry to the nozzle, gH_E, i.e.,

$$\eta_h = \frac{\Delta W}{gH_E} = \left(\frac{\Delta W}{\frac{1}{2}c_1^2}\right)\left(\frac{\frac{1}{2}c_1^2}{gH_E}\right) = \eta_R\eta_N, \tag{9.10}$$

after using eqn. (9.9a).

For *item (iii)* the external losses are responsible for the energy deficit between the runner and the shaft. A good estimate of these losses can be made using the following simple flow model where the specific energy loss is assumed to be proportional to the square of the blade speed, i.e.,

$$\text{external loss/unit mass flow} = KU^2,$$

where K is a dimensionless constant of proportionality. Thus, the shaft work done/unit mass flow is

$$\Delta W - KU^2.$$

Therefore, the overall efficiency of the turbine, η_o, including these external losses is

$$\eta_o = (\Delta W - KU^2)/(gH_e),$$

i.e., the shaft work delivered by the turbine/specific energy available at nozzle entry, which

$$= \eta_R \eta_N - 2K\left(\frac{U}{c_1}\right)^2\left(\frac{c_1^2}{2\,gH_E}\right).$$

Using the definitions of the blade speed–jet speed ratio, $\nu = U/c_1$, and the nozzle efficiency, $\eta_N = c_1^2/c_2^2$,

$$\eta_0 = \eta_N(\eta_R - 2K\nu^2) = \eta_m \eta_R \eta_N, \tag{9.11}$$

where the mechanical efficiency, $\eta_m = 1 -$ external losses/gH_E, i.e.,

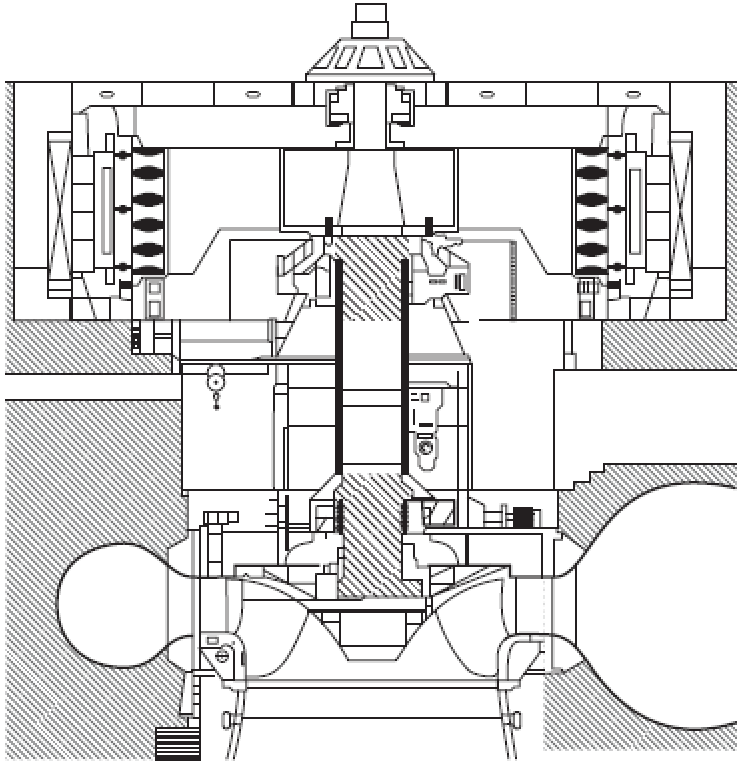
$$\eta_m = 1 - 2K\nu^2/\eta_R. \tag{9.12}$$

9.4 REACTION TURBINES

The primary features of the reaction turbine are

(i) only part of the overall pressure drop has occurred up to turbine entry, the remaining pressure drop takes place in the turbine itself;
(ii) the flow completely fills all of the passages in the runner, unlike the Pelton turbine where, for each jet, only one or two of the buckets at a time are in contact with the water;
(iii) pivotable guide vanes are used to control and direct the flow;
(iv) a draft tube is normally added on to the turbine exit; this is considered as an integral part of the turbine.

The pressure of the water gradually decreases as it flows through the runner and the reaction from this pressure change earns this type of turbine its appellation.



9.5 THE FRANCIS TURBINE

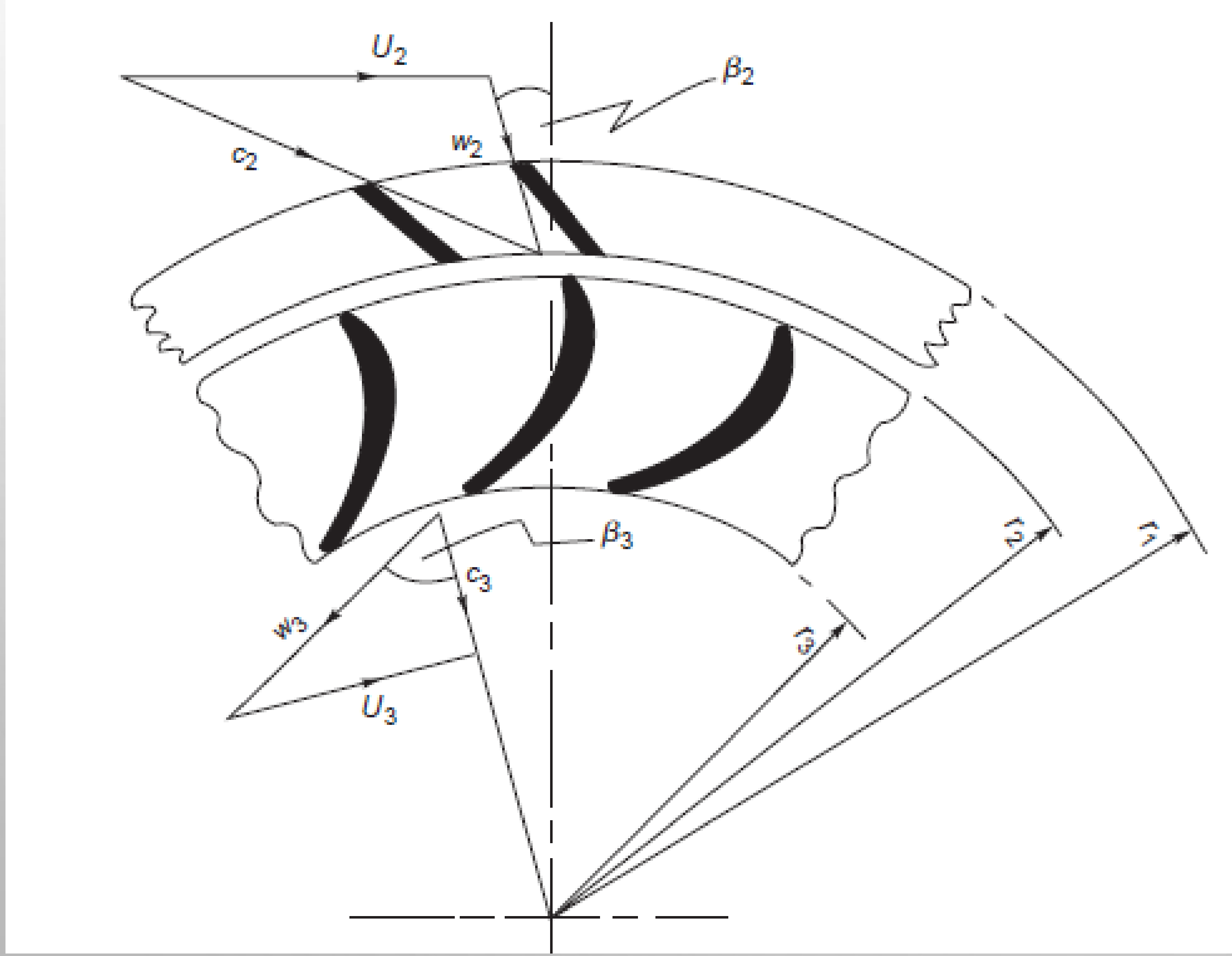
The majority of Francis turbines are arranged so that the axis is vertical (some smaller machines can have horizontal axes). Figure 9.11 illustrates a section through a vertical shaft Francis turbine with a runner diameter of 5 m, a head of 110 m, and a power rating of nearly 200 MW. Water enters via a spiral casing called a *volute* or *scroll* that surrounds the runner. The area of cross-section of the volute decreases along the flow path in such a way that the flow velocity remains constant. From the volute the flow enters a ring of stationary guide vanes, which direct it onto the runner at the most appropriate angle.

Figure 9.12 shows the runner of a small Francis turbine and Figure 9.13 is a sectional view of the turbine together with the velocity triangles at inlet to and exit from the runner at mid-blade height. At inlet to the guide vanes the flow is in the radial/tangential plane, the absolute velocity is c_1 and the absolute flow angle is α_1. Thus,

$$\alpha_1 = \tan^{-1}(c_{\theta 1}/c_{r1}). \tag{9.13}$$

The flow is turned to angle α_2 and velocity c_2, the absolute condition of the flow at entry to the runner. By vector subtraction the relative velocity at entry to the runner is found, i.e., $\boldsymbol{w}_2 = \boldsymbol{c}_2 - \boldsymbol{U}_2$. The relative flow angle β_2 at inlet to the runner is defined as

$$\beta_2 = \tan^{-1}[(c_{\theta 2} - U_2)/c_{r2}]. \tag{9.14}$$


U_2
β_2
c_2
w_2
β_3
c_3
w_3
U_3
r_1
r_2
r_3

Further inspection of the velocity diagrams in Figure 9.13 reveals that the direction of the velocity vectors approaching both guide vanes and runner blades are tangential to the camber lines at the leading edge of each row. This is the ideal flow condition for "shockless" low loss entry, although an incidence of a few degrees may be beneficial to output without a significant extra loss penalty. At vane outlet some deviation from the blade outlet angle is to be expected (see Chapter 3). For these reasons, in all problems concerning the direction of flow, it is clear that the angle of the fluid flow is important and not the vane angle as is often quoted in other texts.

At outlet from the runner the flow plane is simplified as though it were actually in the radial/tangential plane. This simplification will not affect the subsequent analysis of the flow but it must be conceded that some component of velocity in the axial direction does exist at runner outlet.

The water leaves the runner with a relative flow angle β_3 and a relative flow velocity w_3. The absolute velocity at runner exit is found by vector addition, i.e., $\mathbf{c}_3 = \mathbf{w}_3 + \mathbf{U}_3$. The relative flow angle, β_3, at runner exit is given by

$$\beta_3 = \tan^{-1}[(c_{\theta 3} + U_3)/c_{r3}]. \tag{9.15}$$

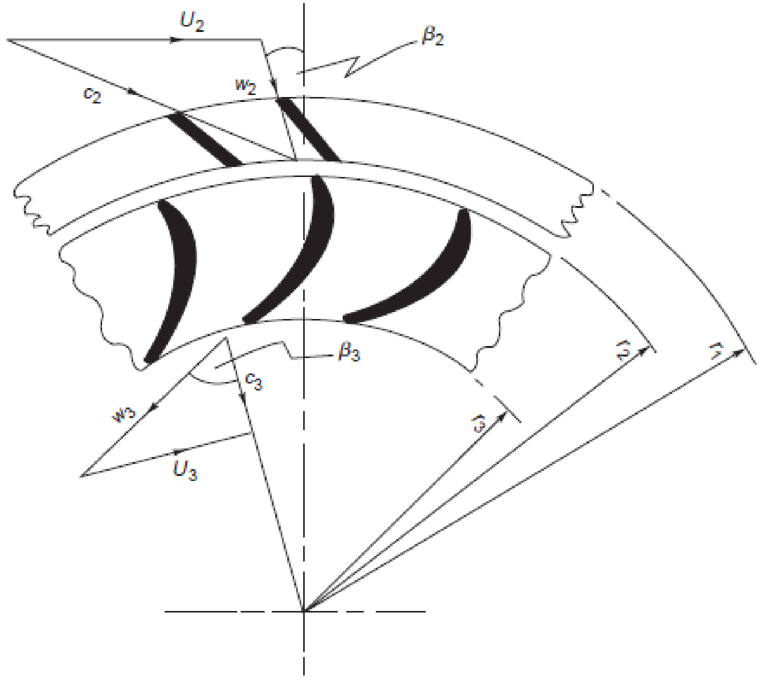


FIGURE 9.13

Sectional Sketch of Blading for a Francis Turbine Showing Velocity Diagrams at Runner Inlet and Exit

When a Francis turbine is required to operate at part load, the power output is reduced by swivelling the guide vanes to restrict the flow, i.e., Q is reduced, while the blade speed is maintained constant. Figure 9.14 compares the velocity triangles at full load and at part load from which it will be seen that the relative flow at runner entry is at a high incidence and at runner exit the absolute flow has a large component of swirl. Both of these flow conditions give rise to high head losses. Figure 9.15 shows the variation of hydraulic efficiency for several types of turbine, including the Francis turbine, over the full load range at constant speed and constant head.

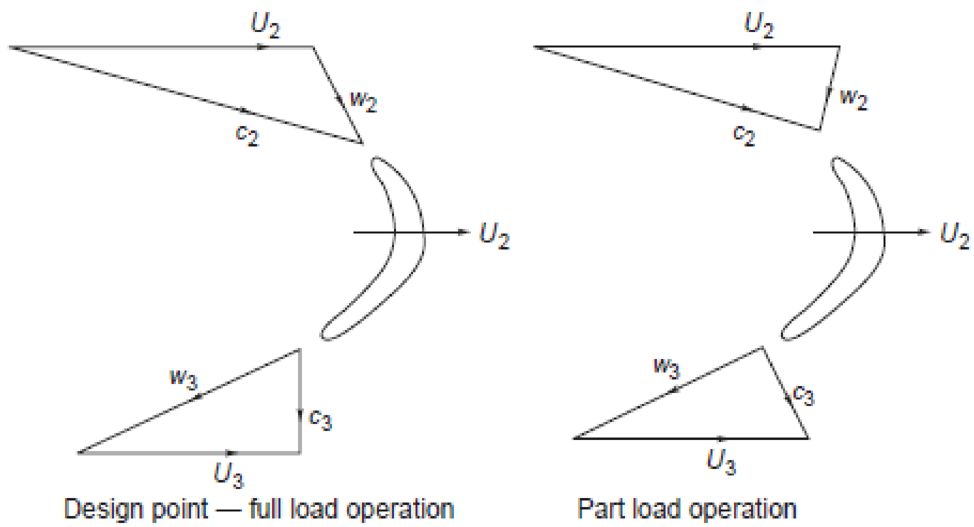


FIGURE 9.14

Comparison of Velocity Triangles for a Francis Turbine at Full Load and at Part Load Operation

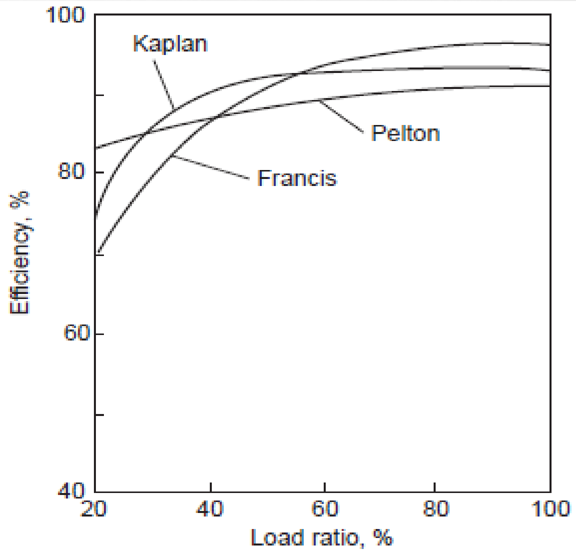


FIGURE 9.15

Variation of Hydraulic Efficiency for Various Types of Turbine over a Range of Loading, at Constant Speed and Constant Head

Basic Equations

Euler's turbine equation, eqn. (1.18c), in the present notation, is written as

$$\Delta W = U_2 c_{\theta 2} - U_3 c_{\theta 3}. \tag{9.16a}$$

If the flow at runner exit is without swirl then the equation reduces to

$$\Delta W = U_2 c_{\theta 2}. \tag{9.16b}$$

The effective head for all reaction turbines, H_E, is the total head available at the turbine inlet *relative to the surface of the tailrace*. At entry to the runner the energy available is equal to the sum of the kinetic, potential and pressure energies:

$$g(H_E - \Delta H_N) = \frac{p_2 - p_a}{\rho} + \frac{1}{2}c_2^2 + gz_2, \tag{9.17}$$

where ΔH_N is the loss of head due to friction in the volute and guide vanes and p_2 is the *absolute* static pressure at inlet to the runner.

At runner outlet the energy in the water is further reduced by the amount of specific work ΔW and by friction work in the runner, $g\Delta H_R$ and this remaining energy equals the sum of the pressure potential and kinetic energies:

$$g(H_E - \Delta H_N - \Delta H_R) - \Delta W = \frac{1}{2}c_3^2 + p_3/\rho - p_a/\rho + gz_3, \tag{9.18}$$

where p_3 is the *absolute* static pressure at runner exit.

By differencing eqns. (9.17) and (9.18), the specific work is obtained:

$$\Delta W = (p_{02} - p_{03})/\rho - g\Delta H_R + g(z_2 - z_3), \tag{9.19}$$

where p_{02} and p_{03} are the absolute total pressures at runner inlet and exit.

Figure 9.16 shows the draft tube in relation to a vertical-shaft Francis turbine. The most important dimension in this diagram is the vertical distance ($z = z_3$) between the exit plane of the runner and the

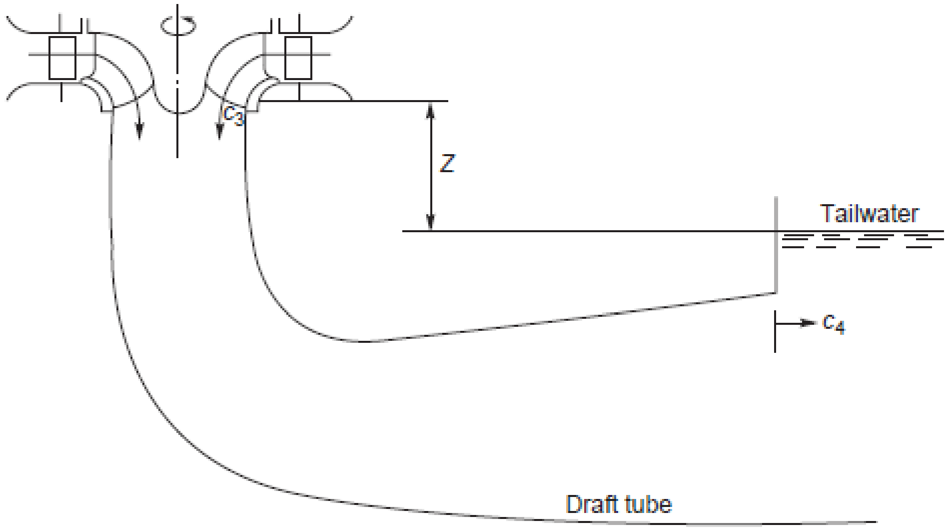


FIGURE 9.16

Location of Draft Tube in Relation to Vertical Shaft Francis Turbine

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$$g(H_E - \Delta H_N - \Delta H_R) - \Delta W = \frac{1}{2}c_3^2 + p_3/\rho - p_a/\rho + gz_3, \tag{9.18}$$

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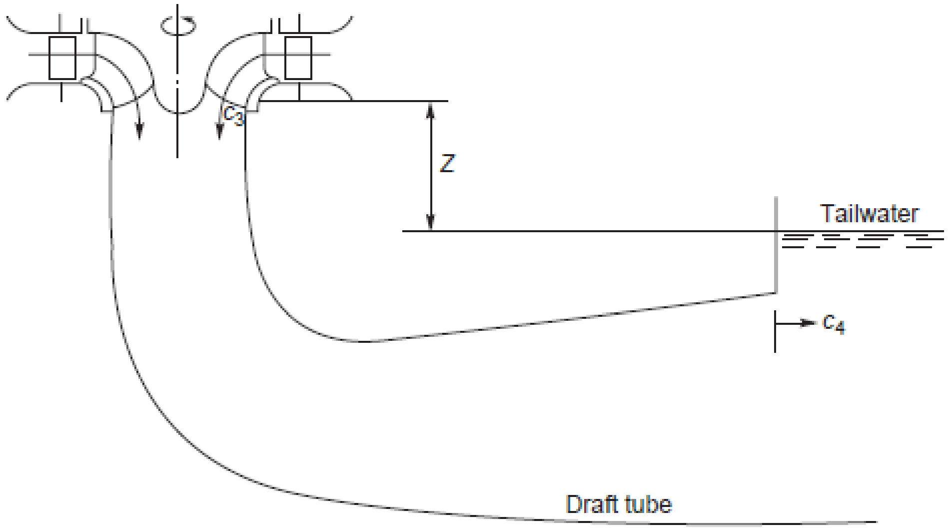

FIGURE 9.16

Location of Draft Tube in Relation to Vertical Shaft Francis Turbine free surface of the tailrace. The energy equation between the exit of the runner and the tailrace can now be written as

$$p_3/\rho + \frac{1}{2}c_3^2 + gz_3 - g\Delta H_{DT} = \frac{1}{2}c_4^2 + p_a/\rho, \tag{9.20}$$

where ΔH_{DT} is the loss in head in the draft tube and c_4 is the flow exit velocity.

The hydraulic efficiency is defined by

$$\eta_h = \frac{\Delta W}{gH_E} = \frac{U_2 c_{\theta 2} - U_3 c_{\theta 3}}{gH_E} \tag{9.21a}$$

and, whenever $c_{\theta 3} = 0$,

$$\eta_H = \frac{U_2 c_{\theta 2}}{gH_E}. \tag{9.21b}$$

The overall efficiency is given by $\eta_o = \eta_m \eta_H$. For very large turbines (e.g., 500–1000 MW) the mechanical losses are then relatively small, $\eta \rightarrow 100\%$ and effectively $\eta_o \approx \eta_H$.

For the Francis turbine the ratio of the runner tip speed to the jet velocity, $\nu = U_2/c_1$, is not as critical for high efficiency operation as it is for the Pelton turbine and can lie in a fairly wide range, e.g., $0.6 \le \nu \le 0.95$. In most applications the Francis turbine is used to drive a synchronous generator and the rotational speeds chosen are those appropriate to either 50 or 60 cycles per second. The speed must then be maintained constant.

It is possible to obtain part load operation of the turbine by varying the angle of the guide vanes. The guide vanes are pivoted and set to an optimum angle via a gearing mechanism. However, part load operation normally causes a whirl velocity to be set up in the flow downstream of the runner causing a reduction in efficiency. The strength of the vortex may be enough to cause a cavitation bubble to form along the axis of the draft tube. (See Section 9.8, Cavitation.)