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Exclusive GATE Coaching by IIT/IISc Graduates

A division of PhIE Learning Center

QUICK REVISION

FORMULA SHEET

for

GATE -ME FLUID MACHINERY

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OUR COURSES

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Fluid Machinery

Chapter 1: IMPACT OF JETS

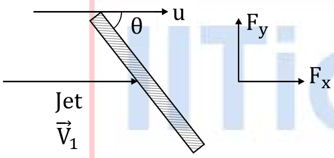
Force = Rate of change of linear momentum.

Torque = rate of change of angular momentum.

$F_{jet} = -F_{plate} = \dot{m}(\vec{V}_1 - \vec{V}_2)$

$\dot{m} =$ Mass flow rate striking the plate.

Flat Plate:



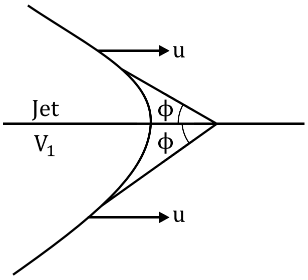
$F_x = \rho a(V_1 - u)^2 \sin^2\theta$

$F_y = \rho a(V_1 - u)^2 \sin\theta\cos\theta$

Work done (WD) $= \rho A(V_1 - u)^2 \sin^2\theta \cdot u$

a = area of jet

Symmetric Curved Plate; Jet Striking at Centre:



$F_x = \rho a(V_1 - u)^2(1 + \cos\phi)$

$F_y = 0$

$W_D = F_x . u$

Fixed Curved Plate (Unsymmetrical) Jet Enters tangentially:

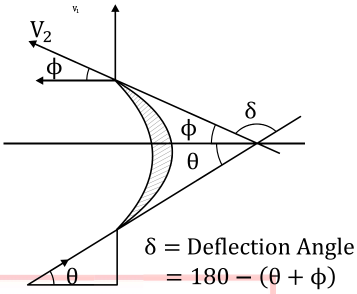
$\dot{m} = \rho a V_1$

$F_x = \rho a V_1^2(\cos\theta + \cos\phi)$

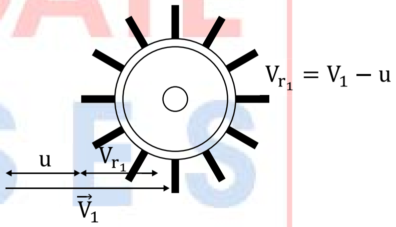
$F_y = \rho a V_1^2(\sin\theta - \sin\phi)$

$\theta =$ Vane angle at inlet

$\phi =$ Vane angle at outlet



Flat Plate Mounted on Wheel:



$\dot{m} = \rho a V_1$

V_{r_1} For single vane

V_1 for multi vane

$u_1 = u_2$

$F_x = \rho a V_1(Vr_1)$ & $F_y = 0$

$WD = \rho a V_1 (Vr_1) u$

$$\eta = \frac{\rho a V_1(V_1 - u)\cdot u}{\frac{1}{2}\dot{m}V_1^2}$$

$$\eta_{max} = 50\% \text{ at } u = \frac{V_1}{2}$$

Curved Plate Mounted on Wheel & jet striking at Center:

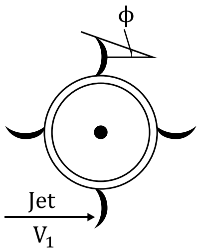
$\dot{m} = \rho a V_1$

$F_x = \dot{m}V_{r_1} - \dot{m}(-V_{r_2}\cos\phi)$

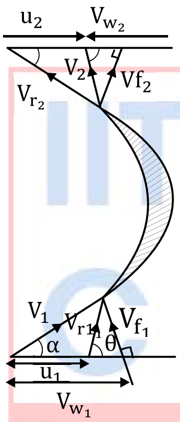
$F_x = \dot{m}(V_1 - u)(1 + \cos\phi)$

$V_{r_2} = V_{r_1}$ (If no friction)

$$\eta_{max} = \frac{1 + \cos\phi}{2} \quad \text{at } u = \frac{V_1}{2}$$



Curved Plate Mounted on Wheel, Jet Enters tangentially:



$\vec{V} = \vec{V}_r + \vec{u}$

$V^2 = V_w^2 + V_f^{\,2}$

(Runner Power)

$RP = \dot{m}\left(V_{w_1}u_1 + V_{w_2}u_2\right)$

It is General Diagram for turbine:

$$u_1 = \frac{\pi D_1 N}{60}, \qquad u_2 = \frac{\pi D_2 N}{60}$$

$V_{r_2}, V_{r_1} \rightarrow$ Relative velocities

V_{w_1} and $V_{w_2} \rightarrow$ Whirl velocities

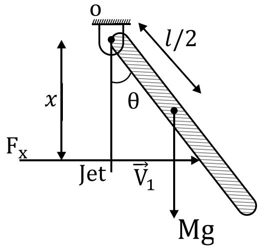
i.e., Tangential component of V_1 & V_2

V_{f_1} and $V_{f_2} \rightarrow$ Flow velocity

$\alpha, \beta \rightarrow$ angle made by absolute velocities at Inlet and exit.

$\theta, \phi \rightarrow$ Vane angle at inlet and Exit

Vertically hinged Plate



$\sum M_o = 0$

$$F_x(x) = Mg\left(\frac{l}{2}\right)\sin\theta$$

$$\rho a V_1^2(x) = W\left(\frac{l}{2}\right)\sin\theta$$

$$\sin\theta = \frac{2 \times \rho A V_1^2(x)}{Wl}$$

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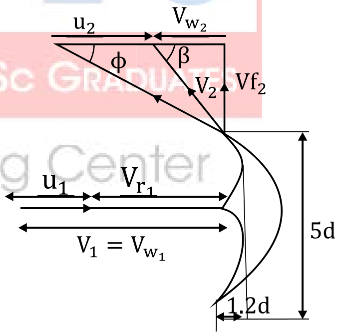
Chapter 2: TURBINES

1st Type Tangential Flow:

Impulse Turbine (Pelton Turbine)



$$u_1 = u_2 = \frac{\pi D N}{60}$$

$V_{r_1} = V_{r_2}$

$H_{net} = H_G - h_f$

$H_G =$ Gross head

$h_f =$ Head loss due to frition

Depth vane = 1.2 d

Width of vane = 5d

$$\text{Jet ratio (m)} = \frac{\text{Diameter of wheel}}{\text{Jet diameter}}$$

$$= \frac{D}{d}$$

$$\text{No. of Vanes} = \frac{m}{2} + 15$$

Water Power (WP) = $\rho g Q H_{net}$

Runner power (RP) $= \dot{m}(V_{w_1} \pm V_{w_2})u = F_x.u$

$\phi =$ Blade outlet angle /side clearance angle.

$$\eta_{nozzle} = \frac{KE}{\rho g Q H_{net}} = \frac{\frac{1}{2}\rho Q V_1^2}{\rho g Q H_{net}}$$

If no losses in Nozzle

$$V_1 = \sqrt{2g\,H_{net}}$$

If loss in Nozzle

$$V_1 = C_V\sqrt{2gH_{net}}$$

$$\eta_{Hydraulic} = \frac{\text{Runner power (RP)}}{\text{Water power (WP)}}$$

$$= \frac{(V_{w_1} \pm V_{w_2})u\,\dot{m}}{\dot{m}gH_{net}}$$

$$\eta_{vol} = \frac{Q - \Delta Q}{Q}$$

$$\eta_{Blade/wheel} = \frac{\text{Runner power (RP)}}{\frac{1}{2}\dot{m}V_1^2}$$

$$\eta_{Mechanical} = \frac{\text{Shaft Power (SP)}}{\text{Runner power(RP)}}$$

$$\eta_{overall} = \eta_{Hyd} \times \eta_{mech} \times \eta_{volumetric}$$

When η_{nozzle} is given:

Then $(\eta_{max})_{Hydraulic}$

$$= \left(\frac{1 + K\cos\phi}{2}\right)\eta_{nozzle}$$

Where

$$K = \frac{V_{r_2}}{V_{r_1}}$$

K = Blade friction coefficient

When $\eta_{nozzle} = 100\%$

$$(\eta_{max})_{Hydraulic} = \frac{1 + K\cos\phi}{2} \text{ at } u = \frac{V_1}{2}$$

$\delta =$ angle of deflection $= 180 - \phi$

Note:

Above analysis is for single nozzle for multi nozzle.

n = No. of Jet

$$= \frac{\text{Total discharge from penstock}}{\text{Discharge through 1 Jet}}$$

$$(\text{Power})_{Total} = n \times (\text{Power})_{1jet}$$

$$\text{Speed ratio} = (k_u) = \frac{u_1}{\sqrt{2gH_{net}}}$$

$$f = \frac{PN}{120}$$

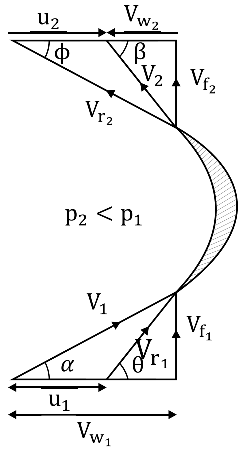
Where P = No. of poles

N = rpm, f – Freguency (Hz)

2nd Type Impulse Reaction Turbine:

$$\text{Now, } H_{net} \neq \frac{V_1^2}{2g}$$

$$H_{net} = \frac{V_1^2}{2g} + \frac{p_1}{\rho g} \qquad H_{net} = H$$



$V_{r_2} >> V_{r_1} \qquad u_1 \neq u_2$

α = Guide Blade angle or absolute velocity angle at inlet.

$A_{f_1} = \pi d_1 b_1$

$A_{f_2} = \pi d_2 d_2$

$Q = A_{f_1} V_{f_1} = A_{f_2} V_{f_2}$

$Q = (\pi d - nt)b\ V_f = k\pi d b V_f$

k = Coefficient of Vane thickness.

$$\text{Speed ratio} = \frac{u_1}{\sqrt{2gH}}$$

$$\text{Flow Ratio} = \frac{V_{f_1}}{\sqrt{2gH}}$$

Degree of Reaction (DOR):

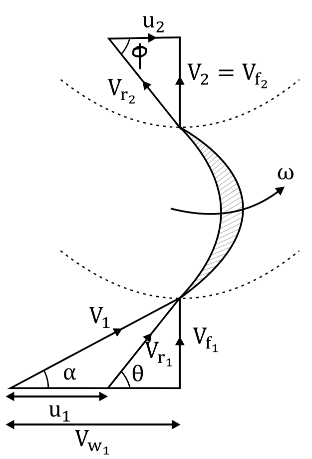
$$\text{DOR} = \frac{\text{Contribution of pressure head in } \left(\frac{\text{Runner Power}}{\dot{m}g}\right)}{\text{Total contibution of KE and Pressure energy head into } \left(\frac{RP}{\dot{m}g}\right)}$$

$$\text{DOR} = 1 - \frac{V_1^2 - V_2^2}{2g\left(\frac{R.P}{\dot{m}g}\right)}$$

Special type of Impulse Reaction Turbine.

Francis Turbine (Radial flow Turbine)

$V_{w_2} = 0;\qquad \beta = 90°$



Most Approximate Equation:

$$H = \frac{V_2^2}{2g} + \frac{RP}{\dot{m}g}$$

Above equation used

When question gets stacked

When no friction

When $V_{w_2} = 0$

Water Power (WP) = $\rho g Q H$

$RP = \rho Q(V_{w_1}u_1 + V_{w_2}u_2) \quad V_{w_2} = 0$

$= \rho Q(V_{w_1}u_1)$

$$\eta_{Hydraulic} = \frac{RP}{WP} = \frac{V_{w_1}u_1}{gH}$$

$$\text{Flow ratio}(K_f) = \frac{V_{f_1}}{\sqrt{2gH}}$$

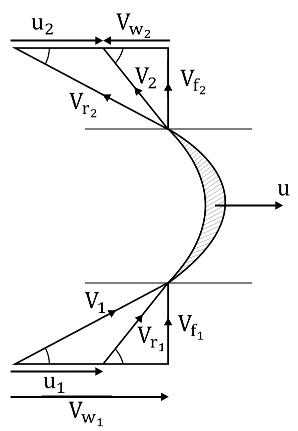
$$\text{Width ratio} = \frac{b_1}{d_1}$$

$$\text{Diameter ratio} = \frac{d_1}{d_2}$$

$$\text{Speed ratio}(K_u) = \frac{u_1}{\sqrt{2gH}}$$

Axial Flow Turbine:

Propeller (Fixed vanes) or Kaplan Turbine (Adjustable vane)



$$u_1 = u_2 = \frac{\pi DN}{60}$$

D = Taken where Analysis its done.

Area of flow $(A_f) = \frac{\pi}{4}(D_o^2 - D_h^2)$

$$\left.\begin{aligned} A_{f_1} &= A_{f_2} \\ V_{f_1} &= V_{f_2} \end{aligned}\right\} \text{Always}$$

(Rest Calculations are same as Francis)

Draft Tube (DT):

$$\eta_{\text{Draft Tube}} = \frac{\text{Change in kinetic energy head in DT}}{\text{Total kinetic energy head at entry of DT}}$$

Specific Speed of Turbine (N_s):

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} \left(\begin{matrix}\text{Valid for single} \\ \text{stage turbine}\end{matrix}\right)$$

Where N in rpm, P in kW, H in meter

Dimensionless Specific Speed of Turbine:

$$K_{S_T} = \frac{N\sqrt{P}}{\rho^{\frac{1}{2}}(gH)^{5/4}}$$

Where N in rps, P in Watt, H in meter

Specific Speed	Turbine	H	Q
0-60	Pelton	High	Low
60-300	Francis	Medium	Medium
300-600	Propeller	Low	High
600-1000	Kaplan	Low	High

Model-Prototype Relations (Valid for both turbine and Pump)

$$\frac{P}{D^5N^3} = K, \quad \frac{H}{D^2N^2} = k,$$

$$\frac{Q}{D^3N} = k$$

Unit Quantities (Used for a single Turbine):

$$N_u = \frac{N}{\sqrt{H}}, P_u = \frac{P}{H^{\frac{3}{2}}}$$

$$Q_u = \frac{Q}{\sqrt{H}}$$

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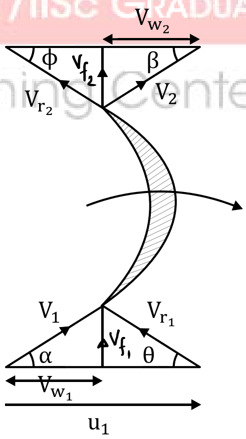
Chapter 3: PUMPS

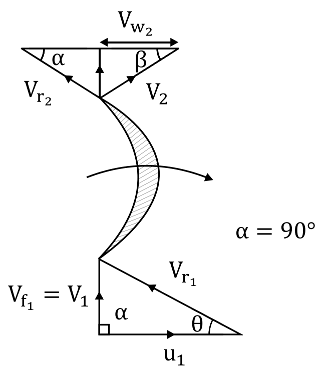
$\Delta P \propto \omega^2$

Centrifugal pump works on principle of forced Vortex.

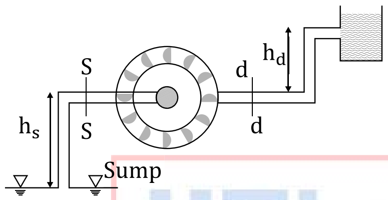
General Pump:



Centrifugal Pump:



$V_{w_1} = 0$



H_m = Manometric head (Head Required to pump the water)

$IP = \rho Q(V_{w_2} u_2 - V_{w_1} u_1)$

$IP = \rho Q(V_{w_2} u_2) \rightarrow$ For Centrifugal pump

$$\eta_{manometric} = \frac{\text{Water Power (WP)}}{\text{Impeller Power (IP)}}$$

$WP = \rho g Q H_m$

$Q = A_{f_1} V_{f_1} = A_{f_2} V_{f_2} \quad A_f = \pi d b$

$$\eta_{vol} = \frac{Q}{Q + \Delta Q}$$

$$\eta_{mech} = \frac{IP}{\text{Shaft Power (SP)}}$$

$$\boxed{IP = SP - \text{Mechanical Losses}}$$

$$\frac{IP}{\dot{m} g} = \frac{V_{w_2} u_2 - V_{w_1} u_1}{g} = H_e$$

H_e = Euler head

$$\text{Speed ratio } (Ku_2) = \frac{u_2}{\sqrt{2gH_m}}$$

$$\text{Flow ratio } (K_f) = \frac{V_{f_2}}{\sqrt{2gH_m}}$$

$$\text{Diameter ratio} = \frac{d_1}{d_2}$$

Specific Speed of Pump

$$(N_S) = \frac{N\sqrt{Q}}{H_m^{3/4}}$$

Where N is in rpm, Q is in m^3/sec, H_m in meter.

General Equation:

$$\frac{IP}{\dot{m} g} = \frac{V_2^2 - V_1^2}{2g} + \frac{u_2^2 - u_1^2}{2g} + \frac{V_{r_1}^{\,2} - V_{r_2}^{\,2}}{2g}$$

At starting, $V_1, V_2, V_{r_1}, V_{r_2} = 0$

$$\frac{IP}{\dot{m} g} = \frac{u_2^2 - u_1^2}{2g} \geq H_m$$

$$\frac{\omega^2 (r_2^2 - r_1^2)}{2g} \geq H_m$$

Where ω = Minimum speed of pump to start.

Static Head $(H_s) = h_s + h_d$

h_s = Suction head

h_d = Delivery head

If there is no loss in pump

$(\eta_{manometric} = 100\%)$

$$H_m = \frac{IP}{\dot{m} g} = \frac{V_{w_2} u_2}{g}$$

If loss in pump is given

$$H_m = \frac{V_{w_2} u_2}{g} - \begin{pmatrix}\text{Loss in impeller +} \\ \text{loss in casing}\end{pmatrix}$$

$$H_m = H_s + H_f + \frac{V_d^2}{2g}$$

$$NPSH = \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} - \frac{p_v}{\rho g}\right), \; h_v = \frac{p_v}{\rho g}$$

$$NPSH = \frac{p_{atm}}{\rho g} - h_s - h_f - h_v$$

NPSH= Net positive suction head

$$\text{Cavitation factor } (\sigma) = \underbrace{\frac{NPSH}{H_m}}_{\text{Pump}} = \underbrace{\frac{NPSH}{H}}_{\text{Turbine}}$$

Note:

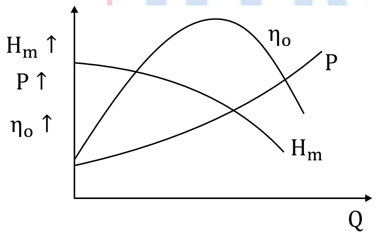
$$\sigma = \frac{NPSH}{H_m} \geq \sigma_c \text{ for no cavitation}$$

where σ_c = Critical cavitation factor

Dimensionless Specific speed of pump.

$$K_{sp} = \frac{N\sqrt{Q}}{(gH)^{3/4}}$$

Characteristics Curve:



Multiple Pump:

For Series

$H_m = nH$

$Q = Q_1 = Q_2$

For Parallel, $Q = Q_1 + Q_2 + Q_3 + \cdots$

$H_m = H_1 = H_2 = H_3 \ldots.$

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