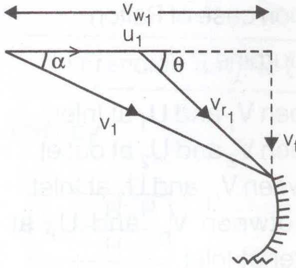
14. TURBINES

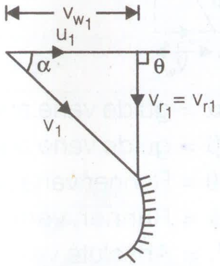
VELOCITY TRIANGLE

1. At Inlet

(a)



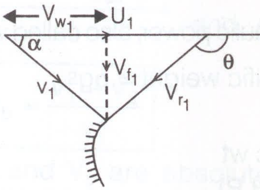
(b)



- $\theta < 90°$
- Forward vanes
- Common case of
- Francis Turbine
- $\boxed{\tan\theta = \dfrac{V_{f_1}}{V_{w_1} - U_1}}$

- $\theta = 90°$
- Radial vanes
- $U_1 = V_{w_1}$
- $V_{F_1} = V_{r_1}$
- $\boxed{\tan\theta = \dfrac{V_{f_1}}{V_{w_1}}}$

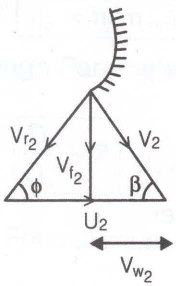
(c)



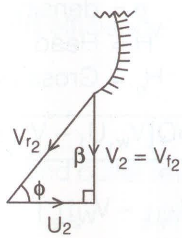
- $\theta > 90$
- Backward vanes
- $\tan(180° - \theta) = \dfrac{V_{f_1}}{U_1 - V_{w_1}}$

2. At Outlet

(a)



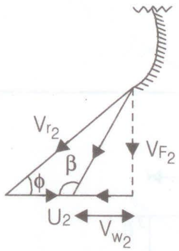
(b)



- $\beta < 90°$
- V_{W_2} is in the direction of U_2.

- $\beta = 90°$
- Common case of Francies turbine
- Use fuel for maximum efficiency.

(c)



- $\beta > 90°$
- Common case of Pelton wheel turbine

Here, α = guide vane angle between V_1 and U_1 at Inlet.
β = guide vane angle between V_2 and U_2 at outlet.
θ = Runner vane angle between V_{r_1} and U_1 at Inlet.
ϕ = Runner vane angle between V_{r_2} and U_2 at outlet.
V_1 = Absolute velocity of water at Inlet.
V_2 = Absolute velocity of water of outlet.

u_1 and u_2 are relative velocities at inlet and outlet respectively.
V_1 and V_2 are tangential also called ci... rential velocity at inlet and outlet respectively.

FRANCIS TURBINE

1. Powers

(i) $\boxed{H.P = w...}$

... = Hydraulic power, also called water power

w = Specific weight = ρg

...enoted by 'γ'

γ = unit wt or specific wt
1 H.P = 746 walt (British H.P)
1 H.P = 736 walt (Metric H.P)
ρ = density, Q = Discharge,
H = Head = $H_g - H_F$
H_g = Gross Head, H_F = Fricloss.

(ii) $\boxed{R.P = \rho Q[V_{w_1}U_1 - V_{w_2}U_2]}$

$= \frac{wQ}{g}[V_{W_1U_1} - V_{W_2U_2}]$ R.P = Runner power

For maximum efficiency $V_{W_2} = 0$.

(a) For maximum efficiency

$\boxed{R.P = \frac{wQ}{g}(V_{W_1U_1})}$ V_{w_1} = Whirl velocity at Inlet.
V_{w_2} = Whirl velocity at outlet.

(iii) $\boxed{S.P = \tau\omega}$ S.P = R.P – Mechanical friction losses

Where, τ = Torque produced by shaft
ω = Angular velocity of shaft
S.P = Shaft power.

NOTE: In Francies Turbine H.P > R.P > S.P

2. **Efficiency**

(i) $\boxed{\eta_h = \frac{R \cdot P}{H \cdot P}}$ $\Rightarrow \eta_h = \frac{(V_{w_1u_1} - V_{w_2u_2})}{gH}$
η_h = Hydraulic efficiency

(ii) $\boxed{\eta_m = \frac{S \cdot P}{R \cdot P}}$ $= 1 - \frac{\text{loss}}{R \cdot P}$
η_m = Mechanical efficiency.

(iii) $\boxed{\eta_0 = \frac{S \cdot P}{H \cdot P}}$ $\eta_o = \eta_m \cdot \eta_h$.
η_o = Overall efficiency.

(iv) $\boxed{\eta_{vol} = \frac{Q - \Delta Q}{Q}}$ η_{vol} = Volumetric efficiency
Q = Discharge (m^3/s)
and ΔQ = Charge in discharge (m^3/s).

(v) $\boxed{\eta_b = \frac{v_1^2 - v_2^2}{v_1^2}}$ η_b = Blade efficiency.

V_1 and V_2 are absolute velocity of water at inlet and outlet respectively.

(vi) $\boxed{\eta_o = \eta_h \cdot \eta_m \cdot \eta_v \cdot \eta_b}$ η_o = Overall efficiency.

3. **Design Parameter of Francis Turbine**

(i) $\boxed{\frac{D_1}{D_2} \sim 2}$ D_1 and D_2 are dia of inlet and outlet respectively.

(ii) For maximum efficiency, $V_{w_2} = 0$.

(iii) $\phi = \frac{u_1}{\sqrt{2gH}} \simeq 0.6 + 0.9$

ϕ = Speed ratio

U_1 = Tangential also called circumferential velocity of inlet.

H = Head = $H_g - H_f$

$\sqrt{2gH}$ = Spout velocity.

(iv) $\psi = \frac{V_{f_1}}{\sqrt{2gH}} \simeq 0.15 \text{ to } 0.30$

ψ = Flow ratio

V_{F_1} = Flow velocity of inlet.

(v) $\eta = \frac{B_1}{D_1} \simeq 0.1 \text{ to } 0.45$

η = width ratio

(vi) $A_{f_1} = (1-k)\pi D_1 B_1$ $\quad A_{f_2} = (1-k)\pi D_2 B_2$

A_{f_1} and A_{f_2} are area of flow at inlet and outlet respectively.

K = Vane thickness coefficiency $\simeq 5\%$

D_1 and D_2 are diameter at inlet and outlet respectively.

B_1 and B_2 are width of plate at inlet and outlet respectively.

(vii) $R = \frac{\frac{P_1}{\gamma} - \frac{P_2}{\gamma}}{\left(\frac{V_{w_1}u_1 - V_{w_2}u_2}{g}\right)}$ $\quad R = \frac{\frac{P_1}{\gamma} - \frac{P_2}{\gamma}}{\left(\frac{V_{w_1}u_1}{g}\right)}$ → For maximum efficiency

Here, R = Degree of reaction

γ = Unit weight or specific wt = rg

ρ = Density

Also, $R = 1 - \left(\frac{V_1^2 - V_2^2}{2V_{w_1} \cdot u_1}\right)$

(viii) $H = \frac{V_2^2}{2g} + \frac{V_{w_1}U_1}{g}$ (Master Formula) where, H = Head

Assumption, $V_{w_2} = 0, \frac{P_2}{\rho g} = 0,$

4. Model Relationship for Turbine

Dimensional Parameter	***Diamensionless Parameter***
1. $N_S = \frac{N\sqrt{P}}{(H)^{5/4}}$	1. $N_S = \frac{\omega\sqrt{\frac{P}{\rho}}}{(gH)^{5/4}}$
2. $C_H = \frac{H}{N^2D^2}$	2. $C_H = \frac{gH}{\omega^2D^2}$
3. $C_Q = \frac{Q}{ND^3}$	3. $C_Q = \frac{Q}{\omega D^3}$
4. $C_P = \frac{P}{N^3D^5}$	4. $C_P = \frac{P}{\rho\omega^3D^5}$

Here, N_S = Specific speed $\quad C_H$ = Head coefficient

C_Q = Discharge coefficient $\quad C_P$ = Power coefficient

$g \rightarrow$ Acln. due to gravity = 9.81 m/s².

$w \rightarrow$ Angular speed = $\frac{2\pi N}{60}$ $\quad N \rightarrow$ No. of revolution/minute

$H \rightarrow$ Head (m) $\quad P \rightarrow$ Pressure (N/m²)

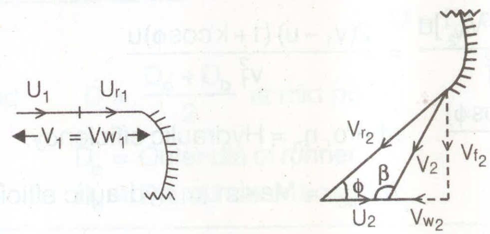
$\rho \rightarrow$ Density (kg/m³) $\quad D \rightarrow$ Diameter (m)

$Q \rightarrow$ Discharge (m³/sec)

PELTON WHEEL TURBINE

1. Velocity Triangle

(i) At Inlet $\quad$ **(ii) At outlet**



$V_1 = V_{w_1}$

$V_{F_1} = 0$

Here, $\alpha = 0°$

$\theta = 0°$

- $\beta > 90°$
- $V_{r_2} = kV_{r_1}$
- k = Friction factor
- k = 1

For frictionless vane

- $|V_{r_2}\cos\phi| = |U_2| + |V_{w_2}|$

2. Power

(i) H.P = WQH ... kW

$= \frac{\rho QH}{75}$ ... H.P

H.P = Hydraulic power.

(ii) Jet power or k.E/sec of Jet $= \frac{1}{2}\rho Q v_1^2 = \frac{1}{2}\cdot\frac{wQ}{g}\cdot v_1^2$

(iii) $R.P = \frac{wQ}{g}[V_{w_1} + V_{w_2}]u$

R.P = Runner power

(iv) S.P = R.P – Mechanical friction losses.

> **NOTE** H.P > K.E/sec. of Jet > R.P > S.P

3. Efficiencies

(i) $\eta_{nozzle} = \frac{k\cdot E/\sec\text{ of jet}}{H\cdot P\text{ available of base of nozzle}}$

$$= \frac{V_1^2}{2gH} = C_V^2 \qquad \left[\because v_1 = C_v\sqrt{2gH}\right]$$

where, η_{nozzle} = nozzle efficiency.

(ii) $\eta_h = \frac{R\cdot P}{k\cdot E/\sec\text{ of Jet}}$

$$\eta_h = \frac{2[V_{w_1} + V_{w_2}]u}{v_1^2} = \frac{2(v_1 - u)(1 + k\cos\phi)u}{v_1^2}$$

$$\eta_h|_{max} = \frac{(1 + k\cos\phi)}{2}$$

where, η_h = Hydraulic efficiency.

$\eta_h|_{max}$ = Maximum hydraulic efficiency.

(iii) $\eta_m = \frac{S\cdot P}{H\cdot P}$ where, η_m = Mechanical efficiency.

(iv) $\eta_o = \frac{S\cdot P}{H\cdot P} = \eta_{nozzle}\cdot\eta_h\cdot\eta_{mechanical}$ where, η_o = overall efficiency

4. Design Criteria

(i) $\phi = \frac{u_1}{\sqrt{2gH}}$ 0.45 to 0.47 where, ϕ = Speed ratio.

(ii) $\psi = \frac{V_{f_1}}{\sqrt{2gH}}$ = zero where, ψ = Flow ratio

(iii) $m = \frac{D}{d} = 11$ to 15 where, m = Jet ratio

D = Dia of pitch circle

d = Dia of jet.

(iv) $n = \left(15 + \frac{m}{2}\right)$ this is Tygon formula.

= 18 to 25 where, η = Number of vanes

(v) $\text{No. of jet} = \frac{\text{Total discharge through penstock}}{\text{Discharge through each jet}} \not> 6$

KEPLAN AND PROPELLER TURBINE (AXIAL FLOW REACTION TURBINE)

1. $Q = A_{F_1}.V_{F_1} = A_{F_2}.V_{F_2}$ where, Q = Discharge

$$A_{F_1} = \frac{\pi}{4}(D_0^2 - D_b^2)$$

2. $U_1 = U_2 = \frac{\pi DN}{60}$ where, $D = D_o$ at expressed edge

$D = D_b$ at inner edge

and $D = \frac{D_o + D_b}{2}$ at mid point

D_o = Outer dia of runner

D_b = Dia of hub or boss.

> **NOTE** The analysis of velocity triangle, powers. Are Similar to that of francies turbines. In this case generally $\theta > 90°$ and $V_{w_2} = 0$.

Master formular (Beronoulies principle) can be applied.

3. Design Parameter

(i) $\phi = \frac{U_1}{\sqrt{2gH}}$ is of the order of 2.0

(ii) $\psi = \frac{V_{f_1}}{\sqrt{2gH}} = 0.5$ to 0.7

(iii) Number of vanes on the runner are generally 3 to 8.

Impulse turbine (Pelton): High head and low discharge.
Francis turbine: Medium head and medium discharge.
Kaplan and Propeler turbine: Low head and high discharge.

Turbine	Specific speed, N_s,(MKS)
1. Pelton wheel turbine (single jet)	10 - 35
2. Pelton wheel turbine (multiple jet)	35 - 60
3. Francis turbine	60 - 300
4. Kaplan turbine	> 300

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