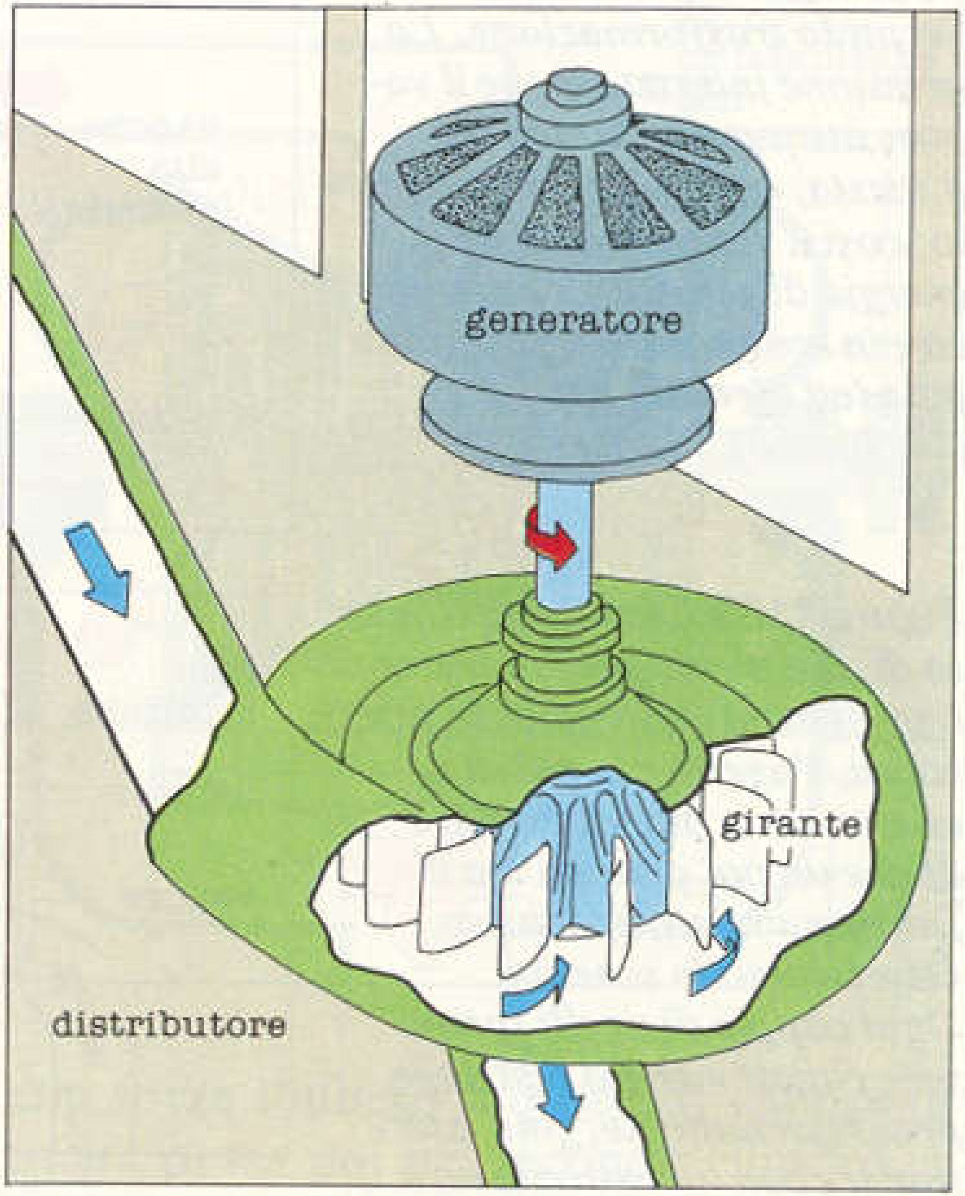
Reaction Turbines



Francis Turbine

In 1849, an eminent hydraulic engineer J. B. Francis designed an inward flow reaction turbine.

Because of somewhat **fixed relationship between the net head and the peripheral speed and hence the diameter of the Pelton wheel**, it is observed that ***as the head diminishes, the diameter of the wheel to develop a given power increases and the speed of revolution decreases***. If the head is less than 150 m, ***Pelton wheels become so slow and unwieldy that they are unsuitable for power generations***. In such a case, the Francis turbine is more suitable.

In the original design of the Francis turbine the flow was purely radial both at the entrance and exit. In such turbines, the inner diameter was quite large. ***To make the turbine more compact, it became necessary to discharge the water in the axial direction***. This resulted in the **mixed flow turbine**, where **the water enters the turbine in the radial direction and discharges in the axial direction**.

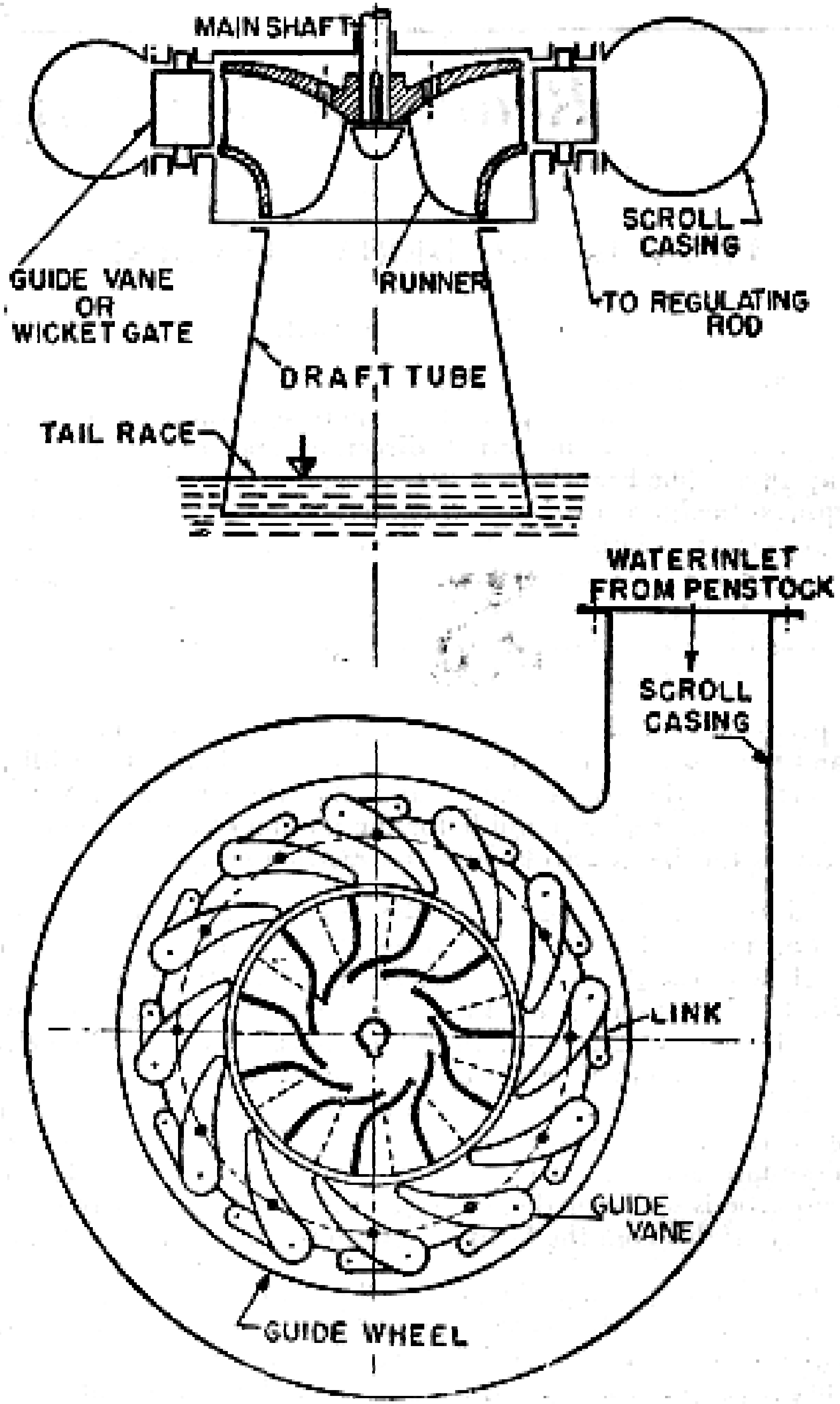
In a Francis turbine, ***only a part of the head acting on the turbine is transformed into kinetic energy and the rest remains as pressure head***. ***There is a difference of pressure between the guide vanes (entrance) and the runner (exit) which is called the*** **reaction pressure**; and is **responsible for the rotation of the runner**. That is why a Francis turbine is also known as a **reaction turbine**.

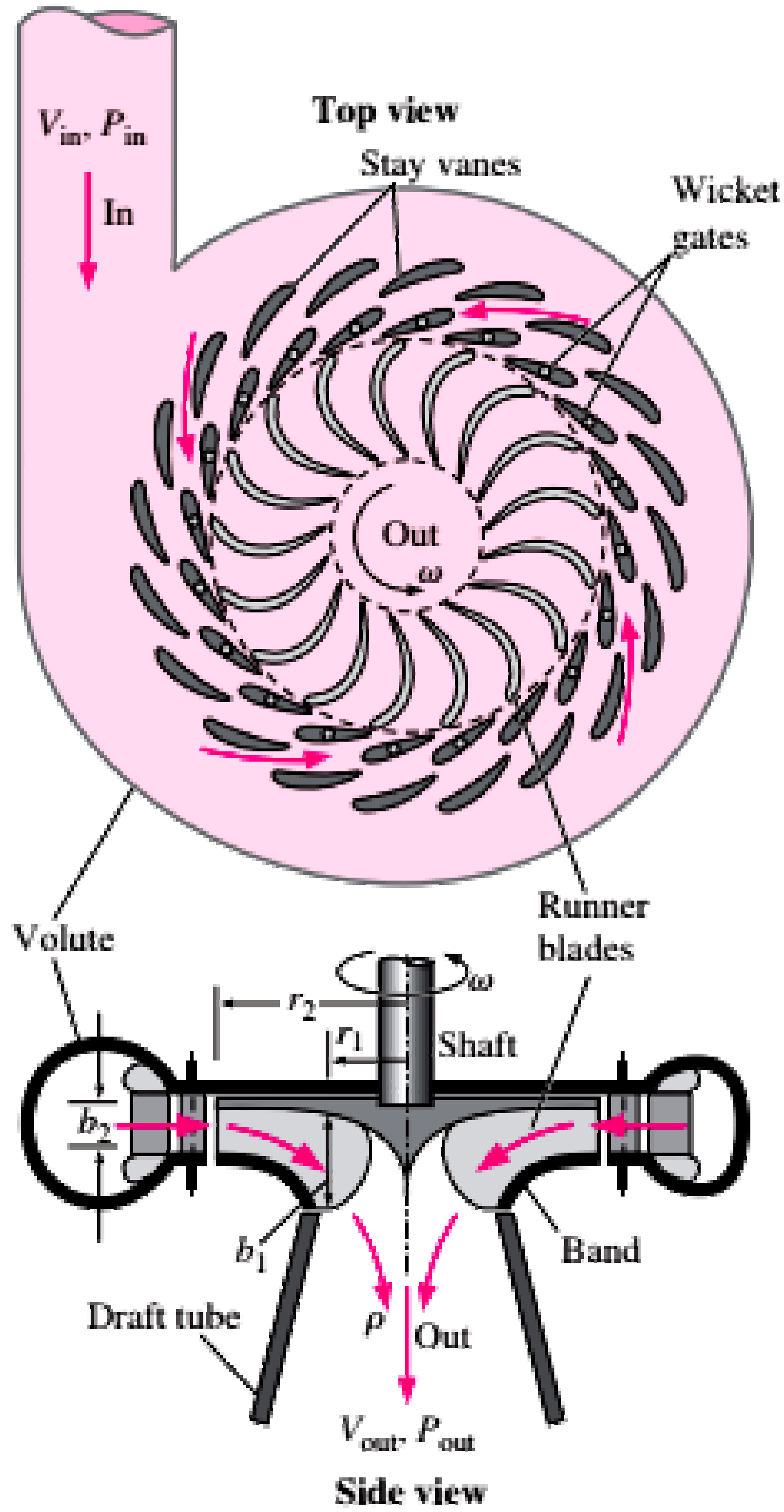
The Francis turbine operates under medium heads and also requires medium quantity of water. It is employed in the medium head power plants. This type of turbine covers a wide range of heads (30 to 450 m).

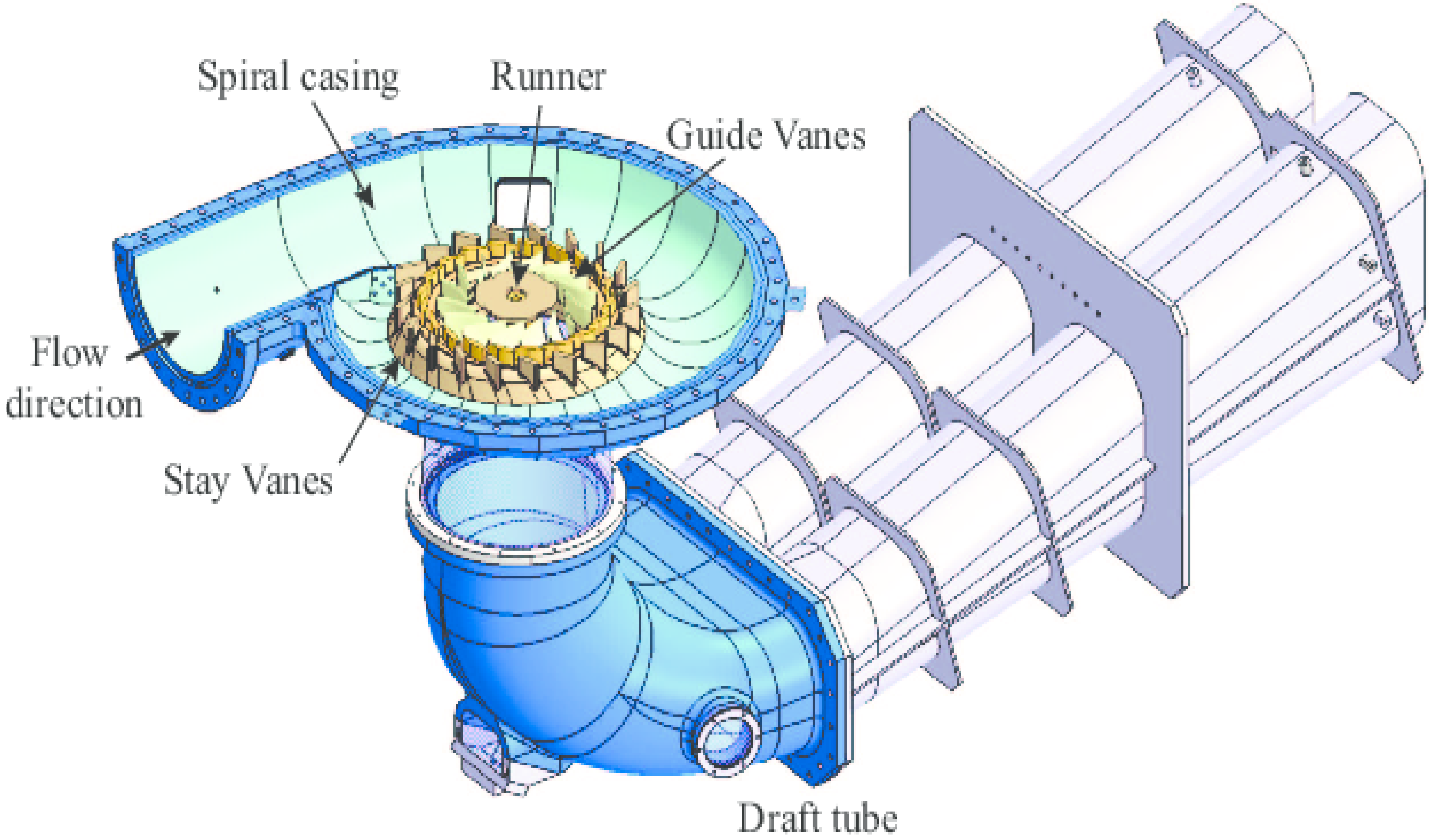
Main Components of Modern Francis Turbine

(a) Penstock: Penstock is a waterway to carry water from the reservoir to the turbine casing. ***Trash racks are provided*** at the inlet of penstock in order to obstruct the debris entering in it.

(b) Spiral or Scroll Casing: The scroll casing surrounds the turbine runner and the guide mechanism. The casing is placed in between the penstock outlet and guide vanes. ***To avoid loss of efficiency, the flow of water from the penstock to the runner should be such that it will not form***


MAIN SHAFT
SCROLL CASING
GUIDE VANE OR WICKET GATE
RUNNER
TO REGULATING ROD
DRAFT TUBE
TAIL RACE
WATER INLET FROM PENSTOCK
SCROLL CASING
LINK
GUIDE VANE
GUIDE WHEEL


V_{in}, P_{in}
In
Top view
Stay vanes
Wicket gates
Out
ω
Volute
Runner blades
ω
r_2
r_1
Shaft
b_2
b_1
Band
Draft tube
ρ
Out
V_{out}, P_{out}
Side view


Spiral casing
Runner
Guide Vanes
Flow
direction
Stay Vanes
Draft tube

eddies. In order to distribute the water around the guide ring evenly, the scroll casing is designed with a cross-sectional area reducing uniformly around the circumference, maximum at the entrance and nearly zero at the tip. This gives a spiral shape and hence the casing is named as spiral casing. **In the case of big units**, the inside circumference of ***casing has stay vanes each directing the water to the guide vanes***. For small heads, casing is made of concrete, but for high heads, it is usually made of steel plates or cast steel.

(c) Guide Mechanism: Francis Turbine consists of ***fixed guide vanes*** called **stay vanes**, ***adjustable guide vanes*** called **wicket gates**, and ***rotating blades*** called **runner blades**. **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***. The guide vanes or wicket gates, as they are sometimes called are fixed between two rings in the form of a wheel, known as **guide wheel**. ***The guide vanes form gradually-contracting passages***. ***The guide vanes have a cross-section known as aerofoil section***. ***This particular cross-section allows water to pass over them without forming eddies and with minimum friction losses***. Each guide vane can rotate about its pivot centre which is connected to the regulating shaft by means of a regulating rods, generally, two in number. ***By rotating the regulating shaft the guide vanes be closed or opened*** thus allowing a variable quantity of water according to the needs. ***The regulating shaft is operated by means of a governor whose function is to keep the speed of the turbine constant at varying loads***. With a decrease in load the speed of the turbine always tends to increase. To bring the speed back to the rated value, the governor is used to reduce the guide vane opening thereby allowing less water to strike the runner. The guide vanes are generally made of cast steel. ***Momentum is exchanged between the fluid and the runner as the runner rotates, and there is a large***

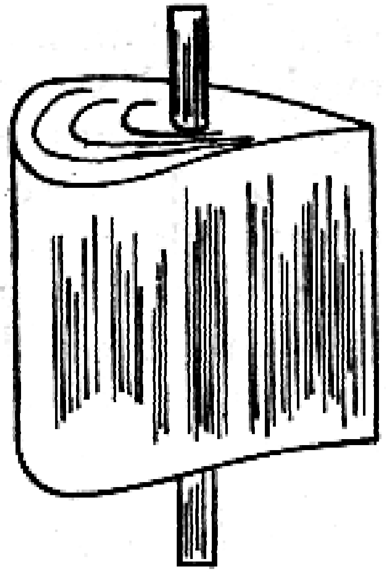


Fig: Guide Vane

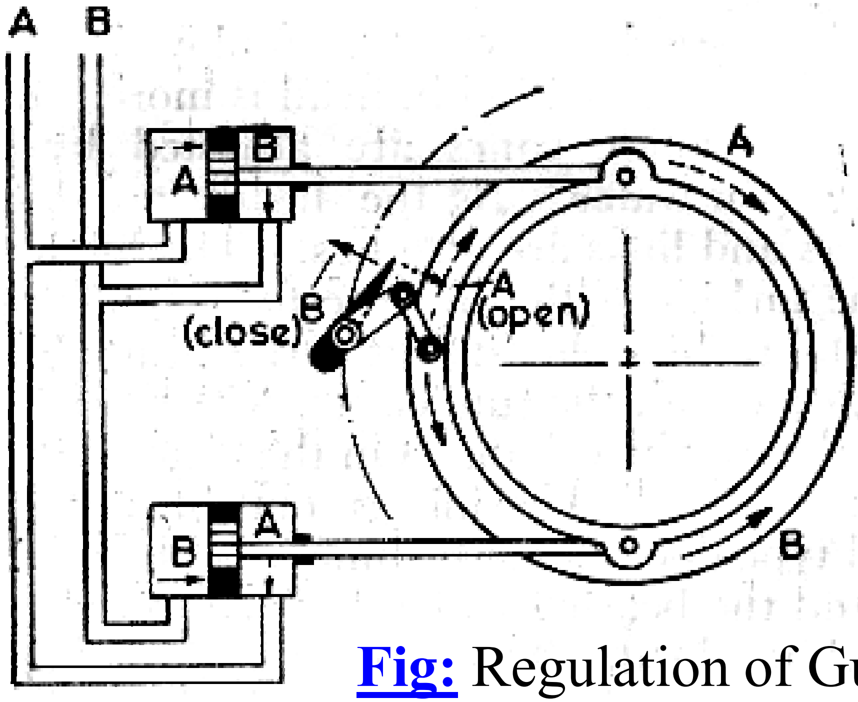


Fig: Regulation of Guide Vanes

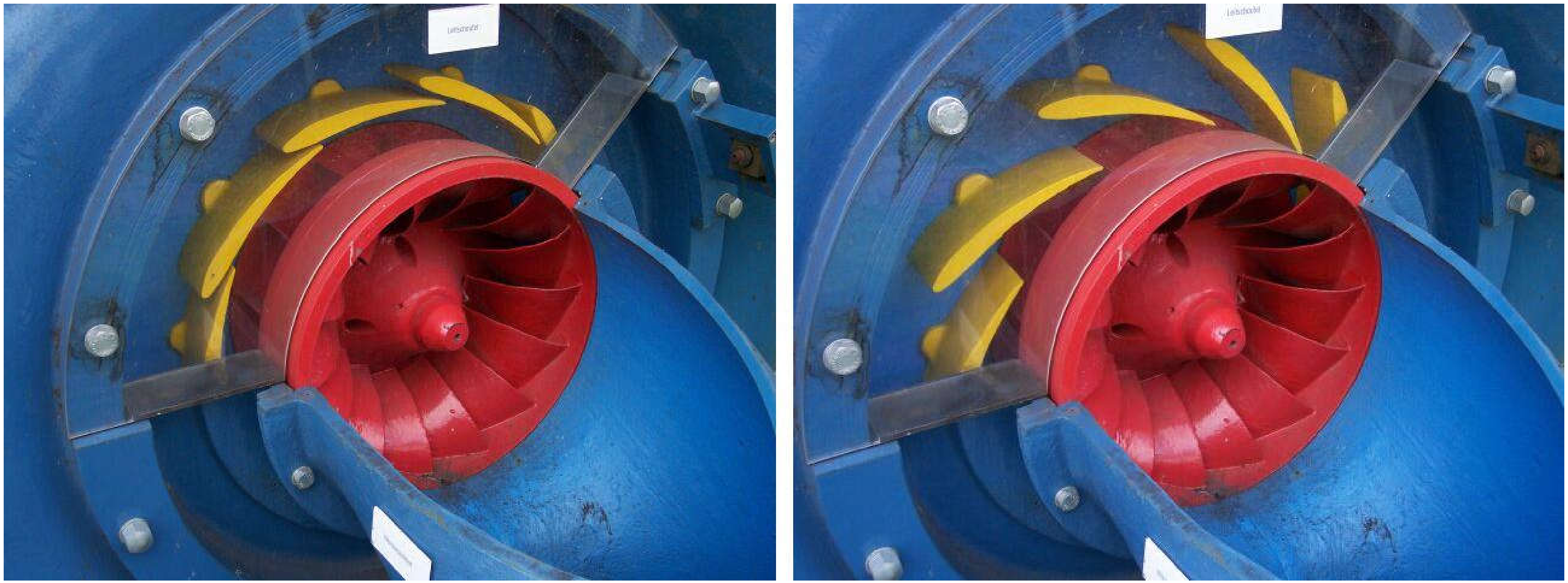


Fig: Francis turbine low and high flow

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.

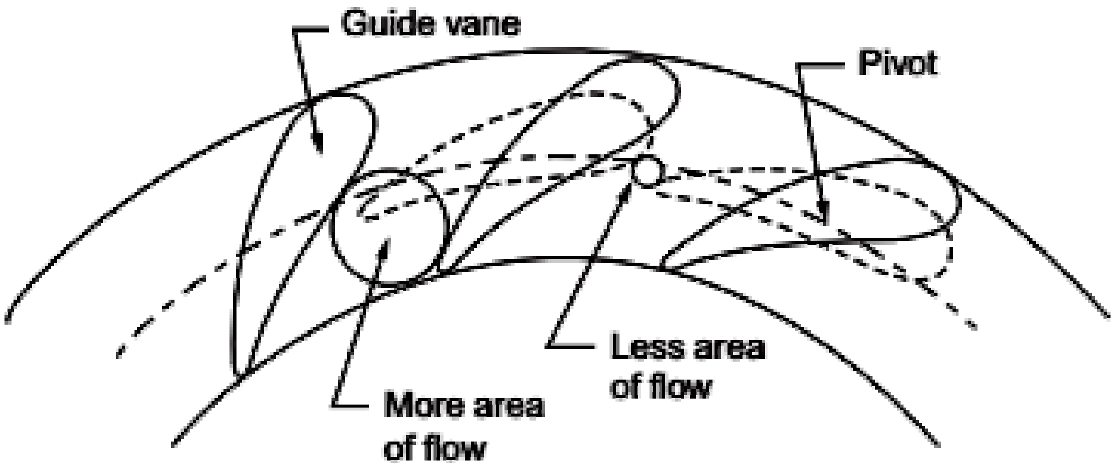
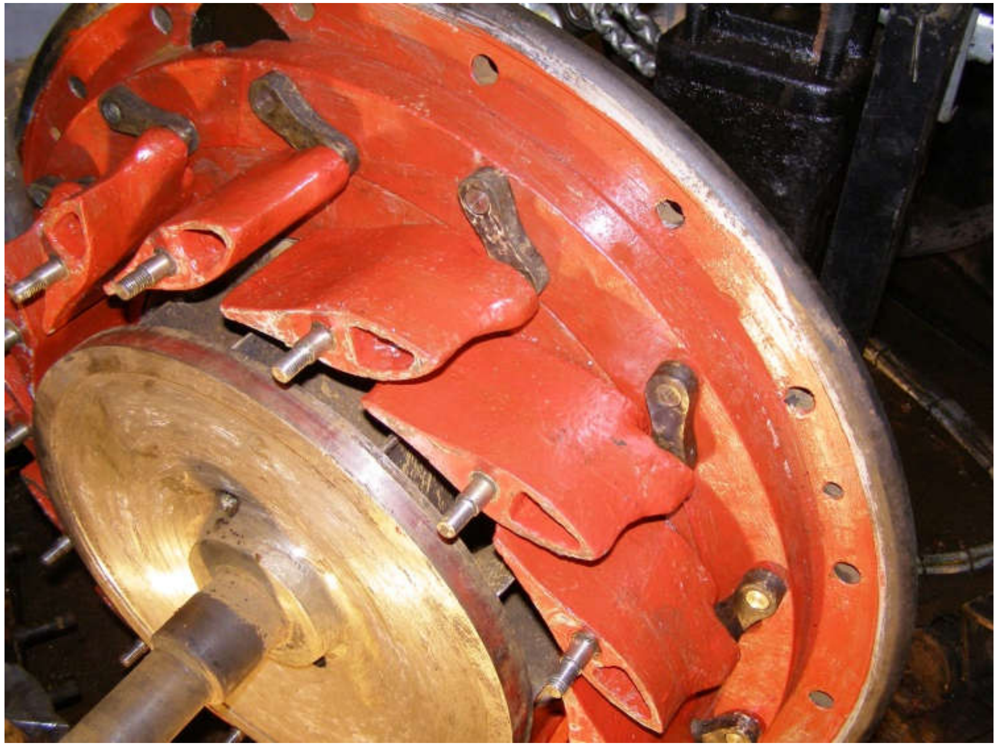


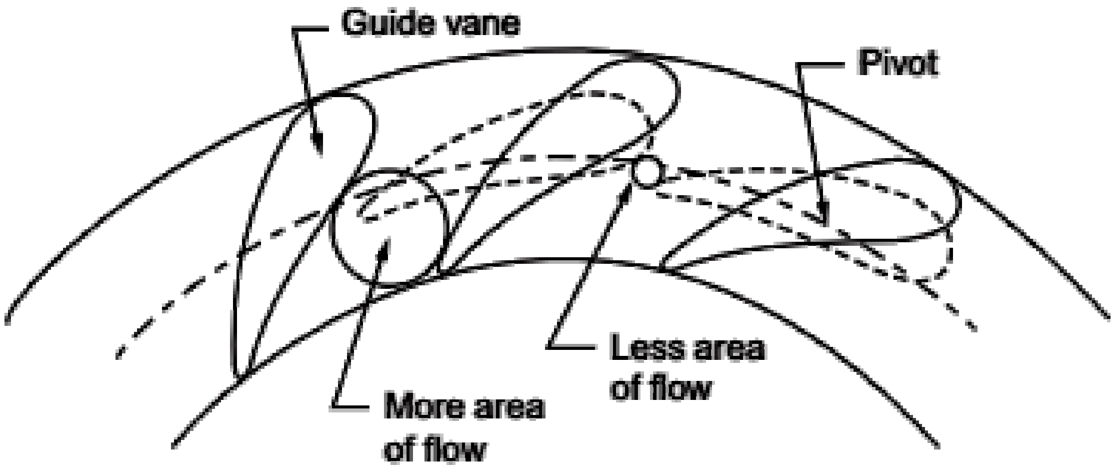
Fig: Guide vanes and guide wheel

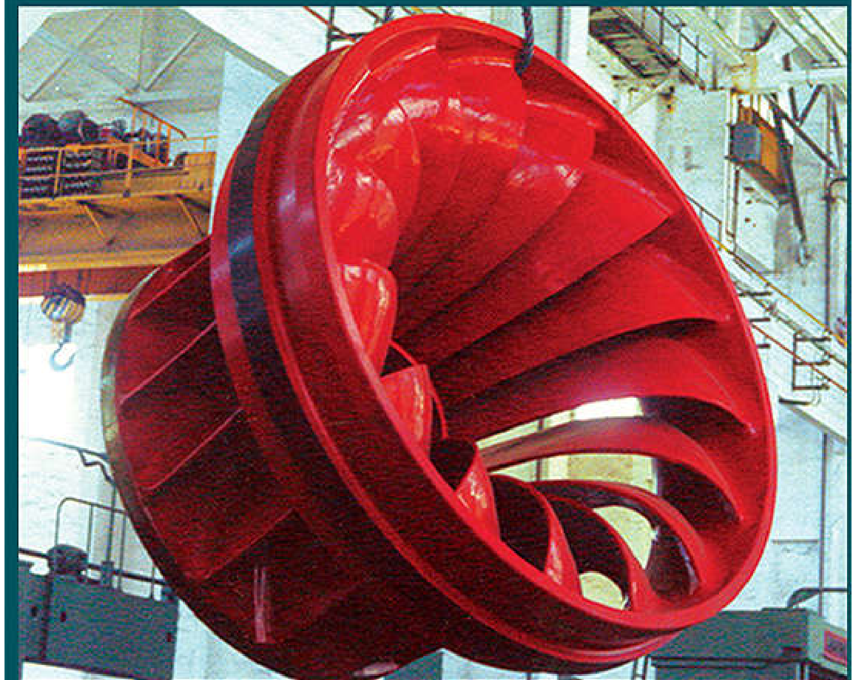
The number of guide blades are generally fewer than the number of blades in the runner. These should also be not simple multiples of the runner blades. The water entering the guide blades are imparted a tangential velocity by the drop in pressure in the passage of the water through the blades. The blade passages act as a nozzle in this aspect.

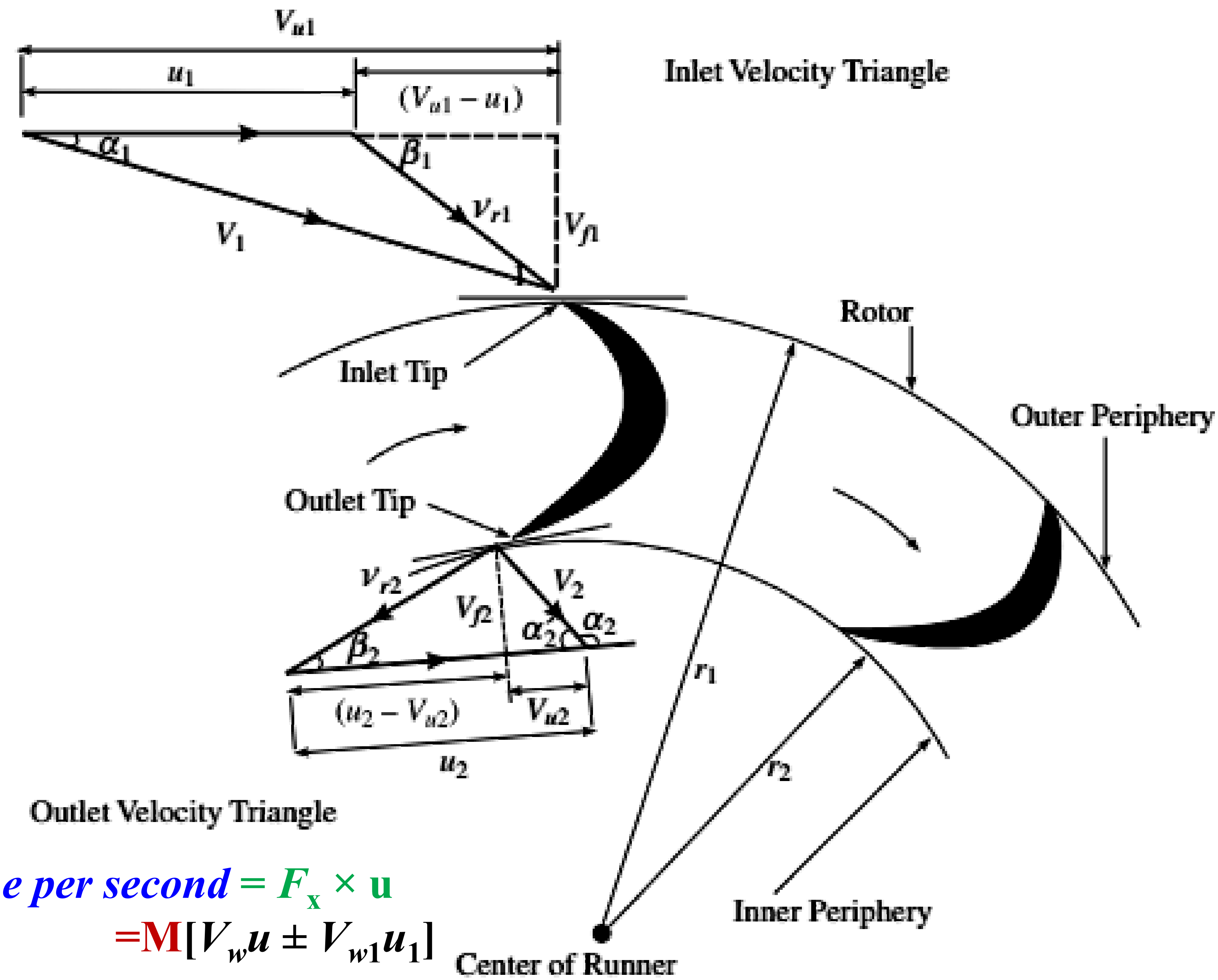
(d) Runner and Turbine Main Shaft: The flow in the runner of a modern Francis turbine is not purely radial but a combination of radial and axial. The flow is inward, that is, from the periphery towards the centre. **The width of the runner depends upon the specific speed**. ***The high specific speed runner is wider than the one which has a low specific speed because the former has to work with a large amount of water***. The runner may be classified as (i) slow; (ii) medium and (iii) fast depending upon the specific speed.

The number of vanes varies from between 16 and 24. The runners are made of cast steel in small units, they are usually are made of stainless steel. The runner may be cast in one piece or it may be made from plates. ***The runner is keyed to the shaft which may be vertical or horizontal.*** The turbine is accordingly specified as vertical or horizontal type.






混流式转轮
Francis turbine runner



Work done per second $= F_x \times u$

$= M[V_w u \pm V_{w1} u_1]$

Fig: Velocity triangles in a Francis-turbine runner

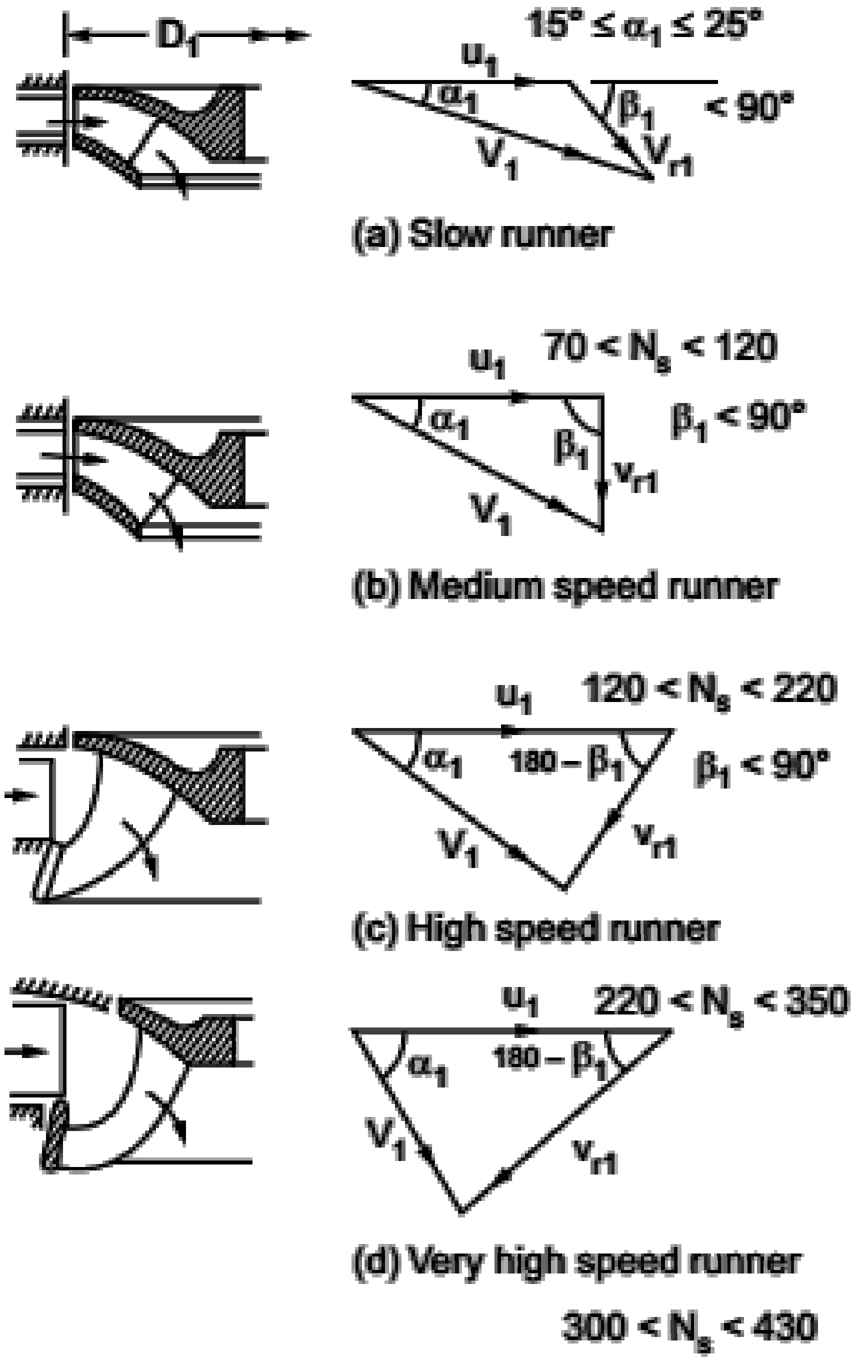
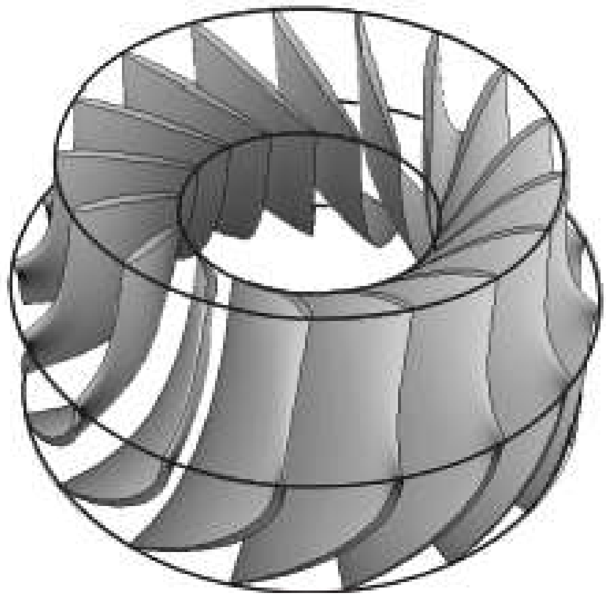
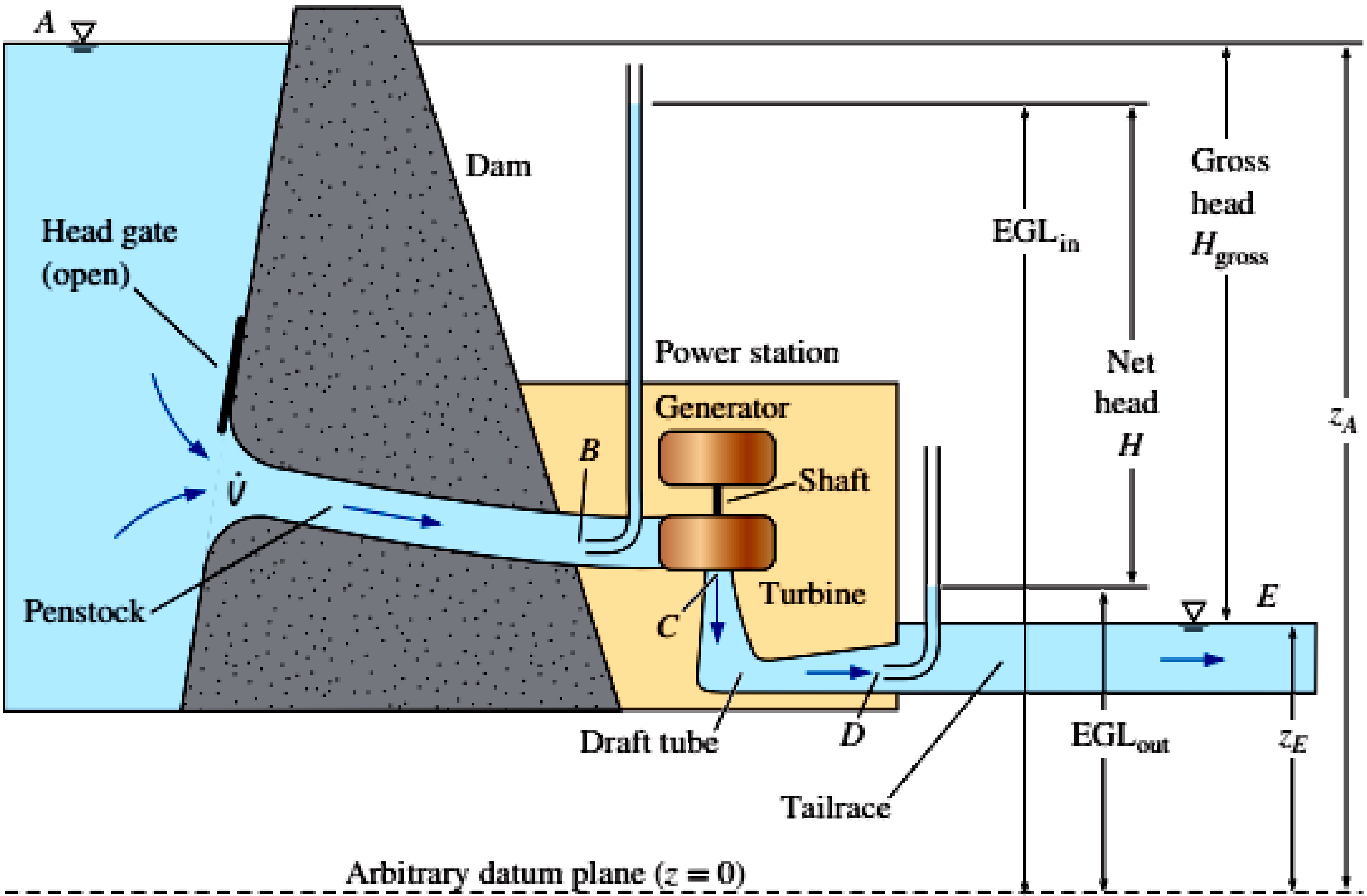


Fig: Variation of runner shapes and inlet velocity triangles with specific speed

The shape of the runner and the corresponding velocity triangles are shown in figure. **A larger exit flow area is made possible by the change of shape from radial to axial flow shape**. This reduces the outlet velocity and thus increases efficiency. **In all cases, the outlet angle of the blades are so designed that there is no whirl component of velocity at exit (V_{w2} = 0) or absolute velocity at exit is minimum**.





A
Dam
Head gate
(open)
$\dot{V}$
Penstock
Power station
Generator
B
Shaft
Turbine
C
Draft tube
D
Tailrace
E
EGL$_{in}$
Net
head
H
Gross
head
H_{gross}
z_A
EGL$_{out}$
z_E
Arbitrary datum plane ($z = 0$)

(e) Draft Tube: As the water flowing through the Francis turbine is under pressure, a closed conduit is required to connect the runner exit to the tail race. This is done with the help of a draft tube. ***The draft rube is a closed conduit which takes the water coming out from the runner exit and discharges it to the tail race***. The draft tube is made of steel plate or concrete. Its cross-section increases in the direction of flow in order to reduce the velocity at the exit of tube. The draft tube is submerged below the lowest tailrace level at its exit.

The functions of the draft tube are as follows :-

(i) If the water is discharged freely from the runner, turbine will work under a head equal to the height of the head race water level above the runner outlet. If an airtight draft tube connects the runner to the tail race, workable head is increased by an amount equal to the height of the runner outlet above tail race. The draft tube will, thus, permit a negative (suction) head to be established at the runner outlet thus making it possible to install the turbine above the tail race without loss of head. This can be explained as follows :-

The pressure in the draft tube at the tail race level is atmospheric. If the cross-section of draft tube is kept uniform, the pressure at the runner outlet is equal to the atmospheric pressure minus the height of runner outlet above the tail race level. The available head, measured from head race level to the discharge side of the turbine, is thus the same as if the turbine were erected at tailrace level and discharged under atmospheric pressure. **The draft tube thus helps to regain the lost static head due to higher level installation of the turbine.**

Thus the reaction turbines may be installed in three ways, (a) at the tail race level, (b) above the tail race level and (c) below tail race level.

Further the turbine installed below the tail race level will reduce the possibility of cavitations because of the absence of negative head.

(ii) The water leaving the runner still possesses a high velocity and this kinetic energy would be lost if it is discharged freely as in a Pelton turbine. **By employing a draft tube of increasing cross section, a large proportion of the kinetic energy of the water at the runner exit is converted (or recovered) into useful pressure energy**. This increases the negative pressure head at turbine runner exit with which the net working head increases and thus increases the output and the efficiency of the turbine.

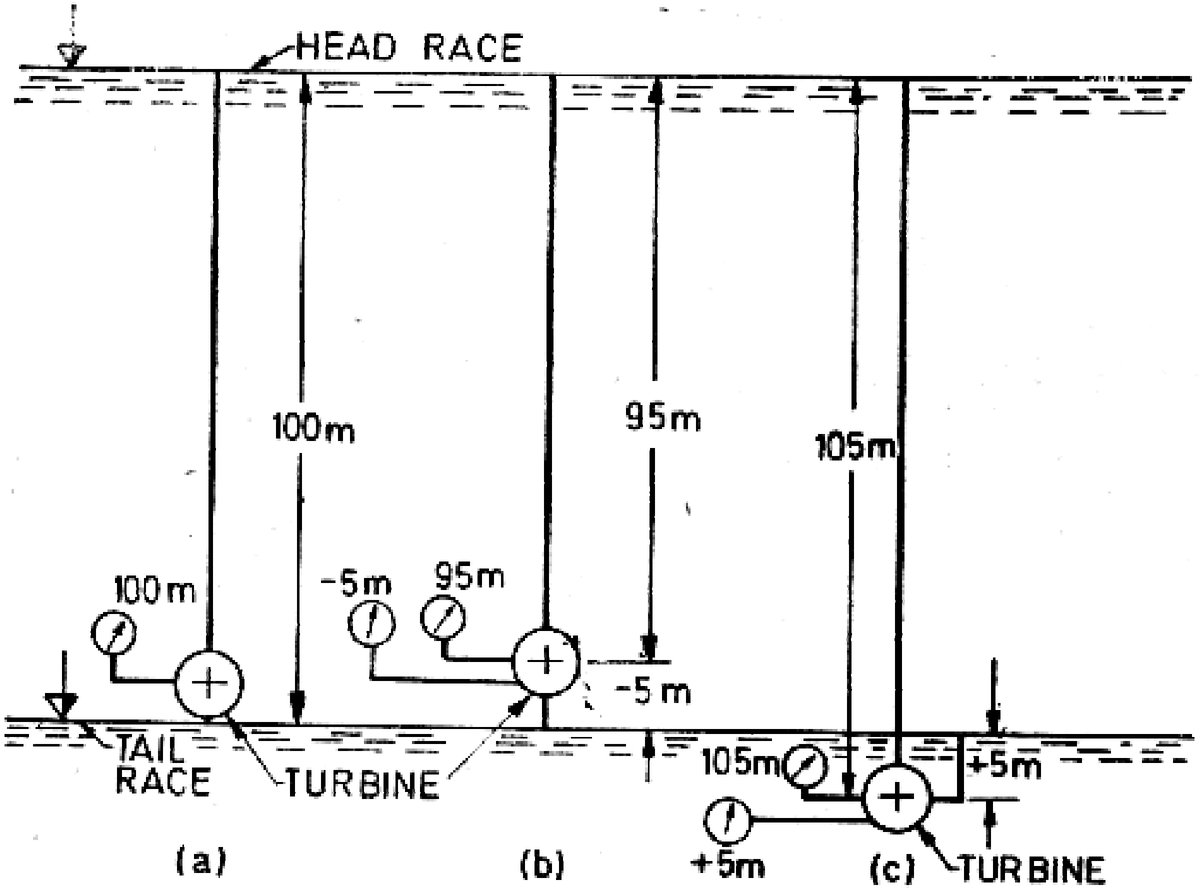


Fig 6.6 Installation of Draft Tube Without Loss of Head

The head recovered by the draft tube will equal the sum of the height of the turbine exit above the tail water level and the difference between the kinetic head at the inlet and outlet of the tube less frictional loss in head.

$$H_d = H + (V_1^2 - V_2^2)/2g - h_f$$

where H_d is the gain in head, H is the height of turbine outlet above tail water level and h_f is the frictional loss of head.

Different types of draft tubes are used as the location demands. These are **(*i*)** Straight diverging tube **(*ii*)** Bell mouthed tube and **(*iii*)** Elbow shaped tubes of circular exit or rectangular exit.

The efficiency of the draft tube in terms of recovery of the kinetic energy is defined us

$$\eta = \frac{V_1^2 - V_2^2}{V_1^2}$$

where V_1 is the velocity at tube inlet and V_2 is the velocity at tube outlet.

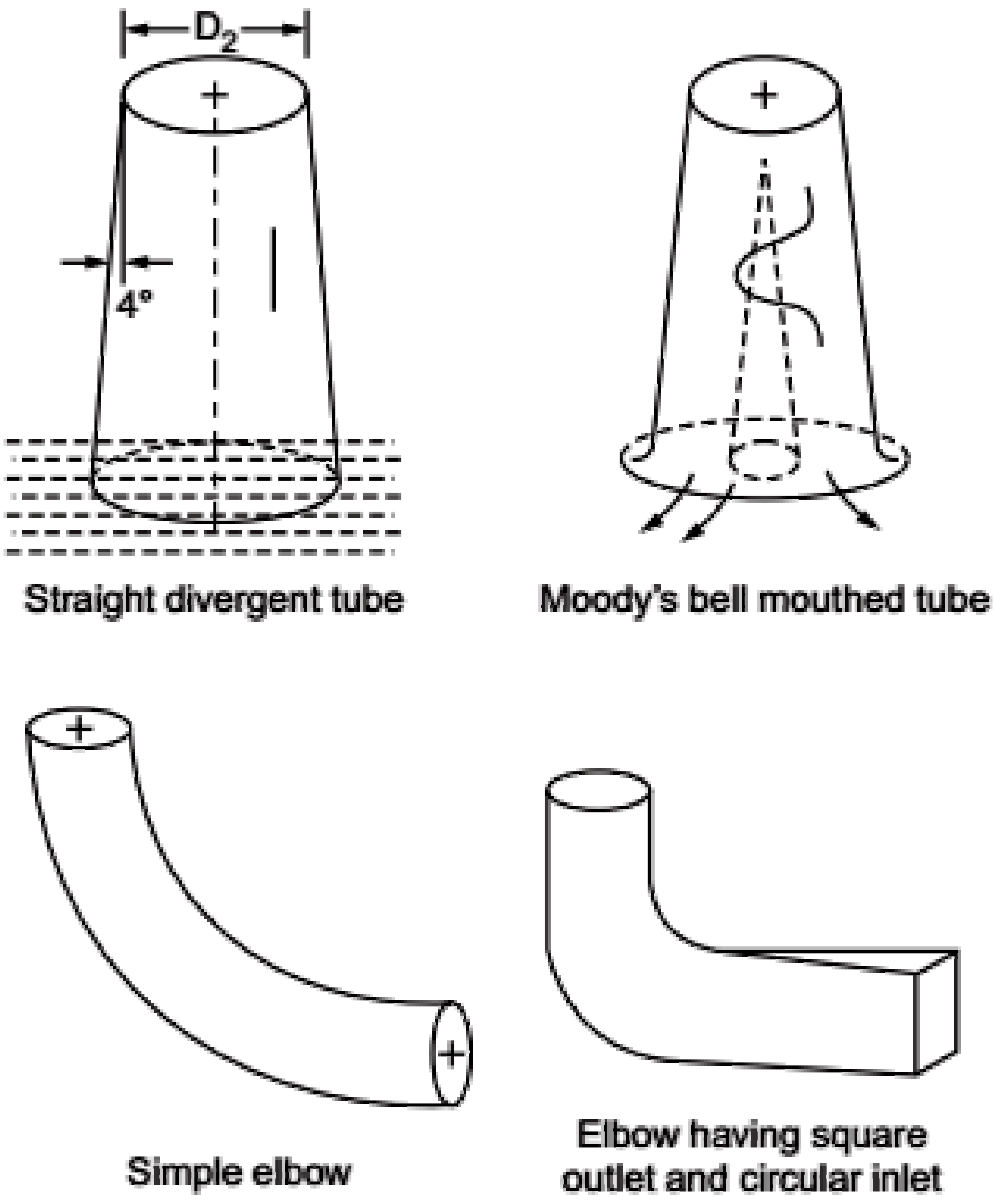


Fig: Various shapes of draft tube.

Expression for Work Done in Francis Turbines

In a reaction turbine, **the net head is equal to the difference of *the total energy of water at the turbine entrance and the draft tube exit*** i.e.

$$H = \left(\frac{p_p}{\gamma} + \frac{V_p^2}{2g} + Z_1\right) - \left(\frac{p_3}{\gamma} + \frac{V_3^2}{2g} + Z_2\right) \quad \text{----- (1)}$$

Where P_p, V_p and Z_1 corresponds to the penstock exit and P_3, V_3 and Z_2 to the draft tube exit.

If the draft tube exit is at tail race level, and the datum is also taken at that level

$$H = \left(\frac{p_p}{\gamma} + \frac{V_p^2}{2g} + Z\right) - \frac{V_3^2}{2g} \quad \text{----- (1b)}$$

where $Z = (Z_1 - Z_2)$.

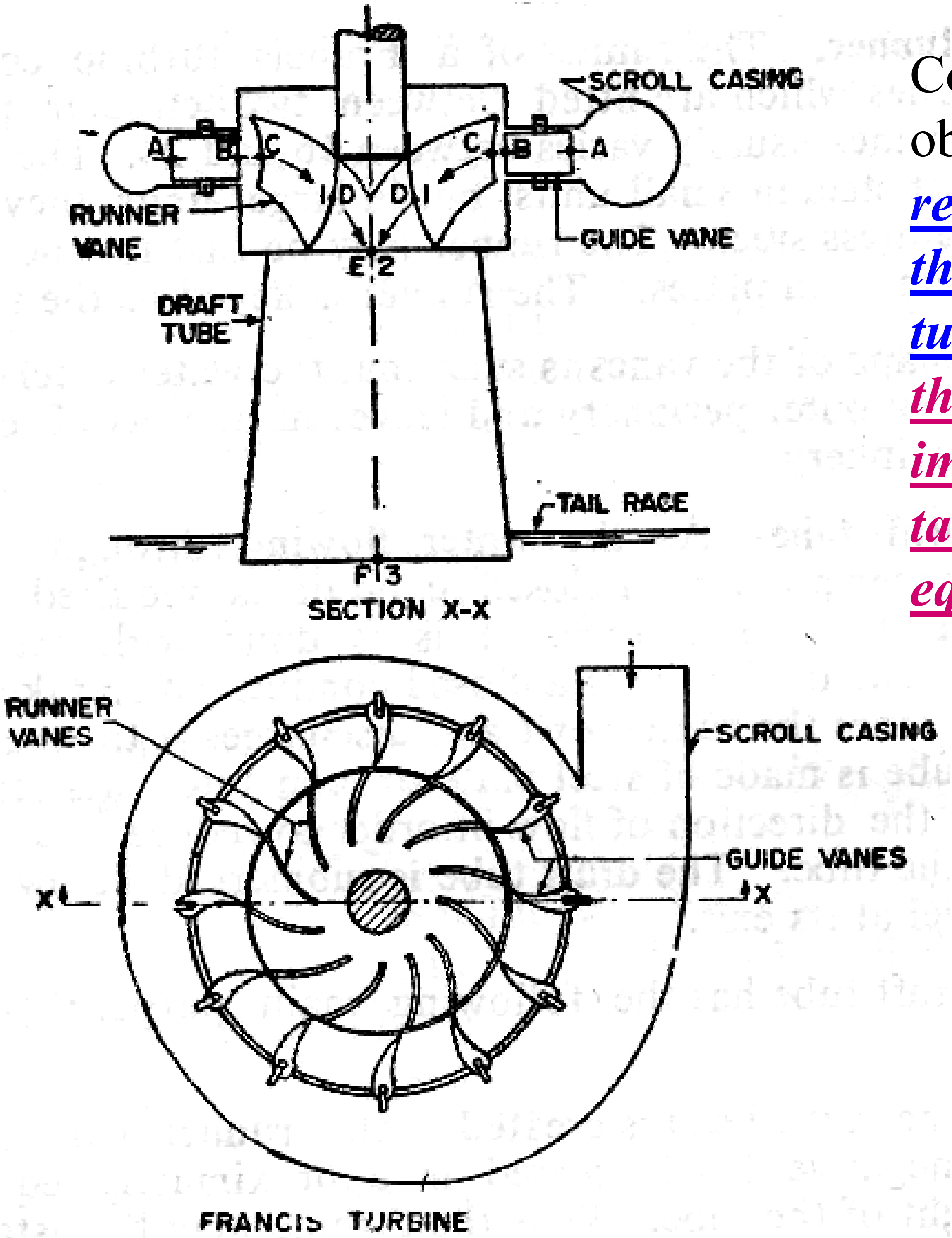
Neglecting the velocity at the draft tube exit (V_3)

$$H = \left(\frac{p_p}{\gamma} + \frac{V_p^2}{2g} + Z\right) \quad \text{----- (1c)}$$

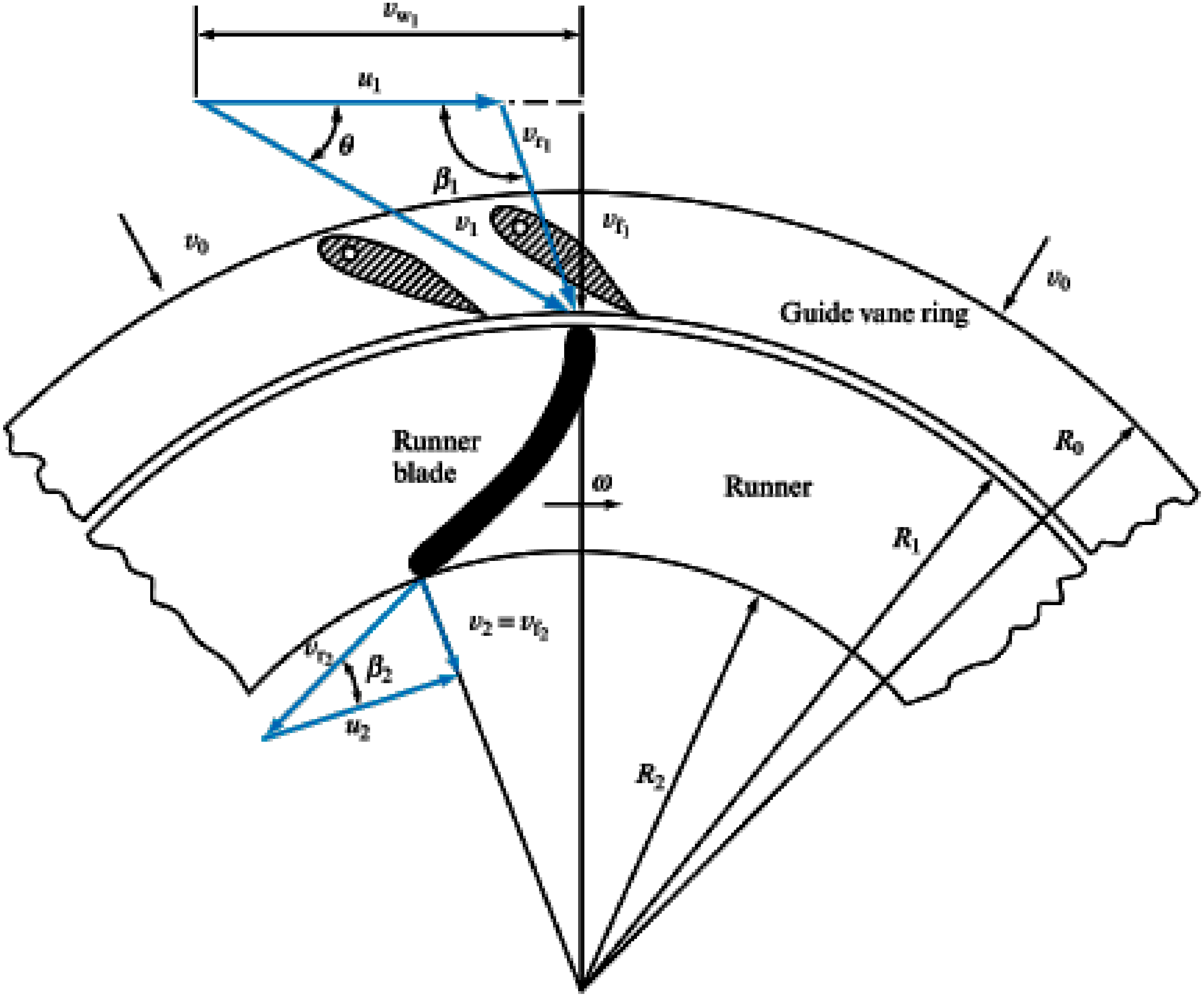
Again, the ***net head developed by impulse turbine*** was

$$H = \frac{p_p}{\gamma} + \frac{V_p^2}{2g} \quad \text{----- (2)}$$

$$H = \frac{V^2}{2g} + \text{Losses in the nozzle}$$



Comparing Eqs. **(1c)** and **(2)**, it is observed that ***the net head on a reaction turbine is more than that on a corresponding impulse turbine.*** ***The difference is due to the fact that the nozzle in an impulse turbine is kept above the tail race level and the head equal to Z is not utilized***.



v_{w_1}
u_1
θ
v_{r_1}
β_1
v_1
v_{r_1}
v_0
v_0
Guide vane ring
Runner blade
ω
Runner
R_0
R_1
$v_2 = v_{r_2}$
v_{r_2}
β_2
u_2
R_2

The general expression for the work done is given by, according to the Euler momentum equation:

$$\textbf{Work done} = M\,(V_w u \pm V_{w1} u_1)$$

(1) The maximum output under given conditions is obtained when $V_{w1} = 0$. Thus the maximum work done is given by

$$\textbf{Work done} = M\,(V_w u) \qquad \text{----- (3)}$$

The discharge in this case is radial. For radial discharge, the absolute velocity at exit is radial. If "*H*" is the net head,

$$\textbf{Input to the turbine} = MgH$$

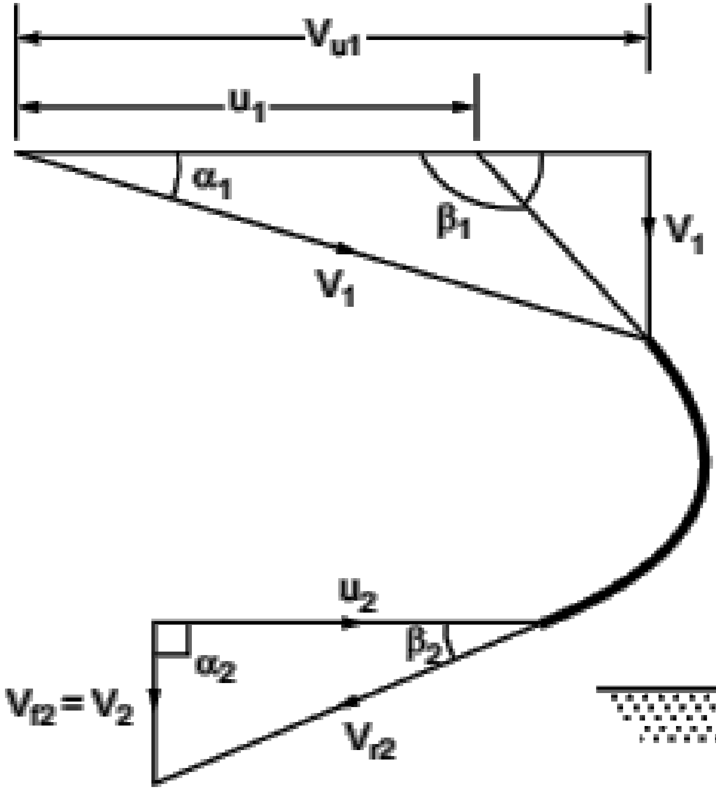
$$\text{Hydraulic efficiency} = \frac{M(V_w\,u)}{MgH}$$

$$\eta_h = \frac{V_w\,u}{gH} \qquad \text{----- (4)}$$

(2) However, **if the velocity of whirl at the exit is not zero**, the hydraulic efficiency is given by

$$\text{Hydraulic efficiency} = \frac{V_w\,u \pm V_{w1}\,u_1}{gH} \qquad \text{----- (5)}$$

A typical velocity diagrams at inlet and outlet are shown in Figure.



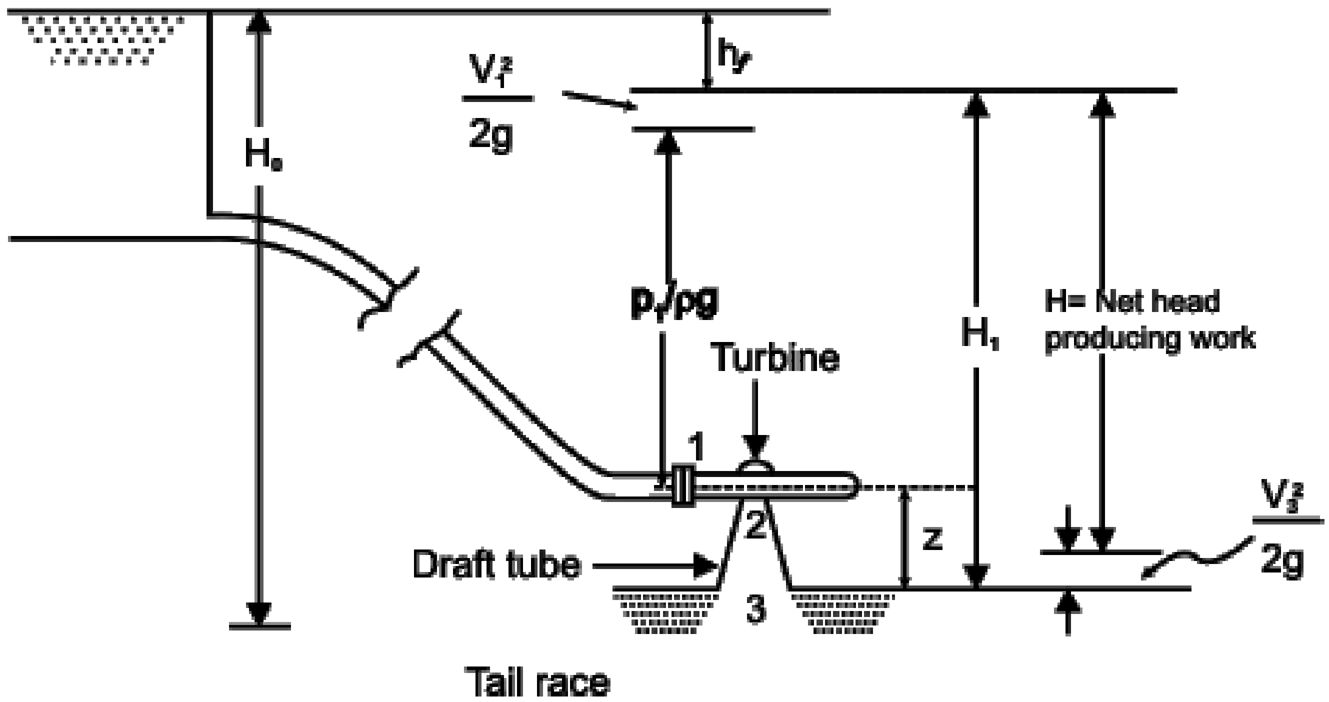
$$n_s = n\frac{\sqrt{P}}{H^{\frac{5}{4}}}$$

Where: n_s = specific speed

n= revolution per minute

P=Power (K.W)

H= head (m)



The ***hydraulic efficiency of the Francis turbine varies from 0.85 to 0.90***.
As in the case of Pelton wheel,

Mechanical efficiency, $(\eta_m) = \dfrac{\text{S.P.}}{\text{Power developed}} = \dfrac{\text{S.P.}}{P}$

Overall efficiency, $(\eta_o) = \dfrac{\text{S.P.}}{\text{Water power}} = \dfrac{\text{S.P.}}{P_t}$

and $\eta_o = \eta_m \times \eta_h$

The ***overall efficiency varies from 0.80 to 0.90***.
The **mass of water (M)** may be obtained from the flow area and the velocity of flow. Let,

N = Number of vanes

B, D, and t = Width, diameter and thickness of vanes at the inlet.

B_1, D_1, and t_1 = Width, diameter and thickness of vanes at the outlet.

Therefore, **area of flow at inlet = $(\pi D - Nt)B = k\pi DB$**

Where 'k' is a factor known as the ***vane thickness factor***. ***Its value is slightly less than unity, but it is usually taken as unity***. In other words, **the vane thickness is neglected.**

Discharge = Area of flow x Velocity of flow

$$Q = k\pi DB \times V_f$$

The mass of water striking per second is given by

$$M = \rho Q = \rho\,(k\pi DB) \times V_f = \rho\,(k_1\pi D_1 B_1) \times V_{f1}$$

Working Proportions of a Francis Turbines

(1) Ratio of width to diameter (B/D): The ratio of width (B) to the diameter (D) of the wheel is represented by n. Thus

$$n = B/D$$

The value of **n varies from 0.1 to 0.45.**

(2) Flow Ratio (ψ): The flow ratio (ψ) is the ratio of the velocity of flow at the inlet to the theoretical jet velocity. Thus

$$\psi = \frac{V_f}{\sqrt{2gh}}$$

The value of **ψ, ranges from 0.15 to 0.30**.

(3) Speed Ratio (ϕ): Speed ratio is the ratio of the peripheral speed at inlet to the theoretical jet ratio velocity.

$$\phi = \frac{u}{\sqrt{2gh}}$$

The value of **ϕ ranges from 0.6 to 0.90**.

Design of Runner:

Design of runner consists of determining the size of runner and vane angles. The design of Francis turbine runner is carried out as follows. It is ***assumed that the available net head (H) and the required shaft power are given***.

(1) Assume suitable values of η_o, η_h, n, ψ, k and k_1.

(2) Calculate the required discharge (Q) for the given **S.P.** from the equation.

$$\text{S.P.} = \eta_0 (WH)$$
$$\text{S.P.} = \eta_0 (\gamma Q H)$$

(3) Obtain the velocity of flow at the inlet from Eqe.

$$V_f = \frac{Q}{k\pi DB}$$
$$V_f = \frac{Q}{k\pi n D^2}$$

$$\psi\sqrt{2gH} = \frac{Q}{k\pi n D^2}$$
$$D = \left[\frac{Q}{\psi(\sqrt{2gH})(k\pi n)}\right]^{1/2}$$

The width B is then obtained using Eq. $B = nD$

(4) Obtain the rim velocity u from the relation:

$$u = \frac{\pi D N}{60}$$

(5) Obtain the velocity of whirl at the inlet from Eq. $V_w = \frac{\eta_h g H}{u}$

(6) Obtain ***the guide vane angle α*** and the ***runner vane angle θ*** from the following relations obtained from the inlet velocity triangle .

$\tan \alpha = \frac{V_f}{V_w}$ and $\tan \theta = \frac{V_f}{V_w - u}$

(7) The ***runner diameter at the outlet D_1 is approximately one-half the diameter at the inlet***. Thus

$D_1 = D/2$ and $u_1 = u/2$

(8) The velocity of flow at the exit may be obtained using Eqs'

$$\frac{V_f}{V_{f1}} = \frac{k_1 D_1 B_1}{kBD}$$

(9) The runner vane angle at exit is obtained from the outlet velocity diagram.

$$\tan \phi = \frac{V_{f1}}{u_1}$$

(10) The number of vanes varies from 16 than to 24. ***The number of vanes should be either one more or one less than the number of guide vanes to avoid periodic impulse.***

Problem:

A Francis turbine runner is to be designed for the following data:

Net head (H) = 60m, shaft power = 367.875kW (500h.p.), speed (N) = 600 rpm, hydraulic efficiency = 85%, overall efficiency = 80%, flow ratio (ψ) = 0.15 and breadth ratio (n) = 0.10.

Assume the inner diameter as one-half the outer diameter. The velocity of flow is constant throughout. The discharge is radial. Neglect vane thickness.

Solution:

$$\eta_o = \frac{Shaft\ power\ (S.P.)}{Water\ power\ (\gamma QH)}$$

$$Q = \frac{(S.P.)}{\eta_0 \gamma H}$$

$$= \frac{367 \cdot 875}{0 \cdot 80 \times 9 \cdot 81 \times 60} = 0 \cdot 78\ m^3/sec$$

Flow Ratio (ψ)

$$\psi = \frac{V_f}{\sqrt{2gh}}$$

$$V_f = \psi \sqrt{2gH}$$

$$= 0 \cdot 15 \sqrt{2 \times 9 \cdot 81 \times 60} = 5 \cdot 15\ m/sec$$

Again, $$\psi\sqrt{2gH} = \frac{Q}{k\pi n D^2}$$

$$0{\cdot}15\sqrt{2\times 9{\cdot}81\times 60} = \frac{0{\cdot}78}{1\times\pi\times 0{\cdot}10\times D^2}$$

$$D = 0{\cdot}695 \text{ m}$$

$$D_1 = 0{\cdot}695/2 = 0{\cdot}348 \text{ m}$$

The width: $B = nD = 0.1 \times 0.695 = 0.0695 \text{ m} = 7 \text{ cm (say)}$

Again, $$B_1 = \frac{DB}{D_1} = \frac{7}{0{\cdot}5} = 14 \text{ cm}$$

Since V_f is constant throughout.

Rim velocity:

$$u = \frac{\pi DN}{60} = \frac{\pi\times 0{\cdot}695\times 600}{60} = 21{\cdot}8 \text{ m/sec}$$

and, $$u_1 = 21{\cdot}8/2 = 10{\cdot}9 \text{ m/sec.}$$

The velocity of whirl

$$V_w = \frac{\eta_h g H}{u}$$

$$= \frac{0{\cdot}85\times 9{\cdot}81\times 60}{21{\cdot}8} = 22{\cdot}9 \text{ m/sec}$$

The guide vane angle α

$$\tan\alpha = \frac{V_f}{V_w} = \frac{5{\cdot}15}{22{\cdot}9} = 0{\cdot}224$$

$$\alpha = 12{\cdot}7^\circ.$$

The runner vane angle θ

$$\tan\theta = \frac{V_f}{V_w - u} = \frac{5{\cdot}15}{22{\cdot}9 - 21{\cdot}80} = 4{\cdot}68$$

$$\theta = 77{\cdot}9^\circ.$$

The runner vane angle at exit ϕ

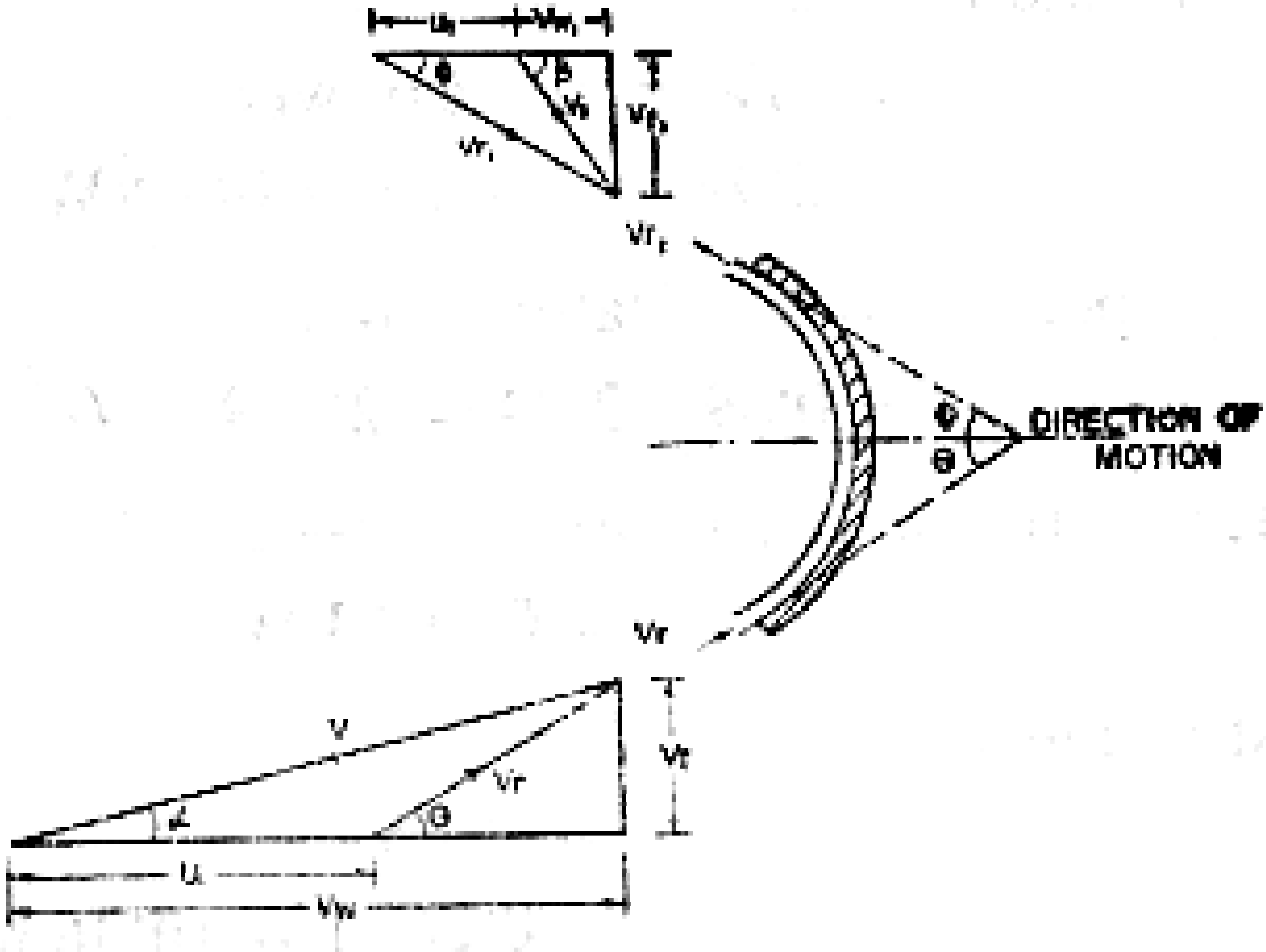
$$\tan\phi = \frac{V_{f1}}{u_1} = \frac{5{\cdot}15}{10{\cdot}9} = 0{\cdot}472$$

$$\phi = 25{\cdot}3^\circ$$

Radial Flow turbine

Depending upon the direction of flow, a radial flow turbine may be inward flow or outward flow. The radial flow turbines work on the principle of free jet when the jet strikes at one tip of the vane. The work done on the turbine is given by the Euler momentum equation

$$\text{Work done} = M(V_w u \pm V_{w1} u_1)$$



Vr
Vr
Vw
V
Vr
Vf
DIRECTION OF
MOTION

The work done is also equal to the difference of kinetic energy at the inlet and outlet. Thus

$$\text{Work done} = \frac{M}{2}(V^2 - V_1^2)$$

Another expression for the work done may be obtained by using velocity triangles.

From **the inlet velocity triangle**

$$V^2 = V_w^2 + V_f^2$$

$$V^2 = V_w^2 + V_r^2 - (V_w - u)^2$$

$$= V_r^2 - u^2 + 2uV_w$$

------ (a)

From **the outlet velocity triangle**

$$V_{r1}^2 = V_{f1}^2 + (u_1 + V_{w1})^2$$

$$V_{r1}^2 = V_{f1}^2 + u_1^2 + V_{w1}^2 + 2u_1 V_{w1}$$

$$= (V_1^2 - V_{w1}^2) + u_1^2 + V_{w1}^2 + 2u_1 V_{w1}$$

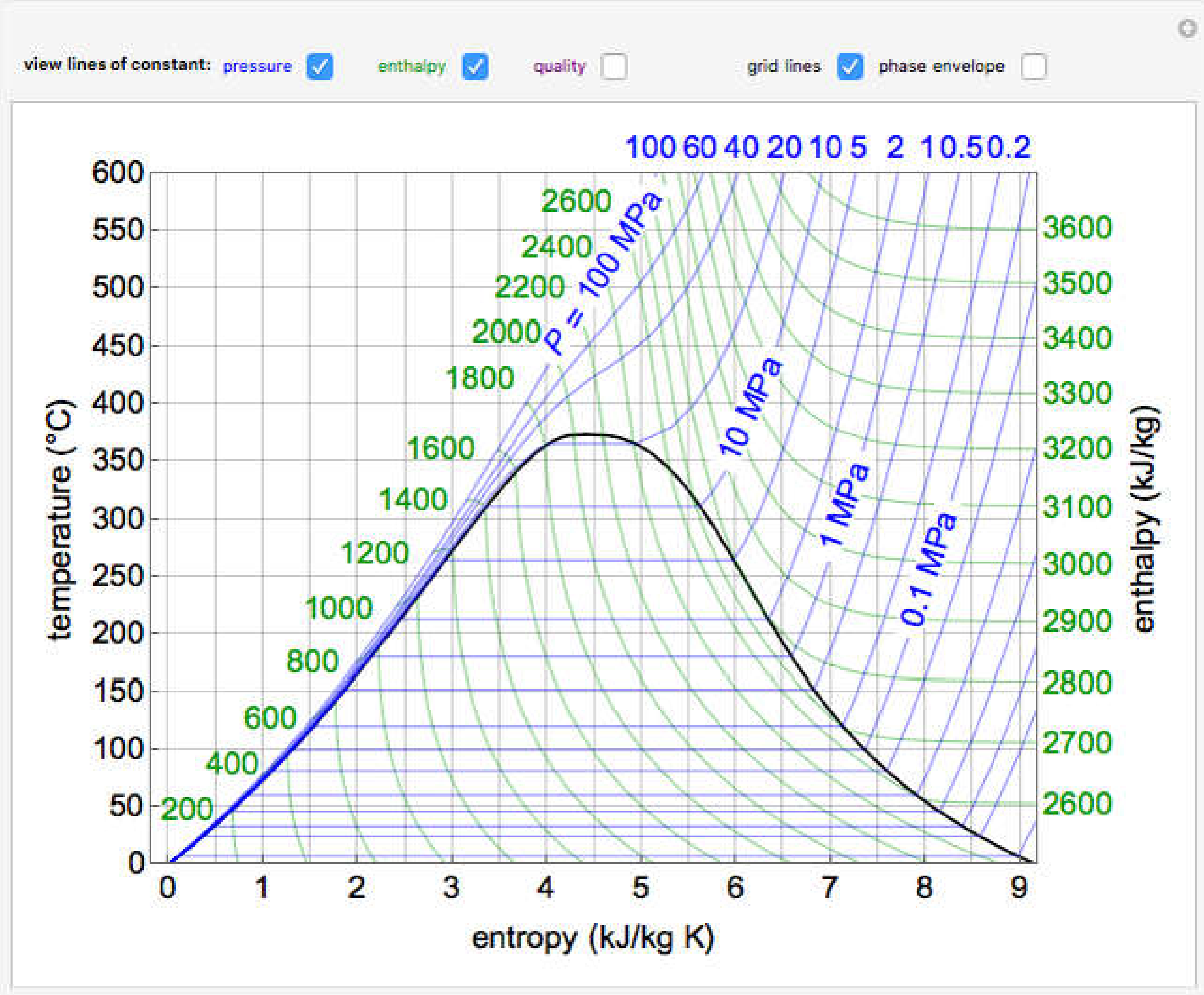
$$V_{r1}^2 = V_1^2 + u_1^2 + 2u_1 V_{w1}$$

------ (b)

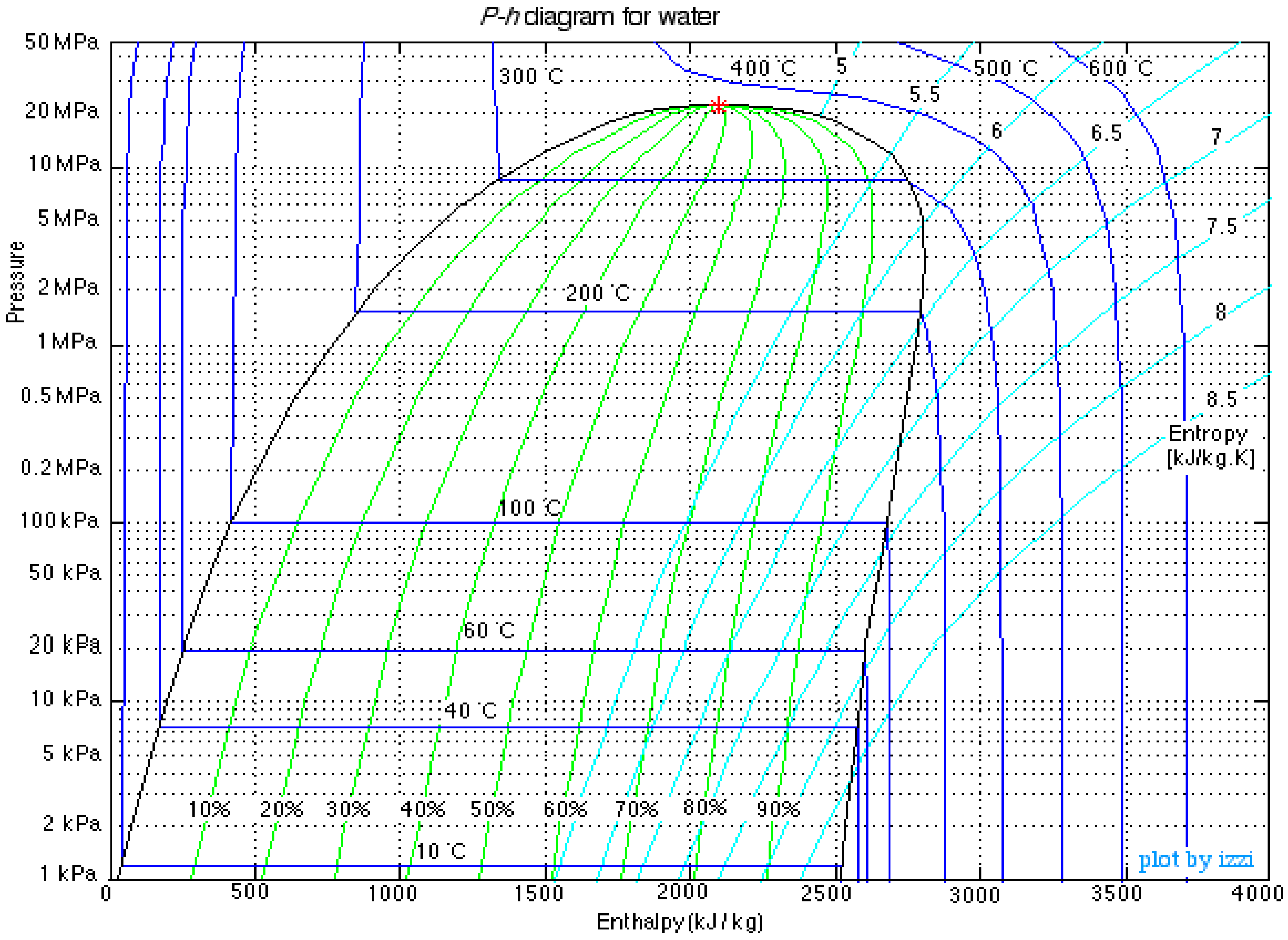
Substituting the values of $V_w u$ and $V_{w1} u_1$ from Eqs (a) and (b):

$$\text{Work done} = M\left[\left(\frac{V^2 - V_r^2 + u^2}{2}\right) + \left(\frac{V_{r1}^2 - V_1^2 - u_1^2}{2}\right)\right]$$

$$= M\left[\left(\frac{V^2 - V_1^2}{2}\right) + \left(\frac{u^2 - u_1^2}{2}\right) + \left(\frac{V_{r1}^2 - V_r^2}{2}\right)\right] \quad \text{------ (c)}$$



view lines of constant: pressure enthalpy quality grid lines phase envelope
100 60 40 20 10 5 2 1 0.5 0.2
P = 100 MPa
10 MPa
1 MPa
0.1 MPa
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
600
550
500
450
400
350
300
250
200
150
100
50
0
temperature (°C)
0 1 2 3 4 5 6 7 8 9
entropy (kJ/kg K)
3600
3500
3400
3300
3200
3100
3000
2900
2800
2700
2600
enthalpy (kJ/kg)



P-h diagram for water
50 MPa
20 MPa
10 MPa
5 MPa
2 MPa
1 MPa
0.5 MPa
0.2 MPa
100 kPa
50 kPa
20 kPa
10 kPa
5 kPa
2 kPa
1 kPa
Pressure
0
500
1000
1500
2000
2500
3000
3500
4000
Enthalpy (kJ / kg)
300 °C
400 °C
500 °C
600 °C
200 °C
100 °C
60 °C
40 °C
10 °C
10%
20%
30%
40%
50%
60%
70%
80%
90%
5
5.5
6
6.5
7
7.5
8
8.5
Entropy
[kJ/kg.K]
plot by izzi