Hydraulic Machines

Presentation Topic: Francis Turbines.

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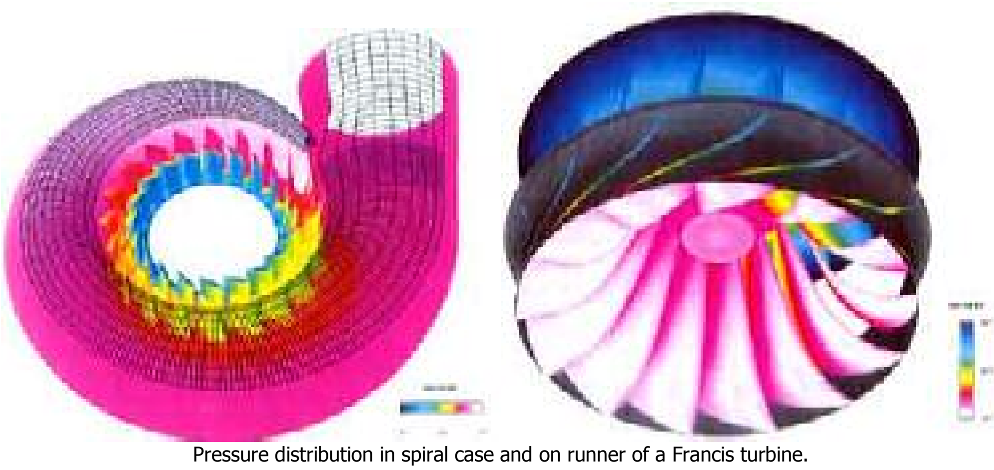

Pressure distribution in spiral case and on runner of a Francis turbine.

Francis Turbines

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Abstract:

A reaction turbine develops power from the combined action of pressure energy and kinetic energy. Early turbines used a radial outward flow of water. The British-born American engineer James B. Francis designed a turbine in which the flow was inward. The so-called **reaction** or Francis turbine is the most widely used hydraulic turbine for medium pressure, medium flow applications.

The water passes through a snail-shaped **scroll case** , through **wicket gates** that control the amount of water and into the submerged **runner**. The curved blades of the runner change the momentum of the water, which produces a net force or reaction in the turbine. This reaction has a tangential component that turns the wheel. With a Francis turbine, downstream pressure can be above zero. Precautions must be taken against an upstream water hammer when this type of turbine is used at the higher pressures. Under the emergency stop, the turbine overspeeds. One would think that more water is going through the turbine than before the trip occurred, since the turbine is spinning faster. However the turbine has been designed to work most efficiently at the design speed, so less water actually flows through the turbine during overspeed. This is called flow “choking”. Pressure relief valves are added to prevent a water hammer due to the abrupt change of flow. Besides limiting pressure rise, the pressure relief valve prevents the water hammer from stirring up sediment in the pipes, which can cause dirty water complaints.

Description:

In the reaction turbine a portion of the energy of the fluid is converted into kinetic energy by the fluid’s passing through adjustable wicket gates before entering the runner, and the remainder of the conversion takes place through the runner. All passages are filled with liquid, including the passage (draft tube) from the runner to the downstream liquid surface. The static pressure occurs on both sides of the vanes and hence does no work. The work done is entirely due to the conversion to kinetic energy.

The reaction turbine is quite different from the impulse turbine. In an impulse turbine all the available energy of the fluid is converted into kinetic energy by a nozzle that forms a free jet. The energy is then taken from the jet by suitable flow through moving vanes. The vanes are partly filled, with the jet open to the atmosphere throughout its travel through the runner.

In contrast, in the reaction turbine the kinetic energy is appreciable as the fluid leaves the runner and enters the draft tube. The function of the draft tube is to reconvert the kinetic energy to flow energy by a gradual expansion of the flow cross section. Application of the energy equation between the two ends of the draft tube shows that the action of the tube is to reduce the pressure at its upstream end to less than atmospheric pressure, thus increasing the effective head across the runner to the difference in elevation between head water and tail water, less losses.

The first successful reaction turbine was built and tested in 1849 by an American engineer named J. B. Francis. His design was superior to that of most earlier forms in that the flow was directed inwards under pressure, so that any tendency towards over-speeding was partly counteracted by the reduction of the flow caused by the increase in centrifugal pressure.

The vaned wheel or runner was shaped rather like a centrifugal impeller, flow being predominantly radial with the radii at entry and exit the same for all flow paths. As the need for greater power outputs at higher speeds developed it became necessary to adapt the runner for larger flows without increasing the diameter. This could only be done by arranging for the water to be discharged in a radial-cum-axial direction, the resulting mixed flow type of design (fig. 1) being now the standart practice.

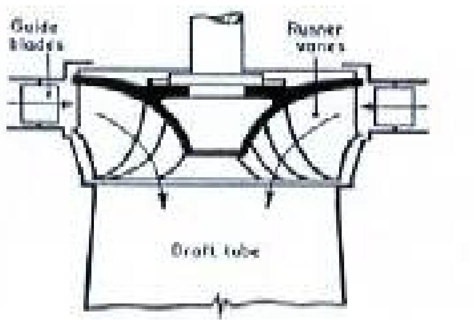


Figure1: Section elevation of Francis turbine runner

Although modern inward-flow turbines bear little resemblance to the original Francis machine, the operating principle is essentially the same and the name has been retained. The present head range is from 30 m to about 450 m and as this is the most common head availability the machine enjoys a great numerical superiority over the other types. There is certainly a preponderance of Francis machines in Scotland.
Water is directed (with appreciable tangential velocity component) into the runner by means of a spiral casing and a number of aerofoil-shaped blades spaced evenly around the periphery (fig. 2).

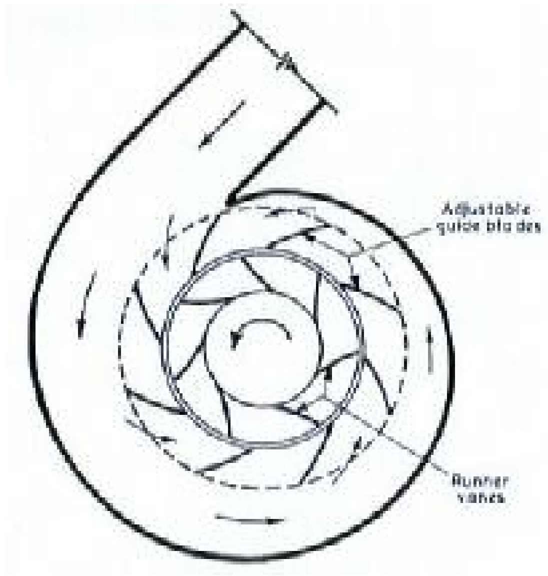


Figure 2: Diagrammatic plan of spiralcasing and runner.

These guide blades, called the turbine gate, are adjustable, the amount of opening being controlled by the turbine governor through a linkage mechanism. Their role is to guide the flow into the runner with the minimum amount of turbulance, as well as to regulate the discharge and hence the power output.

Because of the converging boundries, the velocity energy at entry to the runner is greater than that in the pipeline and the pressure head correspondingly lower. In the course of its path through the runner there is a further drop in pressure, the water being finally discharged at the centre at a low pressure and with little or no tangential velocity component (fig 3).

The driving torque is derived from the deviation in the direction of flow and the change in pressure and velocity energy. Owing to the problems (e.g. leakage through clearance rings) associated with the high pressures and velocities, there is an upper limit on the head for this type of machine.

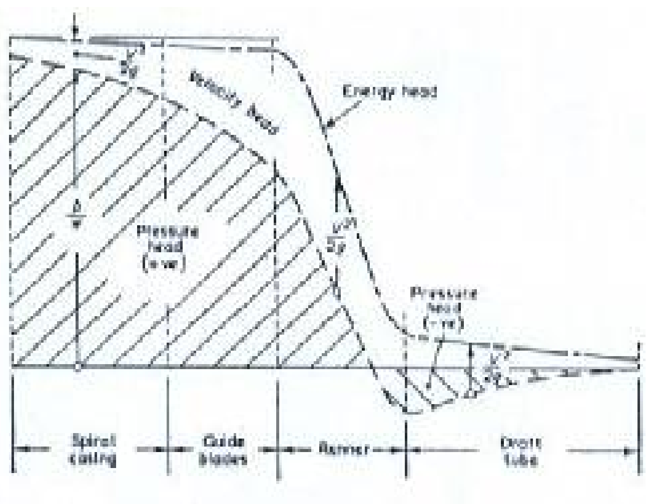


Figure3: Energy head variation in turbine passages

The velocity head at discharge from the runner may amount to 20 per cent, or perhaps more, of the available head and as with pumps it is clearly important to convert as much as possible of this otherwise wasted energy to useful pressure head. This is accomplished by means of an expanding passage, called a *draft tube,* which finally discharges the water at a relatively low velocity to the tailwater. A straight conical tube of adequate length would entail expensive excavation and an elbow shape is preffered; submergence of the exit is essential. Draft tube design cannot be divorced from that of the tubine proper; Fig. 4 shows the cross section of a typical complete installation. The vertical mounting results in cosiderable economy in space demands.

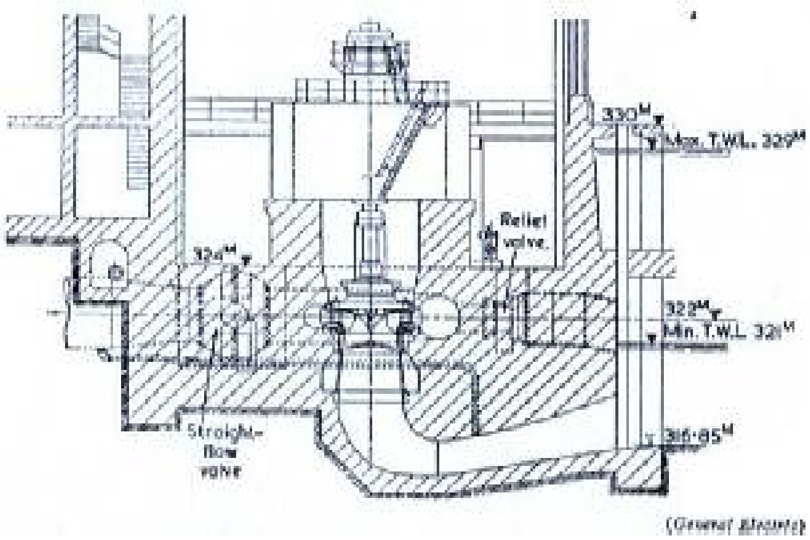


Figure 4: Section elevation of a Francis turbine installation

By recovering pressure head in the draft tube (about 70 percent conversion is possible), the pressure at exit from the runner is reduced below atmospheric, so that water is in effect sucked through the turbine. Thus the full head above tailwater level is potentially available for useful work –an important consideration with a medium or low head plant, and a distinct advantage over the pelton wheel. Moreover, there is some welcome flexibility in the level at which the runner may be set, the upper limit being governed by the need to avoid cavitation.

If the supply pipeline is of appreciable length, a pressure relief valve is fitted in a short conduit leading directly from the pipeline to the tailwater. It performs the same function as the deflector hood on the nozzle of a Pelton wheel. The valve is opened on a sudden reduction of load, thus by-passing the flow, and slowly closed again when the gate movement is completed. The diversion conduit and valve are clearly shown in fig. 4.

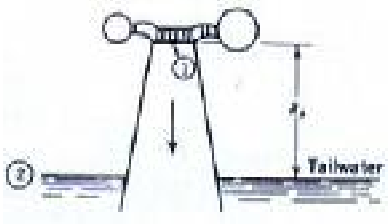


Figure 5: Draft tube

By referring to Fig. 5, the energy equation from 1 to 2 yields

$Z_s + V_1/2g + P_1/\gamma = 0 + 0 + 0 + losses$

The losses include the expansion loss, friction, and velocity head loss at the exit from the draft tube, all of which are quite small; hence,

$P_1/\gamma = -z_s - V_1/ 2g + losses$

Shows that considerable vacuum is produced at section 1, which effectively increases the head across the turbine runner. The turbine setting must not be too high, or cavitation occurs in the runner and draft tube.

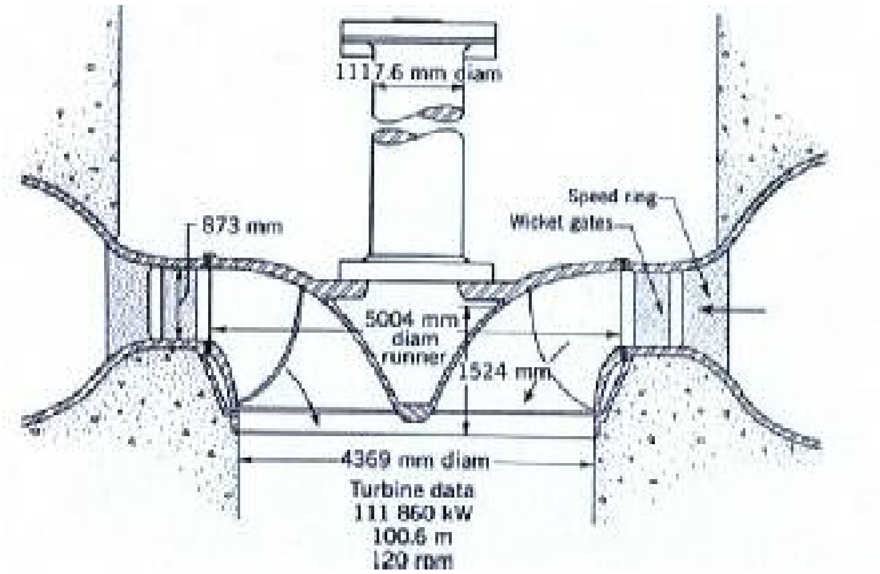


Figure 6: Francis Turbine.

Example:

A turbine has a velocity of 6 m/s at the entrance to the draft tube and a velocity of 1.2 m/s at the exit. For friction losses of 0.1 m and a tail water 5 m below the entrance to the draft tube, find the pressure head at the entrance.

Solution:

$P_1/\gamma = - 5 - 6^2/(2 \times 9.81) + 1.2^2/ (2 \times 9.81) + 0.1 = -6.66\ m$

As the kinetic energy at the exit from the draft tube is lost. Hence, a suction head of 6.66 m is produced by the presence of the draft tube.

There are two forms of the reaction turbine in common use, the Francis turbine and the propeller turbine. In both, all passages flow full, and energy is converted to useful work entirely by changing the moment of momentum of the liquid. The flow passes first through the wicket gates, which impart a tangential and a radially inward velocity to the fluid. A space between the wicket gates and the runner permits the flow to close behind the gates and move as a free vortex, without external torque being applied.

In the Francis turbine (Fig.6) the fluid enters the runner so that the relative velocity is tangent to the leading edge of the vanes. The radial component is gradually changed to an axial component, and the tangential component is reduced as the fluid traverses the vane, so that at the runner exit the flow is axial with very little whirl (tangential component) remaining. The pressure has been reduced to less than atmospheric, and most of the remaining kinetic energy is reconverted to flow energy by the time it discharges from the draft tube. The Francis turbine is best suited to medium-head installations from 25 to 180 m and has efficiency between 90 and 95 percent for the larger installations. Francis turbines are designed in the specific speed range of 40 to 240 (m, kW, rpm) with best efficiency in the range 150 to 230 (m, kW, rpm).

Theories:

As I explained before, a Francis turbine is a reaction machine, which means that during energy transfer in the runner (impeller) there is a drop in static pressure and a drop in velocity head. Only part of the total head presented to the machine is converted to velocity head before entering the runner. This is achieved in the stationary but adjustable guide vanes, shown in Fig. 7. It is important to realize that the machine is running full of water, which enters the impeller on its whole periphery. The guide vane ring may surround the runner on its outer periphery, in which case the flow of fluid is towards the runner centre. In such a case, the turbine is known as an *inward flow* type. The alternative arrangement is for the fluid to enter the guide vanes at the centre and to flow radially outwards into the runner which now surrounds the guide vanes. Such a turbine is known as the *outward flow* type.

Consider an inward flow Francis turbine, represented diagrammatically in Fig. 7. A section of the runner guide vane ring,showing the blades, vanes and velocity triangles, is given in Fig. 8.

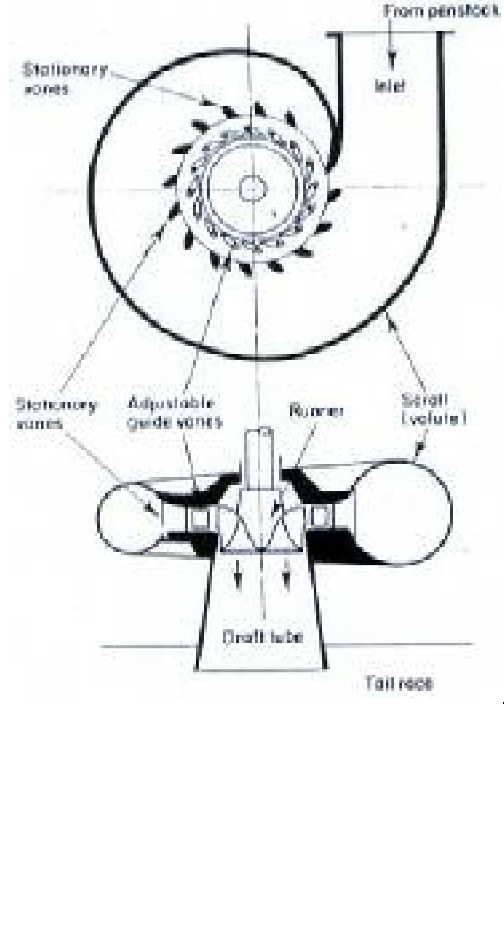


Figure 7: Francis turbine

The total head available to the machine is H and the water velocity on entering the guide vanes is v_o. The velocity leaving the guide vanes is v_1 and is related to v_o by the continuity equation:

$$v_o A_o = v_{f1} A_1 .$$

But $\quad v_{f1} = v_1 \sin\theta,\quad$ so that

$$v_o A_o = v_1 A_1 \sin\theta .$$

The direction of v_1 is governed by the guide vane angle θ. It is chosen in such a way that the relative velocity meets the runner blade tangentially, i.e. it makes an angle β_1 with the tangent at the blade inlet. Thus,

$$\tan\theta = v_{f1} / v_{w1} \qquad \text{and} \qquad \tan\beta_1 = v_{f1} / (u_1 - v_{w1}).$$

Eliminating v_{w1} from the two equations:

$$\tan\beta_1 = v_{f1} / (u_1 - v_{f1} / \tan\theta)$$

or $\quad \cot\beta_1 = (u_1 / u_{f1}) - \cot\theta .$

Therefore,

$$(u_1 / v_{f1}) = \cot\beta_1 + \cot\theta .$$

or $\quad u_1 = v_{f1} (\cot\beta_1 + \cot\theta) \qquad (1)$

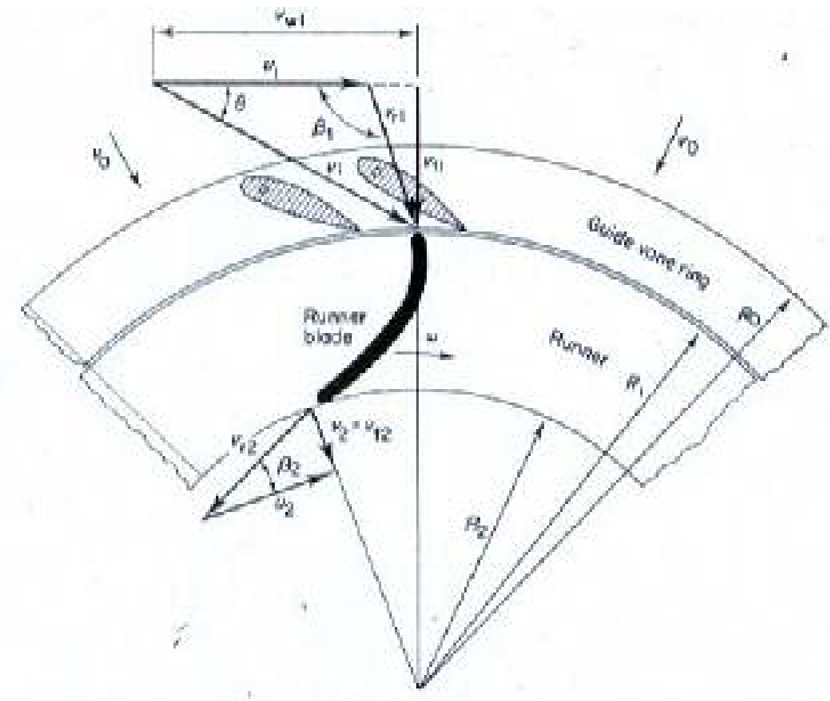


Figure8: Section through part of a Francis turbine

The total energy at inlet to the runner consists of the velocity head $v_1^2 / 2g$ and pressure head H_1 . In the runner, the fluid energy is decreased by E, which is transferred to the runner. Water leaves the impeller with kinetic energy $v_2^2 / 2g$. Thus, the following energy equations hold:

$$H = (v_1^2 / 2g) + H_1 + h'_1$$

and

$$H = E + (v_2^2 / 2g) + h_1 ,$$

In which h'_1 is the loss of the head in the guide vane ring and h_1 is the loss in the whole turbine, including entry, guide vanes and runner.

The energy transferred E is given by the Euler's equation, which, for the maximum energy transfer condition secured when $v_{w2} = 0$, takes the form

$$E = v_{w1} u_1 / g.$$

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 v_2 is radial as shown in Fig. 8. From the outlet velocity diagram, then, it follows that

$$\tan \beta_2 = v_2 / u_2 ,$$

but, since $v_{w2} = 0$, then $v_2 = v_{f2}$ and , by the continuity equation,

$$A_1 v_{f1} = A_2 v_{f2},$$

So that β_2 can be determined.

If the condition of no whirl at outlet is satisfied, then the second energy equation takes the form

$$H = v_{w1} u_1 / g + v_2^2 / 2g + h_1 . \qquad (2)$$

The hydraulic efficiency is given by

$$\eta_h = E / H = v_{w1} u_1 / gH \qquad (3)$$

and the overall efficiency by

$$\eta = P / \rho gQH, \qquad (4)$$

In which P is the power output of the machine, Q is the volumetric flow rate through it and H is the total head available at the turbine inlet.

The relationship between the runner speed and spouting velocity, $\sqrt{(2gH)}$, for the Francis turbine is 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. Relationship between specific speed and operation (rated) speed is shown in Fig. 9.

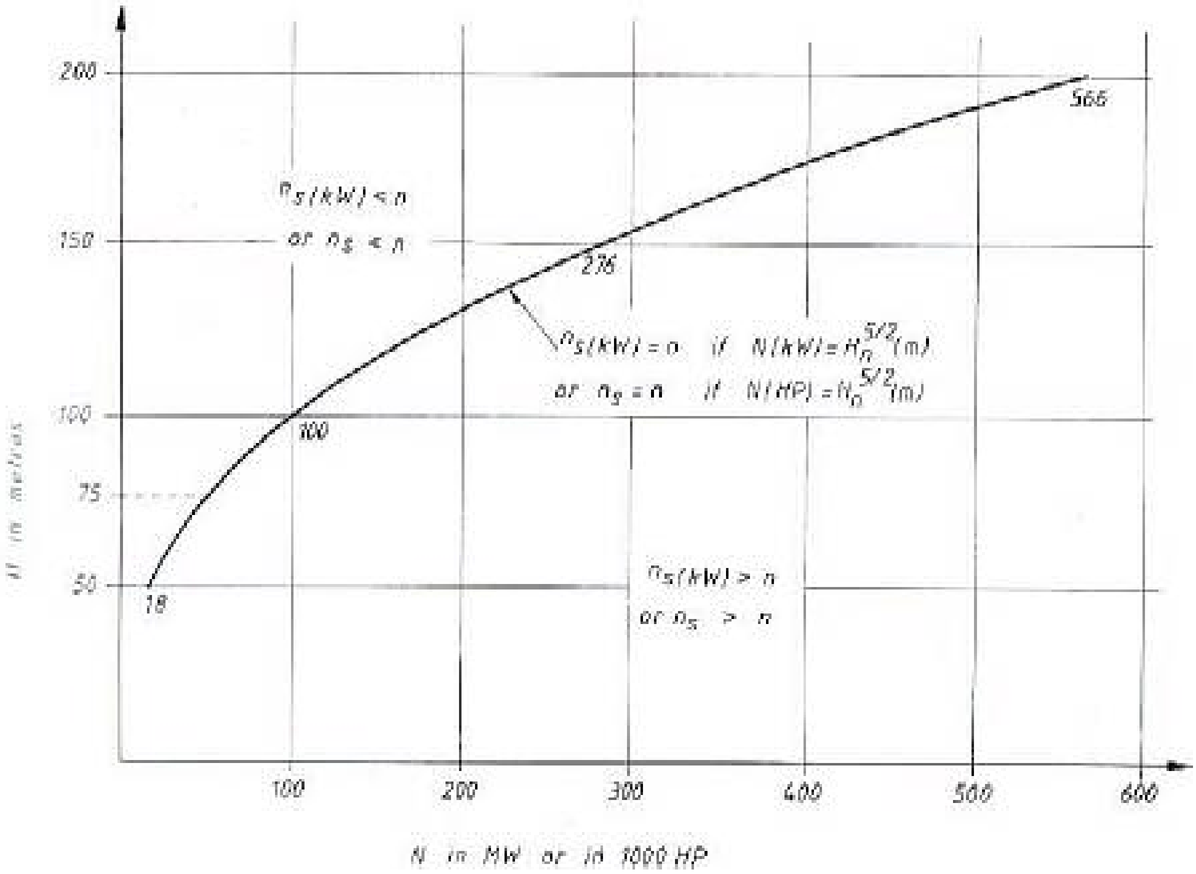


Figure 9: Relationship between specific speed and operation (rated) speed

Similarly to a Pelton wheel, a Francis turbine usually drives an alternator and, hence, its speed must be constant. Since the total head available is constant and dissipation of energy by throttling is undesirable, the regulation at part load is achieved by varying the guide vane angle θ, sometimes reffered to as the gate. This is possible because there is no requirement for the speed ratio to remain constant. A change in θ results in a change in v_w and v_f. Thus, E is altered for given u. However, such changes mean a departure from the 'no shock' conditions at inlet and also give rise to the whirl component at outlet. As a result, the efficiency at part load falls off more rapidly than in the case of the Pelton wheel. Also, vortex motion in the draft tube resulting from the whirl component may cause cavitation in the centre. Sudden load changes are catered for either by a bypass valva or by a surge tank.

Example:

In an inward-flow reaction turbine, the supply head is 12 m and the maximum discharge is 0.28 $m^3\ s^{-1}$. External diameter = 2 (internal diameter) and the velocity of flow is constant and equal to 0.15 $\sqrt{(2gh)}$. The runner vanes are radial at inlet and the runner rotates at 300 rev min^{-1}.

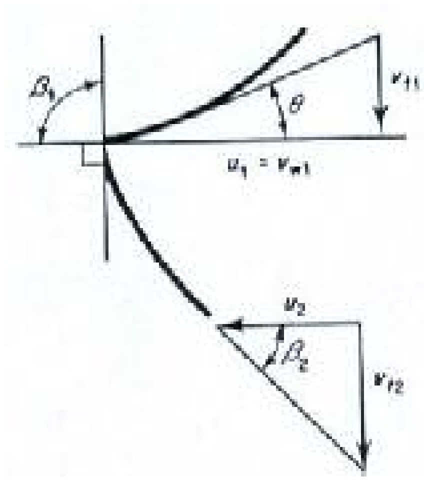


Figure 10:

Determine **(a)** the guide vane angles, **(b)** the vane angle at exit for radial discharge, (c) widths of the runner at inlet and exit. The vanes occupy 10 percent of the circumference and the hydraulic efficiency is 80 percent.

Solution:

(a) Velocity of flow is given by:

$$v_{f1} = v_{f2} = 0.15\ \sqrt{(2gh)} = 0.15\ \sqrt{(2 \times 9.81 \times 12)} = 2.3 \text{ m s}^{-1}.$$

Efficiency is given by equation;

$$\eta_h = E / H = v_{w1}\, u_1 / gH$$

but $u_1 = [\, 1 - (v_{f1} / v_{w1})\tan \beta_1 \,]$ and $v_{f1} / v_{w1} = \tan\theta$.

Therefore,

$$\eta = v^2_{wl}\, (1 - \tan\theta / \tan\beta_1) / gH,$$

but, since $\beta_1 = 90^o$ and η = 80 percent, it follows that

$$v_{w1} = \sqrt{(0.8 \times 9.81 \times 12)} = 9.7 \text{ m s}^{-1}\ .$$

Therefore, $u_2 = 9.7 \text{ m s}^{-1}$ and

$$\tan\theta = v_{f1} / u_1 = 2.3 / 9.7 = 0.237\,,$$

$$\boldsymbol{\theta = 13^o 20'}.$$

(b) Since internal diameter = ½ x external diameter,

$$u_2 = ½\, u_1 = 4.85 \text{ m s}^{-1}\ .$$

Therefore,

$$\tan \beta_2 = v_{f2} / u_2 = 2.3 / 4.85 = 0.475$$

$$\boldsymbol{\beta_2 = 25^o 20' .}$$

(c) Now $u_1 = \omega r_1$, where $\omega = 300 (2\pi/60)$ rad s^{-1}. Therefore,

$$r_1 = u_1 / \omega = 9.7 \text{ x } 60/300 \text{ x } 2\pi = 0.31 \text{ m.}$$

Therefore, breadth at outlet is given by

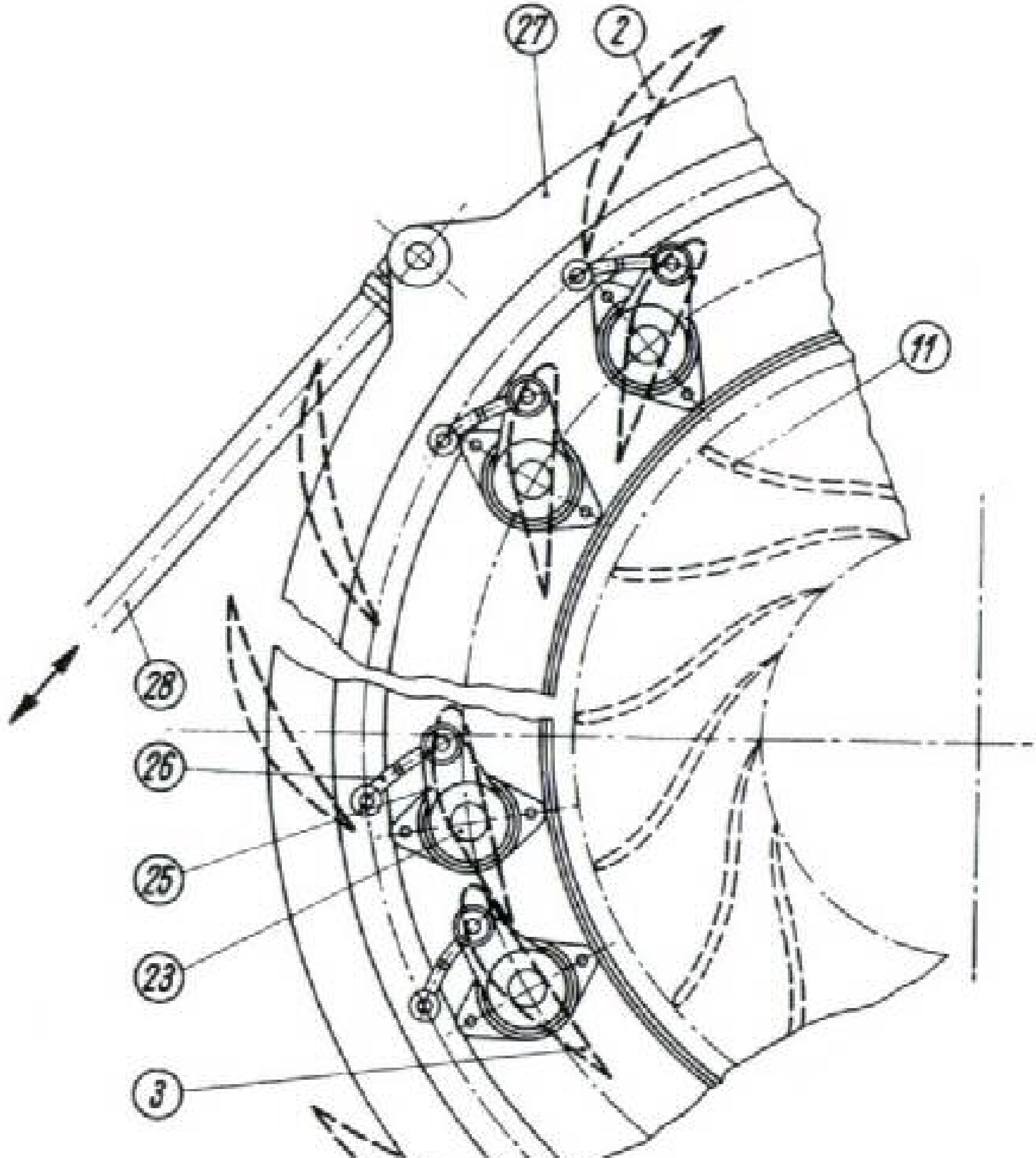
$$b_1 = 0.28 / 2.3 \text{ x } 0.9 \text{ x } 2\pi \text{ x } 0.31 = 0.0696 \text{ m}$$

$$\boldsymbol{b_1 = 69.6 \text{ mm.}}$$

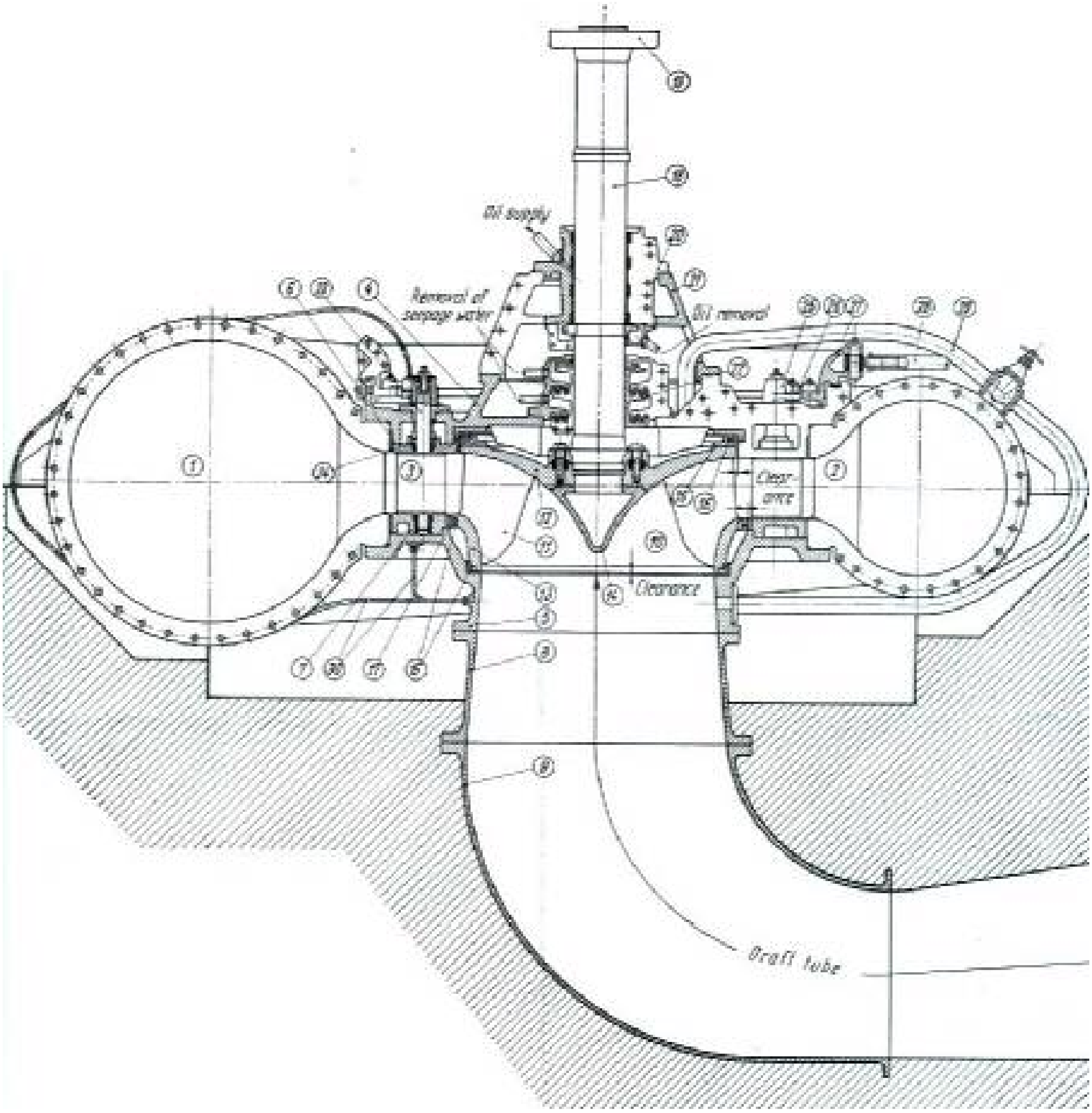
Since velocity of flow is constant, and internal diameter = ½ external diameter, breadth at outlet is given by

$$b_2 = 2 \text{ x } 69.6 = \mathbf{139.2 \text{ mm.}}$$

Francis Turbine Components



2. Stay Vane
3. GuideVane
11. Runner Vane
25. Regulating Arm
23. Guide-vane stem
26. Shearing Pin
27. Shifting ring
28. Drawbar



1. Scroll case
4. Head cover
5. Discharge ring
6. Upper cover inset for the wicket gate
7. Lower cover inset for the wicket gate
8. Throat liner
9. Draft-tube liner
10. Runner
11. Runner vane
12. Runner crown
13. Runner shroud
14. Runner cone
15. Wearing ring on runner-shroud
16. Wearing ring on head cover
17. Wearing ring on discharge ring
18. Turbine shaft
19. Coupling flange
20. Guide Bearing
21. Guide wearing web
22. Labyrinth shaft sealing
24. Guide-vane packing
29. Pressure-equalizing pipe
30. Drain pipes

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