Lecture on

Francis Turbine

by

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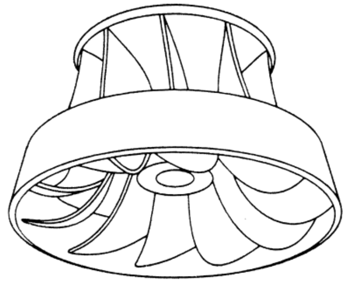
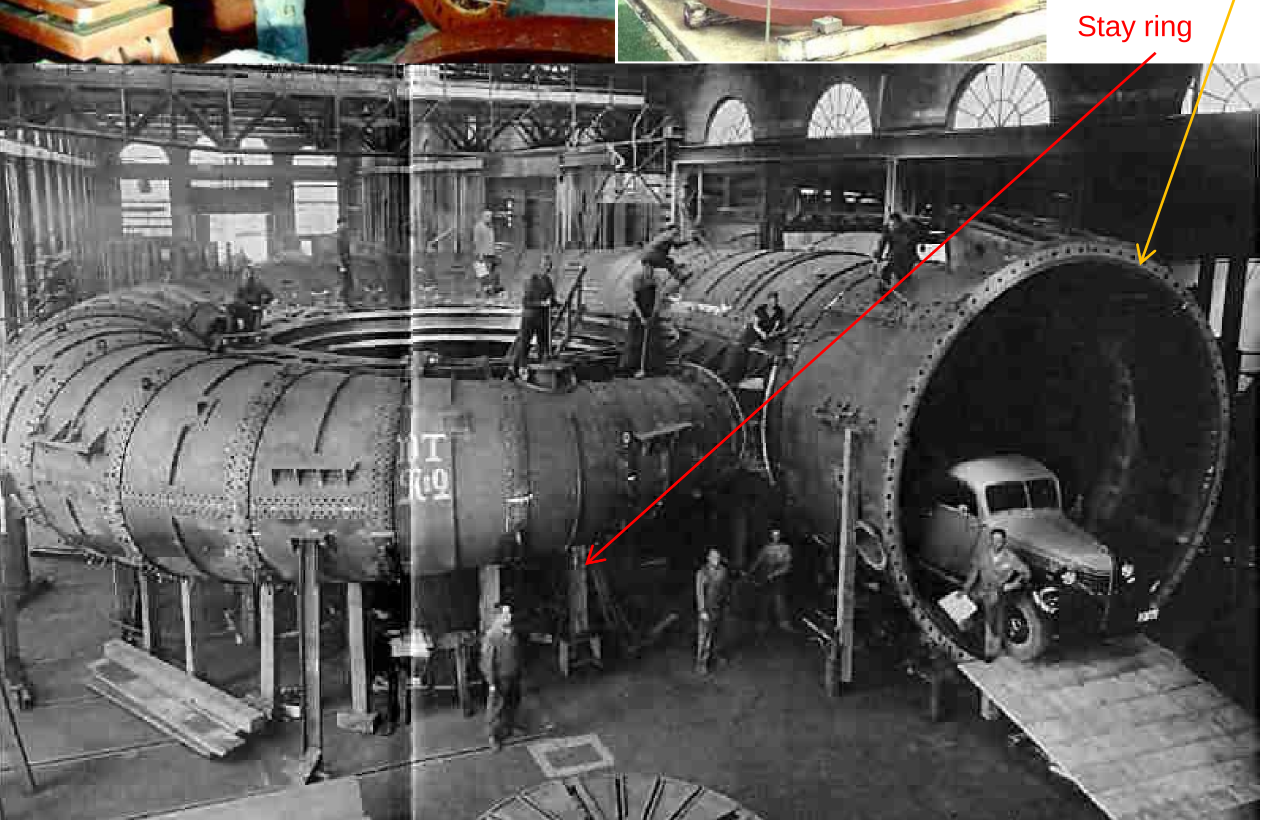
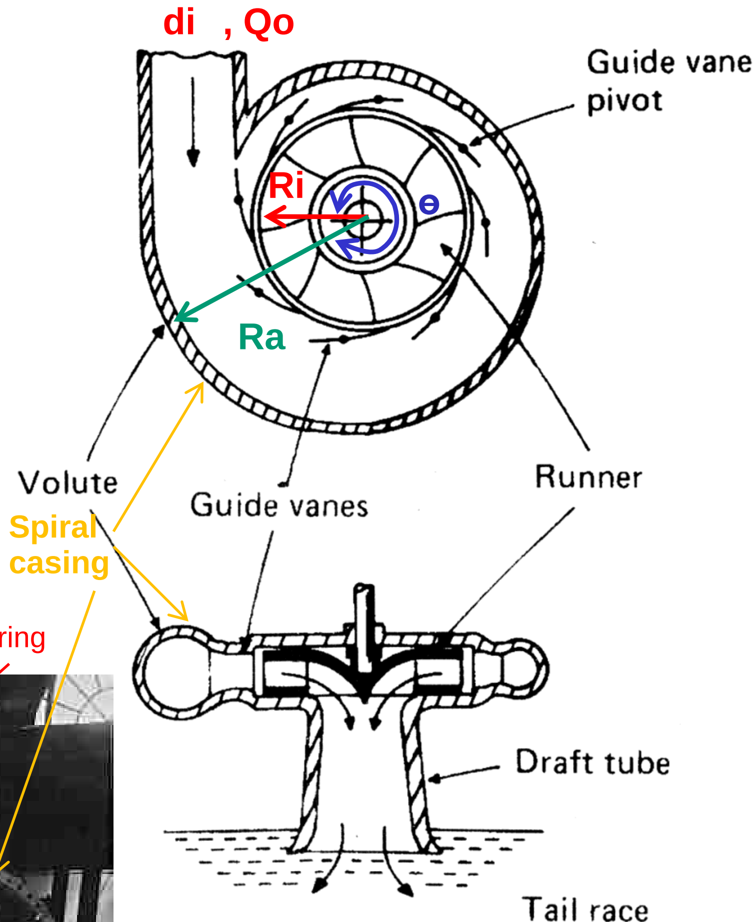
Turbines: Francis (1849)







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

$$v_i = K_v \sqrt{2gH}$$

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

$$Q = \frac{\theta}{2\pi} Q_o$$

Draft Tube:

Static pressure (P/γ+Z) gradually decreases when water glides over the runner blades. Water coming out of the runner posses 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 Bernouli's theory at runner exit (2) and exit of draft tube (4)

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

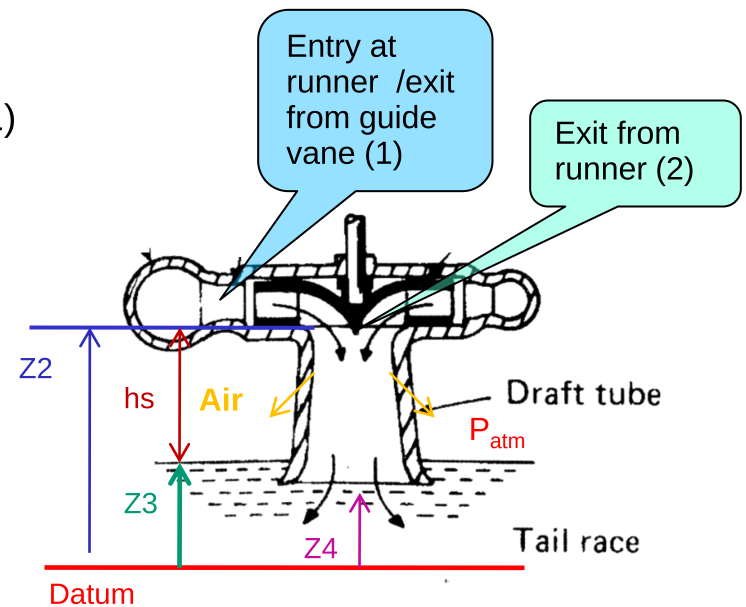
hf = head loss through draft tube ....eq (1)

Apply Bernouli'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 ....eq (2)

$$\frac{P_4}{\gamma}=\frac{P_3}{\gamma}+(Z_3-Z_4)$$ ....eq (3)



Replacing the above value to eq (1)

$$\frac{P_2}{\gamma}=\frac{P_3}{\gamma}-(Z_2-Z_3)-\left[\frac{v_2^2}{2g}-\frac{v_4^2}{2g}-hf\right]$$ ....eq (4)

$$\frac{P_2}{\gamma}=\frac{P_3}{\gamma}-\left[hs+\frac{v_2^2}{2g}-\frac{v_4^2}{2g}-hf\right]$$ ....eq (5)

Static suction head

dynamic suction head

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.

$V_2^2/2g$

(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 .

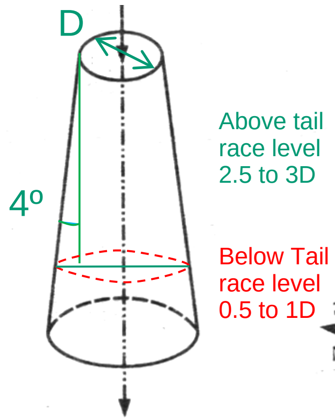
Efficiency of the draft tube is expressed as

$$\eta_{dt} = \frac{\dfrac{(V_2^2 - V_4^2)}{2g} - h_f}{\dfrac{V_2^2}{2g}}$$

$$= \frac{\text{actual conversion of Kinetic Head into Pressure Head}}{\text{Kinetic Head at inlet of the draft tube}}$$

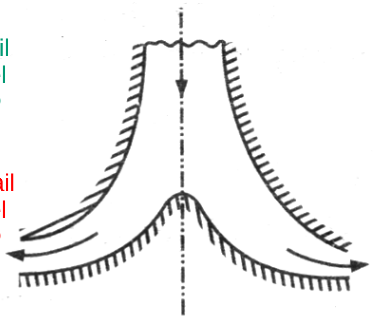
Sometimes friction loss is expressed as

$$h_f = k\frac{(V_2^2 - V_4^2)}{2g}$$, k = const



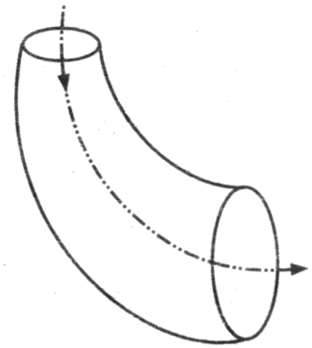
(a) Straight divergent

Prevent possibility of flow separation η_{dt}=90%, angle less than 8º



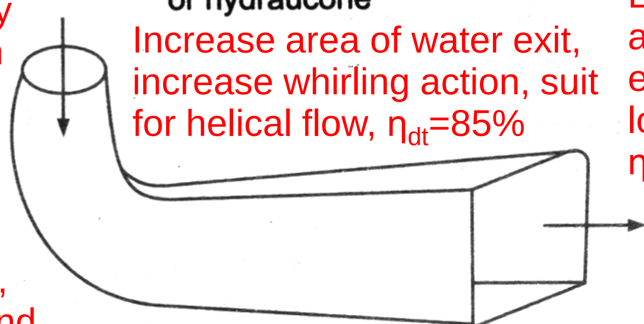
(b) Moody spreading or hydraucone

Increase area of water exit, increase whirling action, suit for helical flow, η_{dt}=85%



(c) Simple elbow

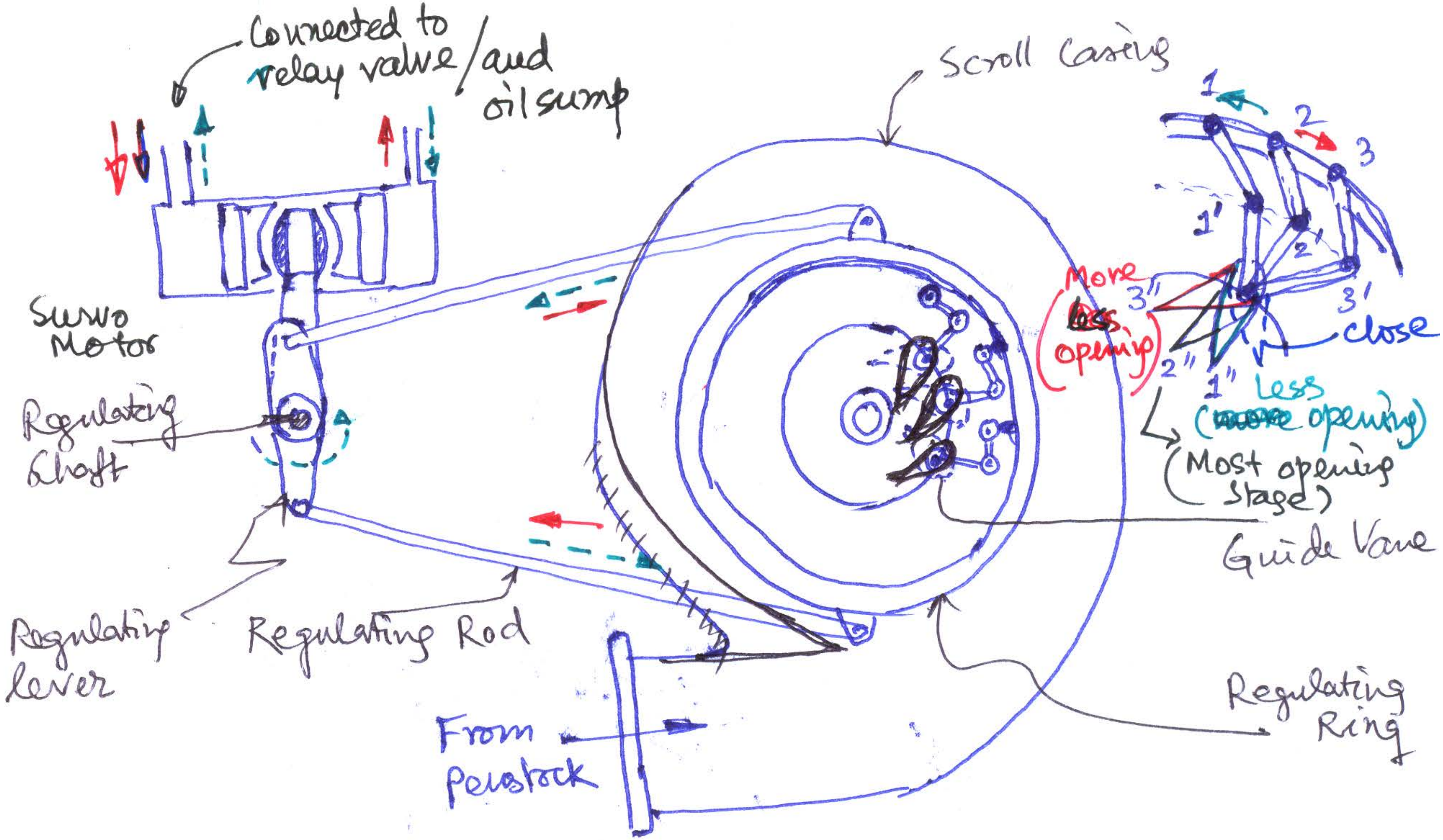
Length minimum and less excavation, head loss at bend, η_{dt}=60%



(d) Elbow with cross-section changing from circle to rectangle

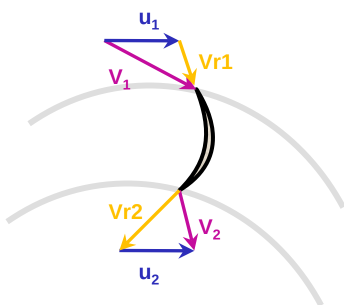
less excavation, head loss at bend, η_{dt}=85%

Governing Mechanism

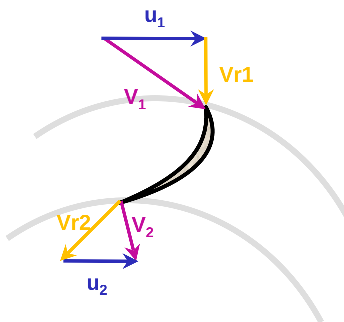


Velocity Diagram

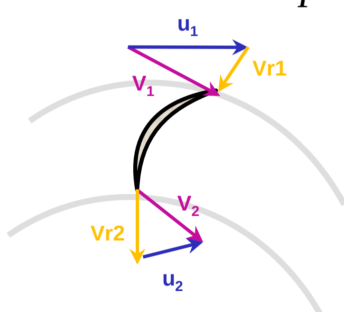
Runners are classified according to the speed, as per their shape and velocity triangle



Slow runner



Medium runner



Fast runner

Degree of Reaction (R)

$$R = \frac{\text{change in pressure energy inside the runner (Hpr)}}{\text{change in total energy inside the runner (He)}}$$

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

Rotation

$$He = \left(\frac{V_1^2}{2} - \frac{V_2^2}{2}\right) + \left(\frac{V_{r2}^{\;2}}{2} - \frac{V_{r1}^{\;2}}{2}\right) + \left(\frac{u_1^2 - u_2^2}{2}\right)$$

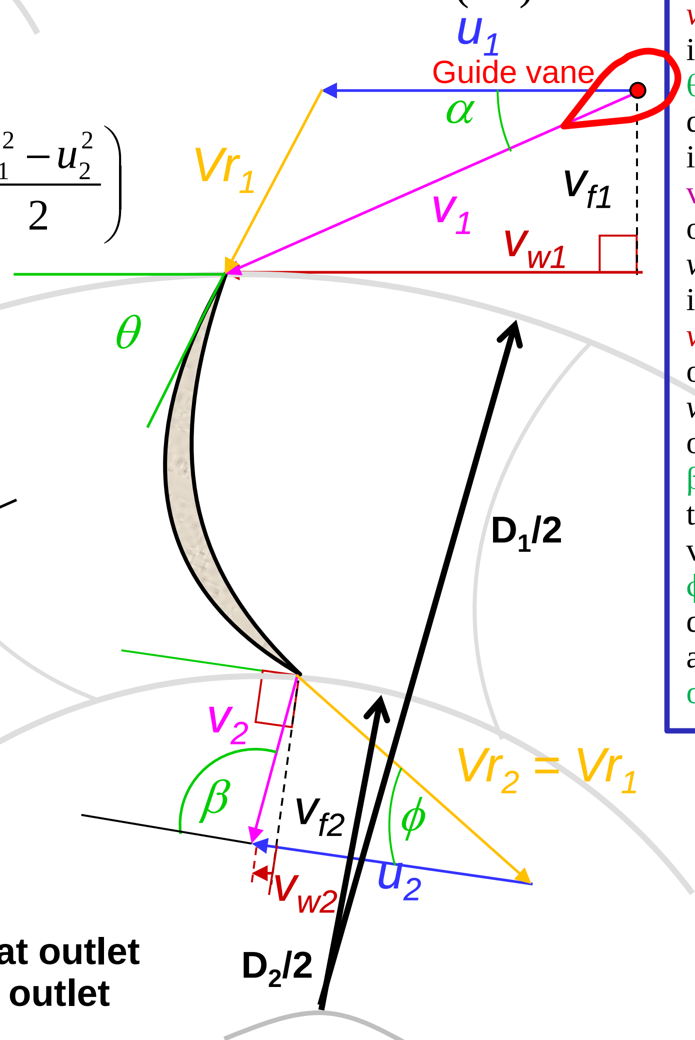
Relationship:

$$\frac{V_{r2}^{\;2}}{2} = \frac{V_{r1}^{\;2}}{2} + \left(\frac{u_2^2 - u_1^2}{2}\right)$$

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



v_1 = velocity of fluid at inlet
u_1 = velocity of the vane at inlet
v_{r1} = relative velocity of fluid at inlet
α = angle between the direction of the fluid and the direction of motion of the vane, guide blade angle
v_{w1} = velocity of whirl at inlet
θ = angle made by v_{r1} with direction of motion at inlet, vane angle at inlet
v_1 = velocity of fluid at outlet
v_{f1} = velocity of flow at inlet
v_{w2} = velocity of whirl at outlet
v_{f2} = velocity of flow at outlet
β = angle between v_2 with the direction of motion of vane at outlet
ϕ = angle made by v_{r2} with direction of motion of vane at outlet, vane angle at outlet

Francis Turbine Equations

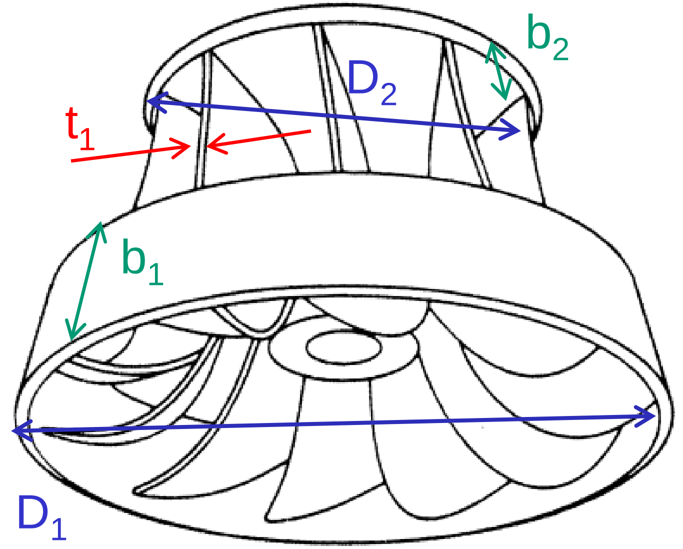
Working Proportions:

(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) $= u_1 / \sqrt{2gH}$ = 0.6 to 0.9

(3) Flow ratio (kf) = $V_{f1} / \sqrt{2gH}$ = 0.15 to 0.30

(4) Coeff of velocity (kv)= $V_1 / \sqrt{2gH}$ = 0.97 to 0.99



Key Term:
Without whirl → V_w=0
Negligible blade thickness → K_1=K_2=1
Radial inlet → θ=90º
Radial exit → β=90º; V2=Vf2
No blade friction → Vr1=Vr2

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 η_h, η_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 A_1= $\pi D_1 b_1 - Z t_1 b_1$, $= \pi D_1 b_1 K_1$ 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}= $Q/ \pi D_1 b_1 K_1$, by continuity equation Q= $\pi D_1 b_1 K_1 V_{f1}$ = $\pi D_2 b_2 K_2 V_{f2}$

(5) u1=πDN/60

(6) Work done by turbomachine $= \rho Q (V w_1 u_1 \pm V w_2 u_2)$,

(7) Hydraulic Efficiency (η_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_i= $p_1/\rho g + V_1^2/2g$

$$H_t = \frac{\rho Q (V w_1 u_1 \pm V w_2 u_2)}{\rho g Q} + \frac{V_2^2}{2g}$$ again gross head $H_g = H_t + \sum h_l$ whereas for pump $H_p = H_g - \sum h_l$