Lecture on
Francis Turbine

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Turbines: Francis (1849)

\[ Q_o = v_i \frac{\pi d_i^2}{4} \]

\[ R_a = R_i + \frac{\theta}{2\pi} d_i \]

\[ v_i = K_v \sqrt{2gH} \]

\[ Q = \frac{\theta}{2\pi} Q_o \]
Draft Tube:
Static pressure \( \left( \frac{P}{\gamma} + Z \right) \) gradually decreases when water glides over the runner blades. Water coming out of the runner possesses large amount of Kinetic energy and pressure at runner outlet, which is less than atmospheric pressure.

1. It makes possible to install the turbine above tail race without loss of head
2. The pressure at the exit of the draft tube is atmospheric.
3. Avoid cavitation, arrest separation of water.

Apply Bernoulli’s theory at runner exit (2) and exit of draft tube (4)

\[
\frac{P_2}{\gamma} + \frac{v_2^2}{2g} + Z_2 = \frac{P_4}{\gamma} + \frac{v_4^2}{2g} + Z_4 + hf
\]

\( hf = \) head loss through draft tube

\( \ldots \ldots \text{eq (1)} \)

Apply Bernoulli’s theory at free surface (3) and exit of draft tube (4)

\[
\frac{P_3}{\gamma} + \frac{v_3^2}{2g} - Z_3 = \frac{P_4}{\gamma} + \frac{v_4^2}{2g} + Z_4
\]

Same, as difference is negligible

\( \ldots \ldots \text{eq (2)} \)

\[
\frac{P_4}{\gamma} = \frac{P_3}{\gamma} + (Z_3 - Z_4)
\]

\( \ldots \ldots \text{eq (3)} \)

Replacing the above value to eq (1)

\[
\frac{P_2}{\gamma} = \frac{P_3}{\gamma} - (Z_2 - Z_3) - \left[ \frac{v_2^2}{2g} - \frac{v_4^2}{2g} - hf \right]
\]

\( \ldots \ldots \text{eq (4)} \)

\[
\frac{P_2}{\gamma} = \frac{P_3}{\gamma} - \left[ hs + \frac{v_2^2}{2g} - \frac{v_4^2}{2g} - hf \right]
\]

\( \ldots \ldots \text{eq (5)} \)
Draft Tube:

(4) Static pressure \((P/\gamma + Z)\) at runner outlet (at the level of 2) is less than atmospheric pressure by an amount equal to the static and dynamic suction head.

(5) Velocity of water at outlet of runner is very high (3-15% of net working head, for high specific speed it is 45% in case of kaplan turbine), by employing draft tube recovers this wasted KE is utilized which increases efficiency of the turbine.

(6) Prevent splashes of water coming out of the runner.

Efficiency of the draft tube is expressed as

\[
\eta_{dt} = \frac{\frac{V_2^2 - V_4^2}{2g} - h_f}{\frac{V_2^2}{2g}}
\]

actual conversion of Kinetic Head into Pressure Head

\[
= \frac{\text{Kinetic Head at inlet of the draft tube}}{\text{Dynamic Head at inlet}}
\]

- Sometimes friction loss is expressed as

\[
h_f = k \frac{(V_2^2 - V_4^2)}{2g}, \quad k = \text{const}
\]
Governing Mechanism

- Governing Motor
- Regulating Shaft
- Regulating Lever
- Regulating Rod
- From Penstock

Connected to relay valve and oil sump

Scroll Casting

- More (openings)
- Less (openings)
- Most opening stage

Guide Valve

Regulating Ring
Runners are classified according to the speed, as per their shape and velocity triangle.

**Velocity Diagram**

**Degree of Reaction (R)**
- change in pressure energy inside the runner (Hpr)
- change in total energy inside the runner (He)

\[
Hpr = \left( \frac{V_{r2}^2}{2} - \frac{V_{r1}^2}{2} \right) + \left( \frac{u_1^2 - u_2^2}{2} \right)
\]

\[
He = \left( \frac{V_{1}^2}{2} - \frac{V_{2}^2}{2} \right) + \left( \frac{V_{r2}^2}{2} - \frac{V_{r1}^2}{2} \right) + \left( \frac{u_1^2 - u_2^2}{2} \right)
\]

**Relationship:**
\[
\frac{V_{r2}^2}{2} = \frac{V_{r1}^2}{2} + \left( \frac{u_2^2 - u_1^2}{2} \right)
\]

Centrifugal head (CFH)

Outward flow: \( u_2 > u_1 \), CFH +ve, \( V_{r2} \) increases at outlet
Inward flow: \( u_1 > u_2 \), CFH -ve, \( V_{r2} \) decreases at outlet
Francis Turbine Equations

Working Proportions:
(1) Ratio to width to diameter (n’) = b1/D1 = 0.1 to 0.45, for slow runner flow is predominantly radial and exit is axial
(2) Speed ratio (Ku) = \( \frac{u_1}{\sqrt{2gH}} \) = 0.6 to 0.9
(3) Flow ratio (kf) = \( \frac{V_{f1}}{\sqrt{2gH}} \) = 0.15 to 0.30
(4) Coeff of velocity (kv) = \( \frac{V_{1}}{\sqrt{2gH}} \) = 0.97 to 0.99

Design Parameters: Head (H), running speed (N), Power output (P) is required size of the runner and its vane angle is to be find out.
(1) Assume probable value of \( \eta_h \), \( \eta_o \), n’ and ratios (Ku, kf, kv)
(2) Workout the mass or volumetric flow rate, where shaft power P = \( \eta_o \gamma QH \)
(3) Flow opening area A1 = \( \pi D_1 b_1 - Z t_1 b_1 \) = \( \pi D_1 b_1 K_1 \) where Z is number of blades, \( t_1 \) is thickness of runner at inlet, \( b_1 = \) width of the runner at inlet, and \( K_1 \) is vane thickness coeff = approx 0.95
(4) Flow velocity \( V_{f1} = \frac{Q}{\pi D_1 b_1 K_1} \), by continuity equation \( Q = \pi D_1 b_1 K_1 V_{f1} = \pi D_2 b_2 K_2 V_{f2} \)
(5) \( u_1 = \pi DN/60 \)
(6) Work done by turbomachine = \( \rho Q(V_{w1} u_1 \pm V_{w2} u_2) \)
(7) Hydraulic Efficiency (\( \eta_h \)) assuming radial exit = \( V_{w1} u_1 / gH \) [\( V_{w2} \) is zero]
(8) Head supplied to turbine = Work Done or head utilized at runner + Kinetic head at exit
\[
H_t = \frac{\rho Q(V_{w1} u_1 \pm V_{w2} u_2)}{\rho g Q} + \frac{V^2_2}{2g} \]
again gross head \( H_g = H_t + \sum h_i \) whereas for pump \( H_p = H_g - \sum h_i \)

Key Term:
Without whirl \( \rightarrow V_w = 0 \)
Negligible blade thickness \( \rightarrow K_1 = K_2 = 1 \)
Radial inlet \( \rightarrow \theta = 90^\circ \)
Radial exit \( \rightarrow \beta = 90^\circ \); \( V_2 = V_{f2} \)
No blade friction \( \rightarrow V_{r1} = V_{r2} \)